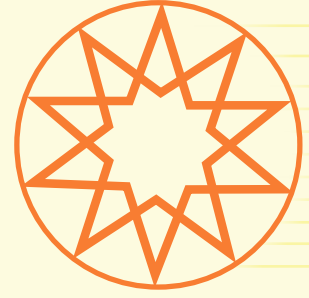


ISSN 2148-7847



Journal of Thermal Engineering

Volume 11
Number 5
Year 2025

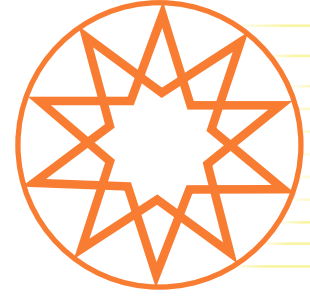
YTÜ
PRESS

www.jten.yildiz.edu.tr



Journal of Thermal Engineering

Volume 11 Number 5 Year 2025



EDITOR-IN-CHIEF

Ahmet Selim Dalkilic, *Yildiz Technical University, İstanbul, Türkiye*

ASSOCIATE EDITORS

Ali Celen, *Erzincan Binali Yıldırım University, Erzincan, Türkiye*

Baha Zafer, *İstanbul University, İstanbul, Türkiye*

Bekir Şener, *Yildiz Technical University, İstanbul, Türkiye*

Erdal Çetkin, *İzmir Institute of Technology, İzmir, Türkiye*

Emre Alpman, *Marmara University, İstanbul, Türkiye*

Erman Aslan, *İstanbul University, İstanbul, Türkiye*

Hasan Köten, *İstanbul Medeniyet University, İstanbul, Türkiye*

Hatice Mercan, *Yildiz Technical University, İstanbul, Türkiye*

Müslüm Arıcı, *Kocaeli University, Kocaeli, Türkiye*

Mustafa Kılıç, *Adana Science and Technology University, Adana, Türkiye*

Nader Javani, *Yildiz Technical University, İstanbul, Türkiye*

Naime Filiz Özgül, *Adana Science and Technology University, Adana, Türkiye*

Şevket Özgür Atayılmaz, *Yildiz Technical University, Adana, Türkiye*

Özgen Açıkgöz, *Yildiz Technical University, İstanbul, Türkiye*

Yaşar İslamoğlu, *Sakarya University, Sakarya, Türkiye*

Yasin Karagoz, *İstanbul Medeniyet University, İstanbul, Türkiye*

Zerrin Sert, *Eskişehir Osmangazi University, Eskişehir, Türkiye*

EDITORIAL BOARD

Adrian Bejan, *Duke University, United States*

Afshin J. Ghajar, *Oklahoma State University, United States*

Ali Chamkha, *Prince Mohammad Bin Fahd University, Saudi Arabia*

Bekir Sami Yilbas, *King Fahd University of Petroleum and Minerals, Dhahran, Saudi Arabia*

Chi-Chuan Wang, *National Chiao Tung University, Taiwan*

Gopal Nath Tiwari, *Indian Institute of Technology, India*

Hakan F. Oztop, *Firat University, Türkiye*

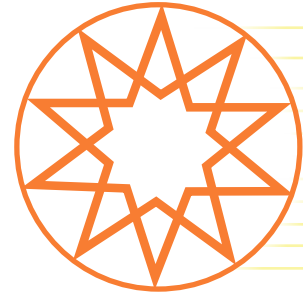
Hari Mohan Srivastava, *University of Victoria, Canada*

Hamadiche Mahmoud, *Université de Lyon 1, France*

Hamid Niazmand, *Ferdowsi University Of Mashhad, Iran*



Journal of Thermal Engineering



Volume 11 Number 5 Year 2025

Josua P. Meyer, *University of Pretoria, South Africa*
Kambiz Vafai, *University of California, United States*
Karan Surana, *The University of Texas, United States*
Mehmet Kanoğlu, *University of Gaziantep, Türkiye*
Mohammad Mehdi Rashidi, *Bu-Ali Sina University, Iran*
Mohammad Hassan Saidi, *Sharif University of Technology, Iran*
Muhammad Ali Rob Sharif, *The University of Alabama, United States*
Moh'd A. Al-Nimr, *Jordan University of Science and Technology, Jordan*
Mohsen Sheikholeslami, *Babol Noshirvani University of Technology, Iran*
Mustafa Turkyilmazoglu, *Hacettepe University, Türkiye*
N.K. Anand, *Texas A&M University, United States*
Omid Mahian, *Xi'an Jiaotong University, Xi'an, China*
Sergio Nardini, *Seconda Università degli Studi di Napoli, Italy*
Wei-Mon Yan, *National Chiao Tung University, Taiwan*

Abstracting and Indexing: Directory of Open Access Journals (DOAJ), EBSCO Host Online Research Databases, Web of Science, Emerging Sources Citation Index (ESCI), Scopus, TRDizin, MIAR, Scilit, OUCI, Research4Life, Ideal Online, Sherpa Romeo, ASCI

Journal Description: The journal is supported by Yildiz Technical University officially, and is a blind peer-reviewed free open-access journal, published bimonthly (January-March-May-July-September-November).

Publisher: Yildiz Technical University

Editor-in-Chief: Prof.Dr. Ahmet Selim Dalkılıç, dalkilic@yildiz.edu.tr

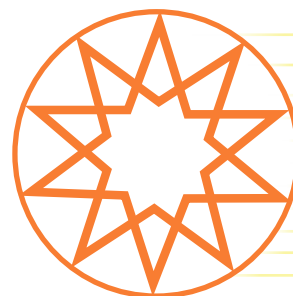
Language of Publication: English

Frequency: Bimonthly (6 issues per year)

Regular Issues: January (1) - March (2) - May (3) - July (4) - September (5) - November (6)

Publication Type: Online e-version

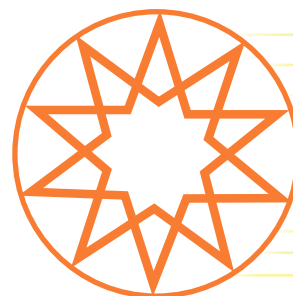
Journal of Thermal Engineering is supported by Yildiz Technical University officially, and is a blind peer-reviewed free open-access journal, published bimonthly. • Journal of Thermal Engineering is a peer reviewed open access journal established as an origin for research in Heat Transfer and Thermodynamics mainly. • Journal of Thermal Engineering make every effort to ensure the accuracy of all the information (the "Content") contained in our publications. However, Journal of Thermal Engineering, our agents, and our licensors make no representations or warranties whatsoever as to the accuracy, completeness, or suitability for any purpose of the Content. Any opinions and views expressed in this publication are the opinions and views of the authors, and are not the views of or endorsed by Journal of Thermal Engineering. The accuracy of the Content should not be relied upon and should be independently verified with primary sources of information. Journal of Thermal Engineering shall not be liable for any losses, actions, claims, proceedings, demands, costs, expenses, damages, and other liabilities whatsoever or howsoever caused arising directly or indirectly in connection with, in relation to, or arising out of the use of the Content.



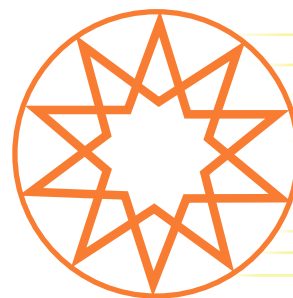
CONTENTS

Research Articles

- 1261** Enhancing VCRS performance: A study of R134a and refrigerant blends with spiral condensers and series evaporators
Sourabh KULKARNI, Shriramshastri CHAVALI, Shruti VEDPATHAK
- 1276** Harnessing response surface methodology for diesel engine optimization using titanium dioxide-enhanced rice bran biodiesel to improve emissions and efficiency
Manoj Kumar GUPTA, Ashok Kumar SRIVASTAVA
- 1293** On the role of biomimetics shark skin flow control in the aerodynamic characteristics of leading-edge protuberanced wing section
S. SMRITHIKA, S. ARUNVINTHAN
- 1312** Highly viable nanoparticle based Thermal Interface Materials (TIM) for electronics device cooling applications
S Uma MAHESHWARI, A. Brusly SOLOMON, G. THILAGAVATHI, Madhukar HEMAMALINI, D. ILLAKKIAM
- 1327** Performance of Mylar and Teflon as compound parabolic concentrators in solar desalination system
Shridhar KEDAR, Deepak WATVISAVE, Prerna MISHRA, Ganesh Vijay MORE, Adarsh Abi MATHEW, Neeta MANDHARE, Sumedh INGLE
- 1339** Thermal design and CFD simulation of an internal heat exchanger with R1234yf as an alternate refrigerant to R134a
B. K. PATEL, A D. PAREKH
- 1355** A novel approach to augment stepped solar still productivity using the heat storage characteristics of water
J.S D'COTHA, P. R. SURESH, P. SAJEESH
- 1374** Experimental adsorption studies of atmospheric water vapor by clay based composite sorbents in vertical fluidized bed
C. R. HIREMATH, Ravikiran KADOLI, R. J. TALAPATI



- 1392** Performance optimization of low-head vertical axis impulse turbine runners for nozzle angle, nozzle diameter, and nozzle standoff distance using response surface methodology
H. N. LAKDAWALA, V. K. PATEL, G. P. BAKHRU, D. N. MODI
- 1420** Advanced thermal management in aircraft lithium-ion battery packs: optimization of heat dissipation using heat spreaders and phase change materials
Srinivas MALLIMOGGALA, K Rama DEVI
- 1439** Comparative performance analysis of the Andasol-1 power plant in M'sila (an Algerian province)
Khaled BOUCHAREB, Nabila IHADDADENE, Razika IHADDADENE, Khellaf BELKHIRI
- 1455** Effect of bristled shark scale structures on the aerodynamic characteristics of leading-edge protuberanced wing section
S. SMRITHIKA, S. ARUNVINTHAN
- 1468** Experimental investigation of the impact of electronic expansion valve opening on vcr system performance: comparative analysis with capillary tube and thermostatic expansion valve
Ankitsinh CHAUHAN, Ashok PAREKH, Vimal PATEL
- 1483** Thermal performance of drying mango slices using a baffled-type hybrid solar dryer with exhaust hot air recirculation
Debashree Debadatta BEHERA, Shiv Sankar DAS, Ramesh CHANDRA MOHANTY, Raj Kumar SATANKAR
- 1497** Experimental investigations on performance improvement of multi point sequential fuel injection engine by reducing back pressure
Chandrakumar PARDHI, Anshul GANGELE, Sanjay CHHALOTRE, C.P. JAWAHAR
- 1507** Technical and economic feasibility analysis of 1kW Rooftop Solar Photovoltaic System for Mysuru, Karnataka, India
Raghavendra Deva Das RAO, Bommegowda Sadashive GOWDA



- 1520** Parametric study and performance evaluation of a multi-layered twin pcm-based vertical thermal energy storage system

Amit Kumar GHOSH, Pabitra HALDER

- 1540** Enhancing carbon neutrality through renewable energy and demand-side management

Yeqing ZHU, Yu WANG, Shen HOU, Yanjun CHEN, Luzheng ZHANG

Review Articles

- 1552** Novel approaches to zeolite deactivation mitigation and regeneration in biomass gasification

Imron ROSYADI, Suyitno, Zainal ARIFIN, Tata SUTARDI

- 1585** Factors affecting and methods of reducing thermal stratification in cryogenic storage tanks of launch vehicles: review article

Krish V. RAIBOLE, Puskaraj D SONAWWANAY



Research Article

Enhancing VCRS performance: A study of R134a and refrigerant blends with spiral condensers and series evaporators

Sourabh KULKARNI^{1,*}, Shriramshastri CHAVALI¹, Shruti VEDPATHAK^{1,2}

¹Department of Mechanical Engineering, Faculty of Engineering Technology, D Y Patil University Ambi, Maharashtra, 410507, India

²Department of Mechanical Engineering, Dr D Y Patil University Institute of Technology, Pimpri, Maharashtra, 411018, India

ARTICLE INFO

Article history

Received: 13 June 2024

Revised: 29 October 2024

Accepted: 31 October 2024

Keywords:

R134a; Refrigerant Blends; Series Evaporator Arrangement; Spiral Condenser; Vapor Compression Refrigeration System (VCRS)

ABSTRACT

The purpose of this work will be the improve the vapour compression refrigeration system, taking into account the usage of environmentally friendly refrigerants. This is important because the alternative solution for achieving a minimal invasion into the environment with maximum energy efficiency is urgently needed. The method used here, namely, system redesigns with spiral condensers and series evaporators, was validated using CoolPack software. Major outcomes are that the coefficient of performance is increased by 77% and power consumption is reduced by 19% when a spiral condenser is used instead of the conventional systems. The hydrocarbon blends of propane and isobutane were found to be possible substitutes for R134a, and using nanofluids of Al_2O_3 or ZnO was shown to increase the coefficient of performance up to 137%. The research does not only add a source of enhancing literature but also provides a practical framework for developing more sustainable and efficient refrigeration technologies.

Cite this article as: Kulkarni S, Chavali S, Vedpathak S. Enhancing VCRS performance: A study of R134a and refrigerant blends with spiral condensers and series evaporators. J Ther Eng 2025;11(5):1261–1275.

INTRODUCTION

Energy conservation is indeed a challenging concern today, especially in systems like refrigeration, where efficiency and sustainability are paramount. The VCRS serves a critical role in maintaining suitable environments and preserving perishable goods. It achieves this by leveraging fundamental thermodynamic principles to efficiently transfer heat from low-temperature spaces to high-temperature ones. Comprising components like compressors,

condensers, expansion valves, and evaporators, the VCRS operates in a continuous cycle involving compression, condensation, expansion, and evaporation processes. Each component works sequentially to circulate refrigerant, facilitating the transfer of heat.

Also in the past few years, the pressure to have safe environmental and energy-efficient refrigeration systems has motivated a lot of research into alternative refrigerants, recovery of waste heat, and innovative system modifications. For instance, illustrated the possibility of using

*Corresponding author.

*E-mail address: sourabhkdypu@gmail.com

This paper was recommended for publication in revised form by Editor-in-Chief Ahmet Selim Dalkılıç



hydrocarbon blends as a substitute to HFC-134a, in that they carry out the task with a significantly lower environmental footprint [1, 2]. More research indicates that the utilization of water-cooled condensers enhances COP by up to 13.4% as alternative refrigerants utilize this, yet their system modification provides considerable performance enhancements in energy efficiency [2]. Furthermore, applied the humidification-dehumidification desalination process, incorporating waste heat recovery, where such configurations improve COP to 7.6 and, in addition, produce distillate water [3].

Despite this, Investigated and found that although mixtures of R290 and R600a in any ratio increase refrigeration effects, there is not much knowledge regarding their performance when combined with highly advanced condenser designs like spiral condensers [4]. Another potential area is nanofluids by using Al_2O_3 and ZnO wherein primary experiments exhibit increased heat transfer and efficiency, but which has not been tested under different combinations with environmentally friendly refrigerants under modified VCRS configurations. These developments are taken one step further by incorporating spiral condenser design and nanofluids in the process, thus an integrated assessment of the interactions on system performance and operational conditions is provided. This helps bridge the essential gap that exists within the literature and helps advance the development of next-generation sustainable refrigeration technologies.

The choice of refrigerant is also critical towards efficiency and to a greater extent in terms of its environmental impact. Though widely used, the use of HFC Refrigerant R134a calls for alternatives due to its high GWP. Promising alternatives are R290, also known as Propane, and R600a known as Isobutene. These alternatives have low ODP and GWP values in addition to favourable properties in thermodynamics. Alternative development in this line provides paths towards improving the sustainability as well as efficiency of refrigeration systems with less environmental harm. Other factors that the research identified are replacement alternatives necessitated due to its GWP being higher which raises concern about environmental degradation, particularly as an alternative to HCFC and HFCs, respectively [5]. For example, exergy analysis was conducted on eighteen refrigerants in the form of an assessment incorporating elements of environmental safety from multistage multi-evaporator vapour compression refrigeration systems that were modelled on Simulink. Results have shown that the exergy loss has been mainly contributed to by compressors, followed by condensers, evaporators, and expansion devices. Environmentally friendly refrigerants, such as R 1234ze (E) and R1234yf, have performed quite well compared to conventional options [6]. The VCRS has mostly been preferred in lots of applications because of its excellent COP and Refrigeration Effectiveness. Despite the favourable thermodynamic properties of HFCs like R134a, their significant GWP has propelled the search for eco-friendly

alternatives that offer augmented COP, effectual heat transfer, with lower values of GWP and ODP. Investigation suggests that mixed refrigeration systems or blends incorporating Nanofluids and R134a can enhance thermal efficiency and overall system performance.

An exergy analysis under AHRI standard conditions compared refrigerants like R12, R134a, R1234yf, and R600a, finding R600a to have superior performance and the lowest TEWI, while CO_2 emerged as a promising alternative due to its affordability and non-toxicity [7]. For very low-temperature applications below -50°C , (Mota-Babiloni et al., 2023) The study explored pure fluid combinations and refrigerants like R-469A for temperatures below -80°C , suggesting a trade-off between environmental concerns and flammability. Five new refrigerants (R440A, R451A, R445A, R444A, and R441A) with GWP below 150 were also investigated, with R440A as well as R451A showing superior thermal characteristics to COP and like R134a, making them eco-friendly substitutes despite flammability concerns [8]. Research Confirms that using environmentally friendly refrigerants like propane and isobutane blends can significantly enhance the system's performance while dropping environmental effects [9].

The European gas regulations have necessitated the replacement of refrigerants with higher GWP, particularly affecting standalone systems using R134a. Alternatives studied include R600a, R290, as well as CO_2 , with commercial beverage cooler tests showing significant energy reductions with these refrigerants, except for R1234yf [10]. Global warming concerns drive the necessity to decrease GHG (greenhouse gas emissions) as well as fossil fuel use, making low GWP refrigerants crucial, especially in heat pumps for waste heat recovery. However, there is a lack of performance analysis and application guidelines for these refrigerants in heat pumps, prompting a study of 17 refrigerants with lower GWP. Moreover, guidelines for utilizing refrigerants in heat pumps are provided [11]. Additionally, experimental investigations on household refrigerators using HC mixtures (R290/R600a) and R436A using different capillary tube lengths and refrigerant charges showed that R436A could achieve design temperatures of -15°C , making it a viable R134a substitute [12]. Studies mandated by the Kyoto Protocol also confirmed that R290 and R600a mixtures offer low energy consumption and higher COP, presenting them as promising alternatives to R134a [13].

Development of an Energy-Efficient Design for a Low-Volume VCRS Model Using R290, R600a, and R1234yf. Environmental and thermos-economic analyses based on COP and exergy efficiency parameters reveal that R290 is the most viable substitute for R134a, showing that its operational costs represent almost 75% of the total costs with the Kyoto Protocol requirement to phase out R134a, a 28:72 weight ratio of R134a and LPG is being considered, showing that it performs better with a higher COP and lower compressor discharge temperatures than R134a alone [14]. Compatibility with mineral oil lubricant is also satisfactory.

Besides, Performance study of various refrigerant mixtures in cascade refrigeration cycles revealed that the R134a/R170 has maximum COP and lowest mass flow rate. Further, the performance increased at a higher temperature of the evaporator with the pair of [15, 16].

The exergy analysis of the VCRS was performed to look for losses at different levels, as well as specifically in refrigerants R134a, R12, and R22, and their impact on evaporator temperature. The use of RefProp software was carried out to compare some parameters, like energetic efficiency and irreversibility, to predict the effects of more elevated evaporator temperatures on exergy loss and second-law efficiency, with the latter showing that R134a experienced the highest exergy loss [17]. The introduction of Al₂O₃ nanoparticles in VCRS without any retrofitting has been shown to enhance thermal performance with less energy consumption due to the high thermal conductivity and low viscosity of nanoparticles. Another requirement suggested in this study is the choice of more efficient materials for the compressor during the inclusion of nano-refrigerants [18]. Further investigation of R449A as a coolant in commercial refrigeration systems through EES analysis and simulation found that its addition had a significant effect on raising the performance of the system. Also, an indication of a reduction in greenhouse gas emissions up to 40.4% was shown by R290 [19]. The application of nano refrigerants and nano lubricants increased the heat transfer efficiency and COP and lowered energy consumption; however, more effort should be dedicated to their application [20].

Various approaches, including experimental and analytical methods, have been explored by researchers to provide insights into the selection of expansion devices for different refrigerants. Additionally, Enhancing COP in VCRS systems has been studied by incorporating SiO₂ nanoparticles mixed with POE oil nano lubricant and modifying condenser designs, such as using spiral condensers with increased surface area compared to conventional

designs [21]. Studies have demonstrated that increasing nanoparticle concentration improves COP. Furthermore, assessing the performance of two-evaporator VCRS systems with different arrangements (series and parallel) has shown that series arrangements yield higher COP ranges due to optimal cooling achieved with refrigerant R134a [22]. Modifications in evaporator design through fin installations have also been explored, comparing the use of Hydrocarbon (HC) refrigerants with R134a. Integration of HC-R134a has been found advantageous, reducing time to reach the similar refrigeration effect, increasing COP, as well as reducing consumption of power [23]. Owing to features like low GWP and enhanced heat transfer rates.

Findings from Indicate that the optimal selection of the working fluid does affect the system's performance [24]. The blend of R134a with nanofluids enhances cold production by improving heat transfer efficiency, leading to greater cooling capacity. It also reduces compressor power consumption by lowering the compressor's workload, resulting in decreased electrical energy use and improved energy efficiency. Previous studies independently investigated the interplay of design parameter modifications and the impact of using different refrigerant mixtures on VCRS performance.

The current investigation introduces innovative VCRS redesign with spiral condensers and series evaporators, which has greatly improved the enhancement of energy efficiency and thus reduces environmental impact by using eco-friendly refrigerants. However, the existing knowledge gaps in the current literature even show how low-GWP refrigerants and nanofluids have not been explored suitably within any VCRS configurations. This research emphasizes the dual importance of thermal performance and environmental impact, advocating for the implementation of a series arrangement of evaporators and spiral condenser configurations to enhance system efficiency. Such configurations hold promises for sustainable refrigeration applications, offering

Table 1. Based on the Selection of Eco-Friendly Refrigerants

Study	Refrigerants Examined	Methodology	Key Findings	COP Improvement	Environmental Impact
[25]	R152a, R1234yf, R134a	Experimental, oil-free VCR system	R1234yf is like R134a in temperature/pressure but lower COP as well as cooling capacity by 16% and 11% correspondingly.	Decrease in COP for R1234yf by 16%	R152a is promising but flammable
[26]	R290/R600a mixtures	Experimental on variable speed hermetic compressors	COP increased by 10-20% over R600a depending on mixture composition and conditions.	10-20% greater than R600a	Enhanced performance with proper mix ratios
[27]	R32, R1234yf, R1234ze(E), R290 vs. R134a	Dynamic model using Modelica	COP changes: +8% (R32), -12% (R1234yf), -3% (R1234ze(E)), -5% (R290) compared to R134a.	COP increases with R32, decreases with others	Significant system charge reduction for R32, R1234yf, R1234ze(E), R290

Table 2. Based on the Effect of Mixing Nanofluids with Primary Refrigerant

Study	Nanoparticle Type	Working Fluid	Methodology	Key Findings	Performance Improvement
[28]	Al_2O_3	R134a	Thermal analysis	Improved heat transfer with Al_2O_3 nanoparticles in R134a refrigerant.	Enhanced heat transfer
[18]	Al_2O_3	R134a	Experimental, no system retrofit	Al_2O_3 nano refrigerant outperformed conventional mixture, achieving faster cooling and improved energy consumption.	Faster cooling, improved energy consumption
[29]	Al_2O_3	R134a, R600a	Experimental, varying nanoparticle concentration	Adding 0.1 wt% of R600a- Al_2O_3 , decreased evaporator pressure, cuts consumption of power by 28.7%, with an increase in COP by 37.2%.	Increased COP, reduced power consumption.

Table 3. Based on the Effect of Modifications in the Design of VCRS

Study	Focus	Methodology	Key Findings
[21]	Expansion devices performance	Analysis	The straight capillary tube demonstrated a greater mass flow rate, while the Thermostatic expansion valve improved flow control.
[30]	Transient mathematical models for VCRS	Simulation, experimental validation	Mathematical models can improve domestic refrigeration system design by manufacturers.

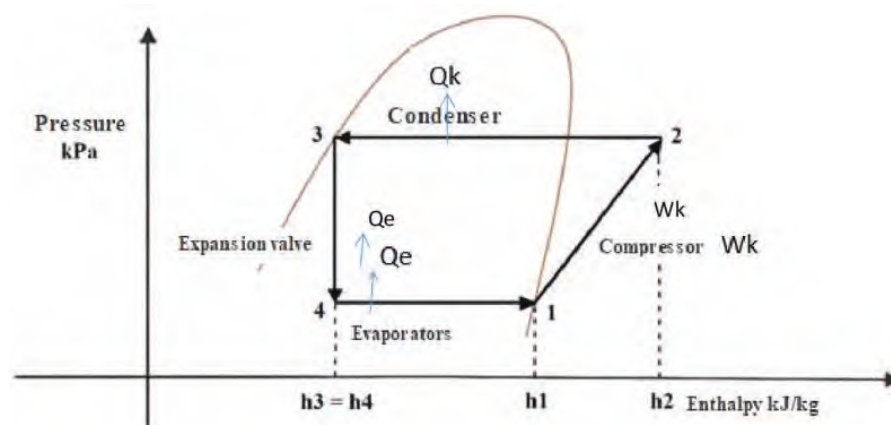
both feasibility and adaptability in addressing the pressing challenges of energy conservation and environmental sustainability. This work demonstrates a remarkable 77% enhancement in COP and a 19% reduction in power consumption relative to traditional systems for the proposed configuration. The results set up a new benchmark for the design of VCRSs and opened doors to further research concerning sustainable refrigeration technologies.

MATERIALS AND METHODS

In the section on materials and methods, the study focuses on the specification of components, the

experimental parameter selection, along with the setup of the experimental apparatus.

Figure 1 demonstrates the thermodynamic cycle of the vapor compression refrigeration system. The $P-h$ relation for the given cycle is shown. At point 1, the refrigerant leaves the evaporators in the form of low-pressure vapour and goes into the Compressor where it gets compressed to the high-pressure stage (point 2). Increasing the temperature as well as the pressure of the refrigerant (work input, W_k). The high-pressure vapour proceeds into the Condenser. In this component, at point 3, it surrenders the heat Q_k and condenses to a high-pressure liquid. Such condensation lowers the temperature but keeps on keeping the pressure at

**Figure 1.** Thermodynamic Cycle.

hand. The refrigerant which has condensed flows through the expansion valve where, from point 3, it drops to point 4, decreasing in both pressure and temperature, thus making a low-pressure liquid-vapor mixture. It is at point 4 that the refrigerant passes into the Evaporators, wherein it absorbs heat Q_e from the surroundings and evaporates completely to revert into a gaseous condition. It is at this step that the system derives its needed heat from the surroundings, hence affording the cooling needed to operate the system. This cycle repeats as the refrigerant reintroduces itself into the compressor. The energy exchange processes that take place at every stage of the cycle are manifest in the diagram, and the figure well represents the thermodynamic transformations important for efficient cooling in the modified VCRS configuration. The Coefficient of Performance (COP) evaluates the system's performance for vapor compression refrigeration as:

$$COP = \frac{\text{Refrigerating Effect}}{\text{Work Done}} = \frac{h_1 - h_4}{h_2 - h_1} \quad (1)$$

Compressor Selection and Mathematical Modelling

The compressor's scientific model incorporates measurements of thermodynamic processes which refers to the quantification of key parameters such as pressure, temperature, enthalpy, and mass flow rate within the compressor's operation. These measurements are crucial for accurately modelling work done during the compression cycle and energy exchanges that occur around this component. The compressor's fundamental equation includes W_{c0} (in kJ/kg) (Work required for a compressor), which is power consumption throughout the compression phase.

Work required for a compressor,

$$W_{c0} = h_2 - h_1 \quad (2)$$

Where it denotes enthalpy h_1 at the compressor intake and enthalpy h_2 at the compressor exit. These equations give a framework for modelling compressors considering numerous aspects such as refrigerant type, operating circumstances, etc. The hermetically sealed compressor employed in this study project was an Emerson 0.32 HP

(Model No. KCN415LAG-BXX) compressor with a voltage of 230V/50HZ.

Condenser Selection and Mathematical Modelling

Heat rejection and energy exchange with this component are included in the mathematical modelling of the condenser. Q_c , or the heat forbidden by the condenser during the condensation process, is the fundamental equation for a condenser.

Heat rejected Q_c (kW) in condenser,

$$Q_c = m(h_2 - h_3) \quad (3)$$

The enthalpy at the condenser intake (h_2) is identical to the refrigerant's mass flow rate (m), whereas at the condenser exit (h_3). The condenser's efficacy (ϵ) is determined as the ratio of actual to maximal heat transfer. These equations serve as a foundation for modelling condensers using various factors such as refrigerant type and mass flow rate, as well as providing insight for analysis and optimization of the system. Table 4 provides the characteristics of the condensers used for the study setting.

Expansion Device Selection and Mathematical Modelling

Measuring variations in enthalpy and pressure during expansion is part of the mathematical modelling of expansion devices. The isenthalpic expansion mechanism is crucial to the fundamental equation for expansion.

$$h_3 = h_4 \quad (4)$$

The expansion device's intake enthalpy is h_3 , whereas the exit enthalpy is h_4 . This equation may be used to calculate the performance of an expansion device, taking into account elements such as efficiency, operating conditions, as well as refrigerant. In the study, an expansion device selected is a capillary tube employed in the system. Capillary tubes have tiny internal diameters and are made with copper for varied lengths depending on the purpose. The capillary tube utilized in this configuration has a diameter of 2 mm.

Evaporator Selection and Mathematical Modelling

Evaporator's mathematical modelling comprises an examination of the heat absorption mechanism along with

Table 4. Choice of parameters for Air Cooled condenser

Parameters	Fins	Spiral Condenser
Material	MS (Mild Steel)	Copper
Length	9 m	7m
Diameter	0.005 m	0.007 m
No. of Turns	10	5
Thermal Conductivity	50 W/m·K	385 W/m·K
Corrosion Resistance	Lower, prone to rust	Higher, resistant to corrosion
Heat Transfer Efficiency	Lower efficiency	Higher efficiency

energy interactions around it. As per the first rule of thermodynamics, the following fundamental equation governs the behaviour of the evaporator,

Absorbed heat Q_e (kW) in the evaporator,

$$Q_e = \dot{m} (h_1 - h_4) \quad (5)$$

Where h_1 refers to the enthalpy at the evaporator outlet (compressor intake) and h_4 refers to the enthalpy at the evaporator inlet (expansion valve outlet), and $\dot{m}$ is the refrigerant's mass flow rate. Moreover, the efficacy (ϵ_{evap}) of the evaporator displays how much heat is transferred compared to how much is transferred at its highest. In the modelling of the evaporator's performance, the above equations are essential, since they consider crucial factors like refrigerant properties, heat transfer efficiency, and mass flow rate. The wood evaporator used for this experiment has an internal coiling built with a 7 mm tube diameter. Inside insulation prevents heat loss to the surroundings. A series configuration of evaporators further ensures identical cooling capacity across both.

Assortment of Additional Components

Suction pressure is measured using WIKA pressure gauges that have an overall range of 150 psi to -30 psi, and discharge pressure is measured using pressure gauges with an overall range of 300 psi and 0 psi. It employed two 500 W heaters in addition to a glass tube rotameter device for measuring the flow rate of fluid or gas. The rotameter with a capacity of 0.50 LPH to 50 LPH is used in this configuration. To ensure operational safety, an excess load cut-off mechanism was supplied to the compressor. At different locations, RTD sensors were employed for temperature measurement. A blend of R290 propane-R600a isobutane refrigerant was utilized. The desirable characteristics like High critical temperature with Low Freezing / Boiling point are important for any refrigerant. The boiling point and critical temperature of R134a are -26°C and 101°C, respectively whereas these values are -11.7 °C and 134.98°C for Isobutane R600 and -42.1°C and 96.7°C for Propane. These values indicate that isobutane and propane show critical temperature features with desirable Boiling points. The molecular weights of R134a R290 and R600a are 102.03 g/mol, 44.1 g/mol, and 58.12 g/mol respectively. Reciprocating compressors benefit from lower molecular weights because it reduce compressor size. Due to the lower molecular weight, Propane and Isobutane are more ideal than R134a. According to global safety standards (IEC/EN 60335-2-24), the limit of maximum refrigerant charge for household appliances is 150 gm which resembles about 360 g of typical refrigerants.

For experimentation, four cases were recognized and attained with the aid of valves as below.

- Conventional VCRS (Expansion Device, Evaporator I, Compressor, as well as Condenser)
- Traditional VCRS with evaporators in series (Evaporator I, Compressor, Evaporator II, Expansion Device and Spiral Condenser)

- Expansion Device, Compressor, Evaporator I, and Spiral Condenser comprise the VCRS utilizing a spiral condenser.
- Utilizing a spiral condenser, VCRS is set up in series with the spiral condenser, compressor, evaporator I, expansion device, and evaporator II. The configurations were made with the aid of valves.

This section gives a summary of the experimental methods, selection of materials, fabrication, and testing methods used in this research work. In this section, we outline the specifications of the components chosen for the experimental investigation of the VCRS system. The setup includes a condenser, evaporator, capillary tube, compressor, pressure gauges, refrigerant, and rotameter. The design of the VCRS system is done by considering some standard parameters. From the literature review following parameters were selected for experimentation purposes. Following are short description of the Specifications of components of the Vapor Compression Refrigeration System.

Condenser

Evaporator

In this experimental setup, the evaporator is constructed from wood, with a copper tube coiled inside having a diameter of 7mm. Insulation has been incorporated within the evaporator to shield it from ambient heat sources. The fact that all evaporators have the same cooling capacity is notable.

Compressor

A compressor is a mechanical device that boosts gas pressure by decreasing volume, facilitating refrigerant circulation. The setup employed a 0.32 hp hermetically sealed (230V/50Hz) compressor (Emerson model KCN415LAG - BXX).

Table 5. Specifications of Fin-type condenser

Parameter	Specifications
Coil Diameter	5 mm
Coil Length	9 mm
Coil Material	Mild Steel
Number of turns	10

Table 6. Specifications of Spiral type condenser

Parameter	Specifications
Coil Diameter	7 mm
Coil Length	7 mm
Coil Material	Copper
Number of turns	5

Capillary Tube

The capillary tube, a key throttling device in refrigeration, comprises a copper tube with a small internal diameter. It's lengthy and coiled to save space, with a 2mm diameter in this setup.

Pressure Gauges

Pressure gauges are vital for measuring fluid pressure in diverse applications. In this study, gauges with a range of 150 psi to -30 psi monitor suction pressure from the compressor, while track discharge pressure ranging from 300 psi to 0 psi.

Refrigerant

Refrigerants facilitate heat transfer in heat cycles. This study employs R134a (Tetra Fluoro ethane) due to its widespread use, well-documented thermodynamic properties, and relevance as an industry standard, allowing for effective comparison with alternative refrigerant blends in the study with specific properties such as the critical pressure and temperature, as well as boiling and freezing point.

Rotameter

In this experiment, a glass tube calibrated for 0.50 to 50 Lph serves as the rotameter, which measures fluid or gas flow rates.

Nanofluid Preparation

Al_2O_3 nanoparticles were used for this research work and specifications were taken from the research work. Nanofluid preparation is an important step for experimental study. Nanofluid can be prepared by single-step method. The initial stage in the experimental study is to prepare Nano-oil for lubrication. Nanofluids may be produced using a one-step technique. In this method nanopowder of Al_2O_3 is taken and

mixed with POE oil in the required proportion and after that kept in an ultrasonic vibrator for 7-8 hours.

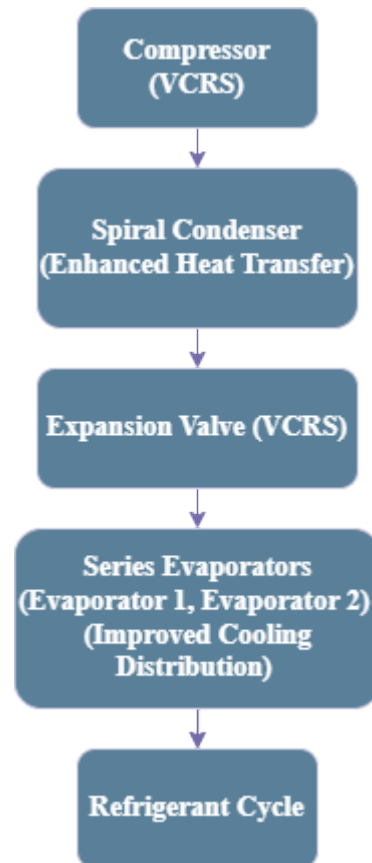


Figure 3. Modified VCRS Configuration (with Spiral Condenser and Series Evaporators)

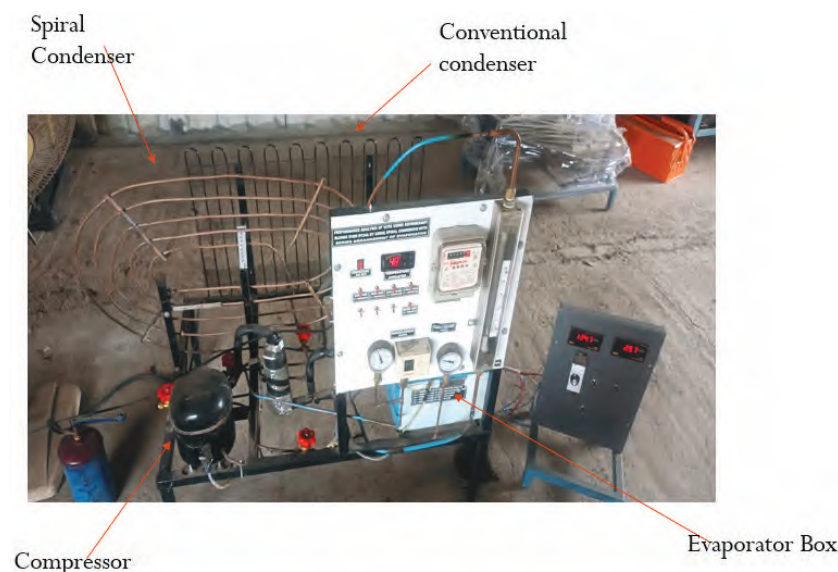


Figure 2. Experimental Set-Up.

As indicated in Figure 2, four cases are considered with distinct refrigerant blends in each case. A pure refrigerant R134a was used in Case 1 as a baseline. In Case 2, the nanofluid Al_2O_3 is blended with R134a to measure the impact of the nanofluid on the system performance and heat transfer effectiveness. The third and fourth cases presented a Propane-Isobutane mixture in diverse proportions for potential use as an alternative refrigerant. For all the cases, a set of experiments was run, with the results reported in tabular form, showing what was obtained in terms of performance metrics such as cooling capacity, COP, and EER.

As shown in Figure 3 the configuration and order of the redesigned Vapor Compression Refrigeration System, with high-capacity components that will enhance performance. A Compressor leads off where it compresses the refrigerant and begins the cycle. Then it passes through a Spiral Condenser. This is among the major improvements suggested to maximize the heat transfer efficiency compared to traditional condensers. This condenser absorbs the heat inside so that the refrigerant cools down and, therefore, proceeds to the next stage. The cooled refrigerant then passes over into the Expansion Valve from the condenser, where, due to the dropping pressure, it becomes cold. The refrigerant remains in this lower pressure and cooler state for subsequent steps in the process to work properly and

efficiently in cooling. Then, there's a series of Evaporators: Evaporator 1 and Evaporator 2, following one another. This arrangement offers better cooling distribution across the area and optimal energy usage. This is because the respective evaporators draw heat out of the space, which in turn lowers the temperature of the target area. The cycle ends with the refrigerant returning to the compressor, thus maintaining the continuous flow of the system with maximum efficiency. A revised design that is now the spiral condenser and series evaporators thus tries to cool the system in an attempt to enhance the efficiency of the overall system.

RESULTS AND DISCUSSION

This section details the validation of experimental results by comparing them with simulations from Cool Pack Software, focusing on various refrigerant blends and parameters across different cases. In this economic investigation, the improved VCRS system is also discussed with the Payback Period. Using Cool Pack software, the results validation is performed indicating decent promise with below 10% variance. The COP of the normal VCRS is 2.25, as shown in Table 7, which also presents the percentage improvements in COP compared to normal VCRS for different arrangements.

Table 7. COP Improvement for Different Arrangements

Arrangement	COP	Percentage Improvement Compared to Normal VCRS
Spiral VCRS	3.08	$(3.08-2.25)/2.25= 0.36$ (36%)
Normal with Series	3.52	$(3.52-2.25)/2.25= 0.56$ (56%)
Spiral with Series	4	$(4-2.25)/2.25= 0.77$ (77%)
Evaporators in series and spiral condenser using R134a together with nanofluid	5.33	$(5.33-2.25)/2.25= 1.37$ (137%)

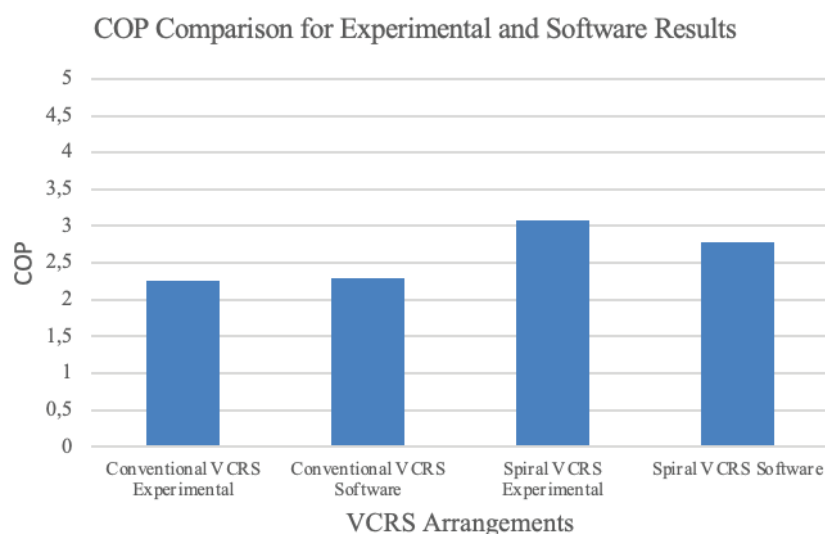


Figure 4. COP Comparison for Experimental and Software Results.

The COP result of both conventional and spiral VCRS configurations is plotted on a graph against experimental and software-simulated values. A slightly higher COP, 2.4, is given by the simulation result as compared to the value of 2.2 yielded by the experimental result for conventional VCRS. In the spiral configuration, experimental COP is enhanced considerably to a level of about 3.4, while in the simulation result, it is somewhat lower at 3.0. It was noted that the VCRS spiral configuration yielded a tremendous increase in efficiency over the conventional type configuration. The matching of the experimental and software results confirms the improvement of the spiral condenser and series evaporator design on system performance, and this was particularly obvious for the spiral configuration. A comparison of different refrigerants based on environmental parameters is carried out. According to a United Nations Environment Program study with a decrease of 1% ozone layer, there will be an increase in 2% chances of skin cancer. Intergovernmental Panel on Climate Change (IPCC) has reported biosphere has already warmed up to 10c and in the next century there will be a rise of about 4°C due to human interference in the global climate.

As shown in Figure 4 the Comparison of Results for Change of system design including Spiral condenser as well as In-Series Evaporators using refrigerant R134a, Al203 Nanofluid along with R134a and using Propane Isobutane blend experiments were conducted and results obtained are tabulated. For reference Results and discussion for Case A] Using R134a are given below. There are various ways to enhance heat transfer and improve the coefficient of performance in VCRS. From past research, it has been found that with the use of a spiral condenser and series evaporator arrangement, there is an enhancement in heat transfer. For this research effect of combining these two parameters is discussed.

Impact on COP

Considering practical applications of the VCRS system there are requirements where simultaneous cooling is required for more than one compartment. Also, a spiral condenser has a high heat transfer coefficient and a compact size, providing a beneficial heat exchange area that improves the coefficient of performance. There is no earlier study reported to find an effect on performance with a combination of parameters which was studied in this research and discussed. Condenser design modifications can have a significant effect on COP. As the condenser plays a crucial role in heat rejection with its optimal design substantial COP improvement can be observed. With innovations in the design of condensers, COP enhancement can be achieved. The spiral condenser has a distinct design from the conventional straight tube condenser and has various advantageous features like increased efficiency and compact design. Also, with the use of a spiral condenser which has increased surface area and because refrigerant will get a longer path to flow more heat exchange will take place. Due to this enhanced heat transfer more condensation or subcooling is achieved resulting in increased COP.

Also, from this research experimental results graph, it is visible that with system modifications, design COP is increased. There is almost a 36% enhancement in COP with the use of a spiral condenser over a conventional condenser and an almost 77% increase in COP utilizing a spiral condenser in a series arrangement. The application of a series arrangement of evaporators has multiple advantages, including efficiency enhancement and ease of varying loads. In the series arrangement, as refrigerant passes one after another, it allows correct cooling distribution. Also, the series arrangement allows the activation of one or more individual evaporators based on thermal requirements. With this arrangement, energy savings as per application temperature control are achieved. From Fig. 5, it

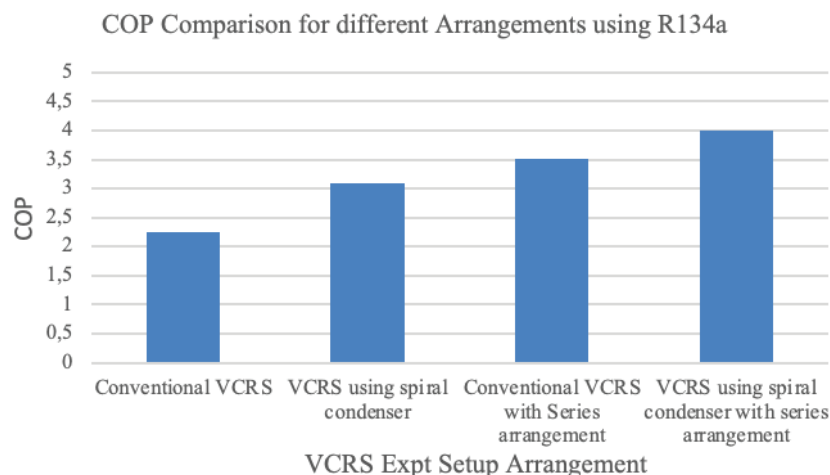


Figure 5. COP Comparison for different Arrangements using R134a.

can be seen that with the use of the series arrangement of evaporators over a conventional VCRS system with a single evaporator, COP increased up to 55%, and when the series system was incorporated along with the series arrangement, it improved by up to 77%. So, with the use of both factor combinations, COP is enhanced.

Within our study, considerable enhancement in COP is attained with spiral condenser configurations and series evaporators that amount to about 77% more than the conventional systems. Interestingly, this is greater than that envisaged by the COP improvement realized [2] When using water-cooled condensers, with a marked performance configuration. More recently, [31] demonstrated that the incorporation of nanofluids like CuO, ZnO, and Al_2O_3 in the R22 refrigerant has enhanced the thermal performance of the air conditioning systems highly. The novel application is then extended into this study by incorporating nanofluids, in the shape of Al_2O_3 . This results in a COP improvement of 137% compared with a performance benchmark that was previously indicated as substantially out of bounds. Also, the reductions in power consumption experienced with our proposed design are higher than those for the work reported [3], proving the effectiveness of our system design in this regard.

Impact on Power Consumption (Wc)

Modification of the VCRS system using a spiral condenser has a noticeable impact on power consumption. As a spiral condenser has a coiled structure, the heat transfer rate is enhanced, which results in enhanced condensation and reduced temperature at the expansion valve inlet. This results in the compressor working at a low-pressure head and using less power to maintain the required cooling effect. This ultimately results in enhanced performance

and an energy-efficient system. In industrial applications where energy conservation is of prime importance, spiral condensers can be used. From the graph, results reveal that with the using a spiral condenser over a conventional condenser, power consumption is reduced by 19%, whereas, with the use of a spiral condenser over a conventional system along with a series arrangement, there is a decline in consumption of 15%.

Impact on Net Refrigeration Effect (RE)

A spiral condenser allows improved heat dissipation, resulting in the low-pressure temperature of refrigerant before entering the expansion valve. Hence refrigerant-entreating evaporators already have low-temperature benefits in absorbing more heat from refrigerated compartments/evaporators. Thus, with the implementation of a spiral condenser, the refrigeration effect is enhanced. Also, the series arrangement optimizes the heat transfer process, allowing for greater cooling efficiency and the ability to maintain lower temperatures within the system. Figure 7, illustrates that including a spiral condenser, there is a 17% rise in the net refrigeration effect, while a spiral condenser used in combination with a series arrangement, is almost doubled.

Economic analysis of the modified VCRS system (payback period).

The time taken for all cash inflows identical to the early investment is known as the payback period. It shows when the experiment has recouped its early expenses. The payback period is the early investment or yearly inflow of net Cash. The outcome reflects how many years are needed to break even on the investment.

As shown in Table 9 and Table 10, modification cost refers to the expenses incurred when making changes or

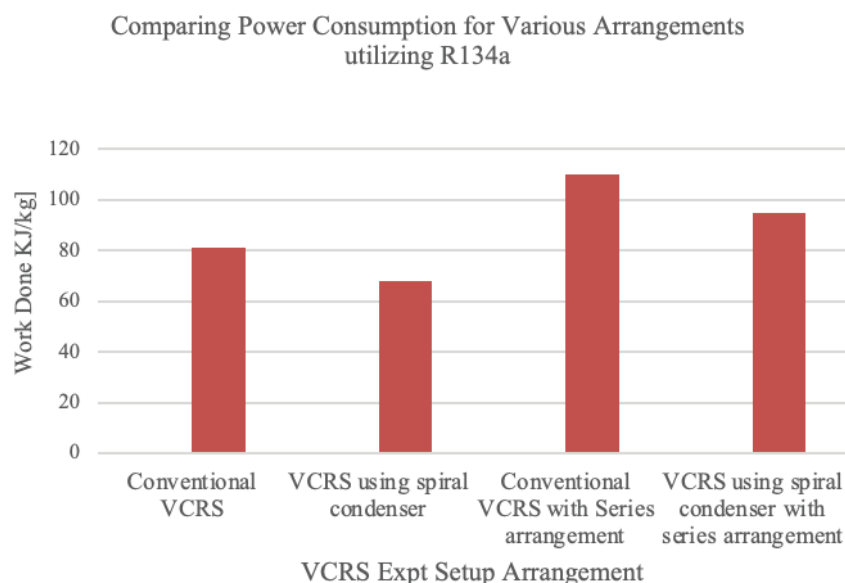


Figure 6. Impact on power consumption using evaporators in series including a spiral condenser and R134a.

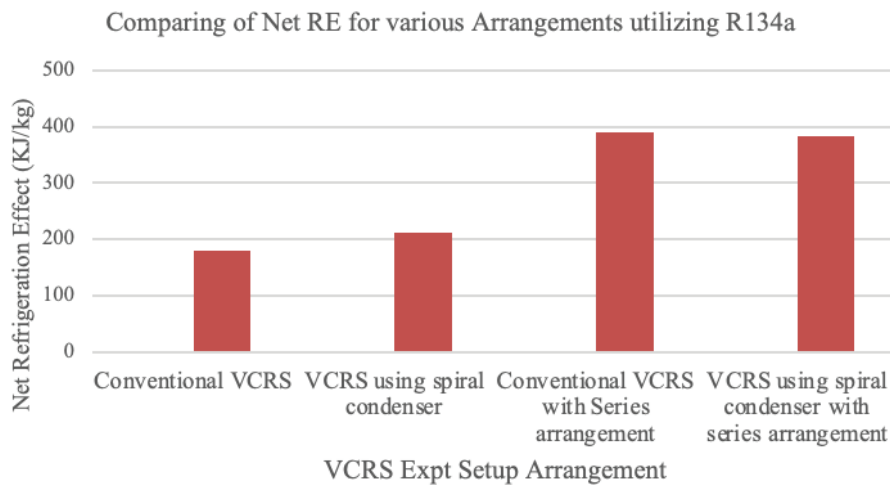


Figure 7. Comparing Net Refrigeration Impact with various arrangements utilizing R134a.

upgrades to an existing system (e.g., the modification cost increase of spiral condenser over conventional fins type is 1000 Rs.), and 0 means no change in cost and 2200 means increase in cost with the use of nanofluid. It is in INR.

Future Scope of the Work

Future research could emphasize growing the study to embrace a comprehensive range of refrigerants and nanofluids to further understand their impact on VCRS performance. Investigations could explore alternative low-GWP refrigerants and advanced nanofluids to enhance energy efficiency while minimizing environmental impact. Additionally, incorporating varying operating conditions

and load profiles could provide a more comprehensive assessment of system performance across different scenarios. This would help identify optimal configurations and materials for specific applications, enhancing the adaptability and sustainability of VCRS technology.

Integrating real-time monitoring and control systems using IoT and machine learning could optimize VCRS performance dynamically. Future studies could investigate the implementation of smart controls and predictive maintenance strategies to improve efficiency and reduce operational costs. By leveraging advanced data analytics and real-time feedback, it may be possible to fine-tune system operations and extend the lifespan of components.

Table 8. Cost Saving for CASE I and CASE II

Parameter	Cost savings for CASE I System design modified with the usage of in-series evaporators as well as spiral condensers utilizing R134a	Cost savings for CASE II System design modified with the usage of the spiral condenser as well as in-series evaporators utilizing R134a and Al_2O_3 nanofluid	Better Results
Initial Investment	Here an initial Investment will be an extra 6000 Rs.	In this, the initial Investment will be an extra 8200 Rs.	Case I Less Initial Investment
Operating Cost Saving	Cost Savings (%) = $77\% \times 40\% = 30.8\%$ Due to the efficiency enhancement. This indicates a reduction of 30.8% expected in whole operating costs.	Cost Savings (%) = $137\% \times 40\% = 54.8\%$ Due to the efficiency enhancement, this indicates a reduction of 30.8% expected in whole operating costs.	Case 2 More Cost Saving
Energy Consumption Cost	The compressor performs 12 hours daily. Every day, consumption of energy = 3.9 kWh. Monthly consumption = 117 kWh. Yearly cost = 13534 Rs. Modification leads to a 30.8% cost reduction. Yearly savings = 4060 Rs.	The compressor runs 12 hours/day. Daily energy consumption = 3.9 kWh. Monthly consumption = 117 kWh. Monthly cost = 1128 Rs. Yearly cost = 13534 Rs. Modification results in a 54.8% cost reduction. Yearly savings = 7417 Rs.	Case 2 More Yearly Savings.
Payback period	Initial Investment/Annual Net Cash Inflow or saving $6000/4060 = 1.47 \sim 1.5$ Year	Initial Investment/Annual Net Cash Inflow or saving $8200/7417 = 1.1$ Year	Case 2 Less Payback Period.

Table 9. Cost of Components for CASE I

Sr. No.	Components	Conventional	Modified	Rise in price
1	Refrigerant	R134a	R134a	0
2	Condenser	Fins type	Spiral	1000
3	Air Heater	I	I and II	1000
4	Evaporator	1	I and II	4000

Table 10. Cost of Components for CASE II

Sr. No.	Component	Conventional	Modified	Increase in Modification Cost
1	Refrigerant	R134a	R134a+Nanofluid	2200
2	Condenser	Fins type	Spiral	1000
3	Air Heater	I	I and II	1000
4	Evaporator	1	I and II	4000

Table 11. Comparison of Refrigerant Blends

R290+R600a	R134a+Al ₂ O ₃
The blend consists of propane (R290) and isobutane (R600a), both of which are known for their high energy efficiency and efficient heat transfer.	R134a, a stable refrigerant, can enhance heat transfer and system efficiency when combined with Al ₂ O ₃ nanoparticles.
The blend's low GWP and ODP make it ecologically benign, in line with worldwide measures to alleviate climate change besides lessening emissions of greenhouse gas.	R134a, with a high GWP, can be enhanced with Al ₂ O ₃ nanoparticles to improve efficiency, reduce environmental impact, and potentially lower refrigerant charge and leakage.
Propane (R290) and isobutane (R600a) are flammable refrigerants.	R134a, a non-flammable refrigerant with a good safety record, may be safely blended with Al ₂ O ₃ nanoparticles if they are correctly disseminated throughout the refrigerant.
Due to its advantageous thermodynamic features, the R290 and R600a blend often provide great energy efficiency and excellent heat transfer characteristics, allowing for effective VCRS operation.	The addition of Al ₂ O ₃ nanoparticles in R134a is aimed at enhancing its heat transfer capabilities, potentially increasing system efficiency, but its overall efficiency depends on various factors.

Exploring these avenues would contribute to more resilient and efficient refrigeration systems, addressing both economic and environmental challenges in the industry.

CONCLUSION

The study proposed a design for environment-friendly refrigerants, comparing performance with conventional system valve arrangement. The conventional system valve arrangement in a Vapor Compression Refrigeration System (VCRS) typically refers to the standard layout and types of valves used to control the pressure and flow of the refrigerant throughout the system. An arrangement of spiral condensers was employed in addition to in-series evaporators. Outcomes of system validation using the cool pack's

software of the base case highlight performance in the allowable (10%) limit.

The major conclusions derived from the study project are as follows.

- The use of spiral condensers and series evaporators boosts the COP (Coefficient of Performance) of VCRS (Vapor Compression Refrigeration System) systems largely. The structure of the spiral condenser lifts the COP by 36% whilst the arrangement of the series evaporator lifts the Coefficient of Performance (COP) by 55%. When both enhancements are combined, they yield an excellent 77% improvement in the Coefficient of Performance (COP).
- Spiral condenser saves 19% of power as compared to conventional condenser existing in the system. If it is taken together with a series arrangement, power

consumption is reduced by 15%. This only suggests a significant betterment of energy efficiency with the new format of the system.

- In residential freezers, one possible replacement for R134a would be a hydrocarbon blend of propane and isobutane.
- Dispersed Al_2O_3 or ZnO outperformed standard working fluid mixtures.
- Redesign of the system including spiral condenser with in-series evaporators using R134a enhanced COP by 77% over conventional VCRS along with a 1.5-year pay-back period.
- Redesign of the system including spiral condenser with in-series evaporators using R134a along with nanofluid R134a enhanced COP by 137% over conventional VCRS along with a 1.1-year payback period.

This study redesigned a vapour compression refrigeration system, and changes brought about by spiral condensers and series evaporators did inform one of better energy efficiencies and mitigating greenhouse gas emissions with the adoption of low-GWP refrigerants. This means in turn significant value savings and great power efficiency, which will save consumers and companies significantly and will encourage the use of green technologies. Second, and more importantly, the findings of this study can inform manufacturers to develop the following generation refrigeration systems to be flexible in the changing operation, so that it will eventually lead to sustainable commercial and residential applications. It puts at the disposal of stakeholders who are eager to take energy-efficient refrigeration solutions toward practical application the most crucial literature, thus filling the gap between innovative design and practical application in the face of global challenges on climate.

NOMENCLATURE

HFC	Hydrofluorocarbon
GWP	Global Warming Potential
ODP	Ozone Depletion Potential
HCFC	Hydrochlorofluorocarbons
COP	Coefficient of Performance
$\dot{m}$	refrigerant mass flow rate
h	Enthalpy
Q_c	Heat rejected in Condenser
W_c	Work Required for the Compressor
VCRS	Vapor Compression Refrigeration System
Q_e	Heat absorbed in Evaporator
GWP	Global Warming Potential
TEWI	Total Equivalent Warming Impact
POE	Polyol ester oil
AHRI	Air Conditioning, Heating, & Refrigeration Institute

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Wongwises S, Chimres N. Experimental study of hydrocarbon mixtures to replace HFC-134a in a domestic refrigerator. *Energy Convers Manag* 2005;46:85-100. [\[Crossref\]](#)
- [2] MM. Performance Improvement of VCRS Using Waste Heat Recovery Method. *Int J Emerg Trends Eng Res* 2020;8:1964-71. [\[Crossref\]](#)
- [3] Srithar K, Rajaseenivasan T, Arulmani M, Gnanavel R, Vivar M, Fuentes M. Energy recovery from a vapour compression refrigeration system using humidification dehumidification desalination. *Desalination* 2018;439:155-61. [\[Crossref\]](#)
- [4] Thangavel P, Somasundaram P, Sivakumar T, Kumar CS, Vetrivelvan G. Simulation analysis of compression refrigeration cycle with different refrigerants. *Int J Eng Innov Technol* 2013;2:127-31.
- [5] Malwe PD, Shaikh J, Gawali BS. Exergy assessment of a multistage multi-evaporator vapour compression refrigeration system using eighteen refrigerants. *Energy Reports* 2022;8:153-62. [\[Crossref\]](#)
- [6] Kulkarni S, Chavali S, Dikshit S. A review on analysis of Vapour Compression Refrigeration System (VCRS) for its performance using different eco-friendly refrigerants and nanofluids. *Mater Today Proc* 2023;72:878-83. [\[Crossref\]](#)
- [7] Islam MA, Mitra S, Thu K, Saha BB. Study on thermodynamic and environmental effects of vapour compression refrigeration system employing first to next-generation popular refrigerants. *Int J Refrig* 2021;131:568-80. [\[Crossref\]](#)

- [8] Bolaji BO. Theoretical assessment of new low global warming potential refrigerant mixtures as eco-friendly alternatives in domestic refrigeration systems. *Sci African* 2020;10:e00632. [\[Crossref\]](#)
- [9] Selloum A, Triki Z, Engineering YC-J of T, 2021 undefined. Thermodynamic analysis of a solar-driven vapour compression refrigeration system using R1234ze for cooling applications in the Ghardaïa region (Southern Algeria). *DergiparkOrgTrA Selloum, Z Triki, Y ChibaJournal Therm Eng* 2021•*dergiparkOrgTr* 2024;10:130-41. [\[Crossref\]](#)
- [10] Sánchez D, Andreu-Nácher A, Calleja-Anta D, Llopis R, Cabello R. Energy impact evaluation of different low-GWP alternatives to replace R134a in a beverage cooler. Experimental analysis and optimization for the pure refrigerants R152a, R1234yf, R290, R1270, R600a and R744. *Energy Convers Manag* 2022;256:115388. [\[Crossref\]](#)
- [11] Wu D, Hu B, Wang RZ. Vapor compression heat pumps with pure Low-GWP refrigerants. *Renew Sustain Energy Rev* 2021;138:110571. [\[Crossref\]](#)
- [12] Reddy DVR, Bhramara P, Govindarajulu K. Experimental Evaluation of the Effect of Refrigerant Charge and Capillary Tube Length on the Performance of Household Refrigerator with Different Configurations of R290 and R600a. *Mater Today Proc* 2018;5:11845-52. [\[Crossref\]](#)
- [13] Mohanraj M, Jayaraj S, Muraleedharan C, Chandrasekar P. Experimental investigation of R290/R600a mixture as an alternative to R134a in a domestic refrigerator. *Int J Therm Sci* 2009;48:1036-42. [\[Crossref\]](#)
- [14] de Paula CH, Duarte WM, Rocha TTM, de Oliveira RN, Mendes R de P, Maia AAT. Thermo-economic and environmental analysis of a small capacity vapour compression refrigeration system using R290, R1234yf, and R600a. *Int J Refrig* 2020;118:250-60. [\[Crossref\]](#)
- [15] Gill J, Singh J. Energy analysis of vapour compression refrigeration system using a mixture of R134a and LPG as a refrigerant. *Int J Refrig* 2017;84:287-99. [\[Crossref\]](#)
- [16] Logesh K, Baskar S, Md Azeemudeen M, Reddy BP, Jayanth GVSS. Analysis of Cascade Vapour Refrigeration System with Various Refrigerants. *Mater Today Proc* 2019;18:4659-64. [\[Crossref\]](#)
- [17] Ozair Arshad M, Azam Q, Twaha Irfan Ahmad S, Khan F, Atif Wahid M. Analysis of vapour compression refrigeration system with R-12, R-134a and R-22: An exergy approach. *Mater Today Proc* 2021;46:6748-52. [\[Crossref\]](#)
- [18] Ajayi OO, Ukasoanya DE, Ogbonnaya M, Salawu EY, Okokpuije IP, Akinlabi SA, et al. Investigation of the Effect of R134a/Al₂O₃ -Nanofluid on the Performance of a Domestic Vapour Compression Refrigeration System. *Procedia Manuf* 2019;35:112-7. [\[Crossref\]](#)
- [19] Giménez-Prades P, Navarro-Esbrí J, Udriou CM, Mota-Babiloni A. Influence of subcooling in R-449A supermarket refrigeration system and screening of refrigerant mixtures for its energetic and environmental improvement. *Appl Therm Eng* 2024;236:121787. [\[Crossref\]](#)
- [20] Said Z, Rahman SMA, Sohail MA, Bahman AM, Alim MA, Shaik S, et al. Nano-refrigerants and nano-lubricants in refrigeration: Synthesis, mechanisms, applications, and challenges. *Appl Therm Eng* 2023;233:121211. [\[Crossref\]](#)
- [21] Sivaraman N, Vaidyanathan RM, Patel M, Markos M. Performance analysis of refrigerants with various expansion valves in vapour compression refrigeration and air-conditioning system. *Mater Today Proc* 2021;45:2465-9. [\[Crossref\]](#)
- [22] Garg I, Bajpai V. To Study Various Performance Parameters of a Vapor Compression Refrigeration System with Two Evaporators using R-134(a). *J Basic Appl Eng Res* n.d.;5:268-71.
- [23] Subramanian S, Joswa Lazaras IP, Srinivasan S, Sundaresan D, Balakrishnan SK. Performance analysis of fin tube evaporator using various refrigerants. *Therm Sci* 2020;24:609-12. [\[Crossref\]](#)
- [24] Goyal A, Sherwani AF. Optimization of energetic performance and payback period of organic Rankine cycle integrated vapour compression refrigeration system based on exergy, economic, and environmental criteria. *J Therm Eng* 2023;9:648-58. [\[Crossref\]](#)
- [25] Chen X, Liang K, Li Z, Zhao Y, Xu J, Jiang H. Experimental assessment of alternative low global warming potential refrigerants for automotive air conditioners application. *Case Stud Therm Eng* 2020;22:100800. [\[Crossref\]](#)
- [26] Ozsipahi M, Kose HA, Kerpici H, Gunes H. Experimental study of R290/R600a mixtures in vapour compression refrigeration system. *Int J Refrig* 2022;133:247-58. [\[Crossref\]](#)
- [27] Dhumane R, Ling J, Aute V, Radermacher R. Performance comparison of low GWP refrigerants for a miniature vapour compression system integrated with enhanced phase change material. *Appl Therm Eng* 2021;182:116160. [\[Crossref\]](#)
- [28] Prasad US, Mishra RS, Das RK. Experimental studies of vapour compression refrigeration system with eco-Friendly primary refrigerant and brine mixed with nanoparticles as secondary refrigerant. *Mater Today Proc* 2021;45:3857-9. [\[Crossref\]](#)
- [29] Yogesh Joshi, Dinesh Zanwar, Sandeep Joshi. Performance investigation of vapour compression refrigeration system using R134a and R600a refrigerants and Al₂O₃ nanoparticle-based suspension. *Mater Today Proc* 2021;44:1511-9. [\[Crossref\]](#)
- [30] Gardenghi ÁR, Lacerda JF, Tibiriçá CB, Cabezas-Gómez L. Numerical and experimental study of the

- transient behaviour of a domestic vapour compression refrigeration system - Influence of refrigerant charge and ambient temperature. *Appl Therm Eng* 2021;190:116728. [\[Crossref\]](#)
- [31] Vandaarkuzhali S, Xavier JF, Ramadoss R, Jayaseelan V, Sudhakar K. Energy performance of R22 blended with nanofluids in the refrigeration system. *J Optoelectron Biomed Mater* 2021;13:33-44. [\[Crossref\]](#)



Research Article

Harnessing response surface methodology for diesel engine optimization using titanium dioxide-enhanced rice bran biodiesel to improve emissions and efficiency

Manoj Kumar GUPTA^{1,*}, Ashok Kumar SRIVASTAVA²

¹Research Scholar, Maharshi University of Information Technology, Lucknow, 226001, India

²Department of Mechanical Engineering, Maharshi University of Information Technology, Lucknow, 226001, India

ARTICLE INFO

Article history

Received: 26 February 2025

Revised: 29 May 2025

Accepted: 31 May 2025

Keywords:

CI Engine; Optimization;
Response Surface Methodology;
Rice Bran Biodiesel; Titanium
Dioxide Nanoparticles

ABSTRACT

The present work is associated with synergistic optimization and the effect of three operational variables, namely Load on the engine, rice bran biodiesel Blend, and TiO₂ Nanoparticle for emission attenuation of HC, CO₂, NO_x, O₂, and CO, along with BTE amelioration of diesel engine sustainability. Under the category of Empirical Research work, Central Composite Design-based Response Surface Modeling and analysis of variance were done to find out the best suitable mathematical relationship with the most and least significant operational variable for all responses at a 95% level of confidence and 5% significance factor. Experimental data were obtained using a diesel engine test rig with biodiesel blends up to 30% and TiO₂ nanoparticles up to 200 ppm, under low, medium, and high load conditions. The responses were optimized using MINITAB with contour and surface plots.

The optimum combination of three operation parameters reported as 2.64094 kW engine Load, 30 ppm rice bran Biodiesel, and 141.742 ppm TiO₂ nanoparticles. Additionally, the optimal combination included CO 0.050% of the total sample, CO₂ 3.30% of the total sample, HC 13.29 ppm, NO_x 385.78 ppm, and BTE 32.50%, resulting in a desirability effect of 73.74%; Improves BTE by 31.4% compared to pure diesel at low load and 22.9% at high load. A confirmation test run validates the result with a forecast error of less than 4%, with remarkable emissions reduction and performance gain of regular diesel engines without major changes, making it an eco-friendly option for the future.

Cite this article as: Gupta MK, Srivastava AK. Harnessing response surface methodology for diesel engine optimization using titanium dioxide-enhanced rice bran biodiesel to improve emissions and efficiency. J Ther Eng 2025;11(5):1276–1292.

INTRODUCTION

Diesel combustion in CI engines emits gases such as CO, CO₂, HC, NO_x, and SO_x, all of which contribute

significantly to atmospheric pollution. These emissions are linked to environmental issues, including global warming, ozone depletion, acid rain, and reduced visibility. Human

*Corresponding author.

*E-mail address: mail2manojgupta@gmail.com

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkılıç



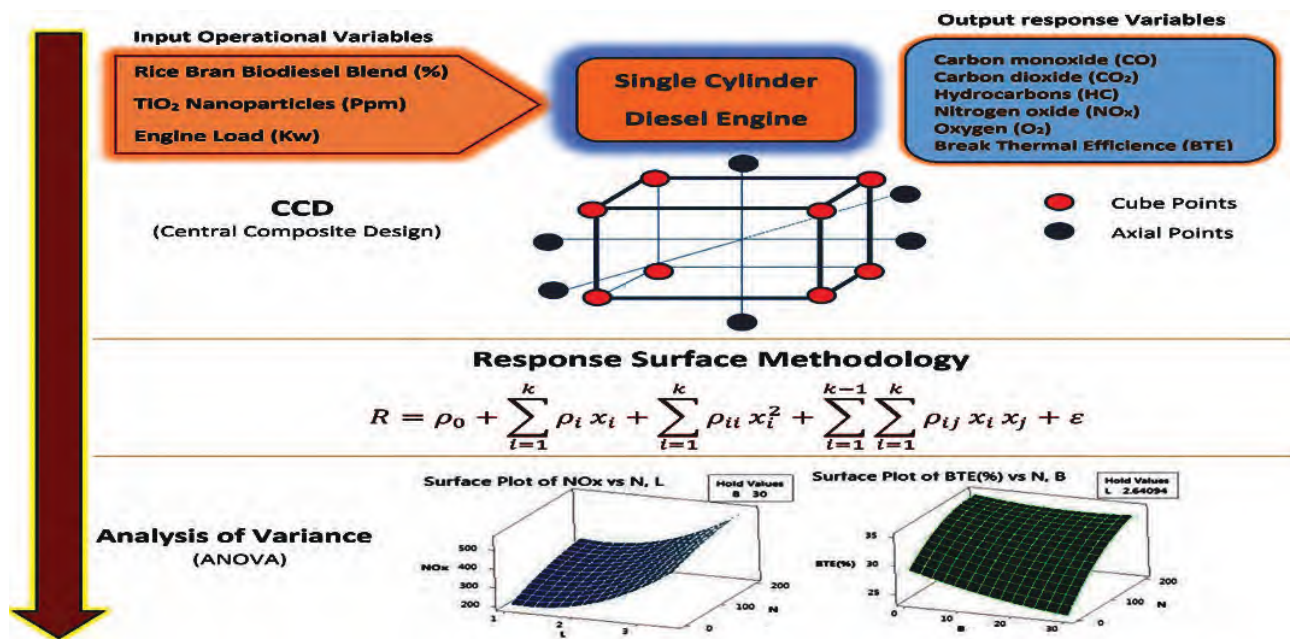


Figure 1. Graphical presentation of work.

exposure to these pollutants is associated with non-cancerous and cancerous health risks, such as respiratory issues, immune system damage, reproductive and neurological disorders, and cardiovascular diseases [1, 2, 3, 4].

As of 2021, the transportation sector accounts for 19% of global annual energy consumption. Projections by the International Energy Agency and Mobility 30 predict a 53% rise in global energy demand by 2030. According to the International Energy Outlook 2017, the energy share of on-road transportation stood at 13%, with on-road freight and passenger transport consuming 25.3 quadrillion British thermal units [5, 6, 7].

From now until 2030, oil use in road transport will increase by roughly 6 million barrels per day, with a particularly sharp increase in 2021, and it will climb by close to 8 million barrels per day in aviation, shipping, along petrochemicals [5]. Transport is a major source of pollution and climate change in the environment. Transport emissions, particularly from automobiles, contribute significantly to the levels of carbon dioxide in the atmosphere. With a range of 22 percent, transportation is the second-largest source of worldwide CO₂ emissions [8].

As a result of climate change, global warming is 0.8 °C above the level of pre-industrial, leading to a rise of sea level around 3.2 cm per decade. A rise of sea level of 0.5 to 1 meter is estimated before the 2060s by an increase of 4 °C global temperature. Although the greenhouse effect plays an impactful factor in global warming, naturally, though emission of transportation means has a major contribution. By 2100, the IPCC predicts a rise in global temperature of 1.1 to 6.4 degrees Celsius and a rise in global sea level of 7 to 23 inches, according to the Fourth Assessment Report.

Global greenhouse emissions must be cut by 50–85 percent below 2000 levels by 2050, as per the IPCC, to keep warming to 2–2.4 degrees Celsius. To meet this goal, greenhouse emissions from all sectors must be decreased in a multi-generational effort [9, 10].

Therefore, several researchers are putting their effort into overcoming this problem by using alternative fuel to existing diesel engines without measuring the modification of the engine design. Biodiesel is one of the best choices of researchers as an alternative fuel due to its renewability, biodegradability, sulfur-free, non-toxic, and can greatly lower exhaust emissions [11,12].

In 2016, Vijayakumar Chandrasekaran et al. found that adding 50 ppm CuO nanoparticles to a 20% mahua biodiesel blend improved BTE by 2.19% and reduced HC, CO, and smoke emissions. However, NO_x increased by 3.2%. The blend was considered a viable alternative for CI engines [13]. Mr. S. Padmanaba Sunder and his team found that using 200 ppm TiO₂ with plastic oil improved BTE by 2.64% and reduced HC by 18%. Additionally, 150 ppm TiO₂ decreased smoke opacity by up to 38% [14].

Samson K. Fasogbon in 2023 concluded that B20 biodiesel with 10 g/L Al₂O₃ nano-additive offers optimal CI engine performance and emissions [15]. Dhananjay Khankal in 2024 found that B10 mango kernel biodiesel improves BTE and BSFC over diesel. However, it shows higher CO₂ and NO_x emissions at low injection pressure. At 25% load and high injection pressure, CO₂ and NO_x emissions decrease by 12.5% and 13.23%, respectively, compared to diesel [16].

K. Yamini in 2024 found that isobutanol-based biodiesel) With 10% diethyl ether significantly reduced at full

load compared to diesel: HC by 24.39%, CO by 4.6%, NO_x by 9.33%, and smoke opacity by 8.84% [17].

Researchers reported that biodiesel is comparable to conventional diesel with minimal impact on performance and emissions reductions. Additionally, it may be blended with conventional diesel or utilized as biodiesel oil in diesel engines. The use of biodiesel as an environmentally friendly fuel is growing in popularity [18, 19].

The literature review helps to find the answer: why my research work is important. It explains the problems with diesel engine pollution and the need for cleaner fuels. It also shows that biodiesel and nanoparticles can help, but most past studies looked at them separately. My review finds a gap very few studies have tested the combined effect of engine load, biodiesel, and nanoparticles or tried to find the best combination for optimization of the nanoparticle dose rate, which has yet to be done. The novelty of this work lies in the synergistic effect of three main operational variable engine load, rice bran biodiesel blend, and TiO₂ nanoparticles, to improve diesel engine performance. Unlike many studies that examine these variables independently, this study combines them using central composite design-based response surface methodology and ANOVA to develop meaningful models for emissions and engine performance. The use of rice bran biodiesel together with TiO₂ nanoparticles is an innovative approach to reduce engine emissions with a gain in performance without requiring any major modifications in the diesel engine making it a promising and environmentally friendly alternative for future diesel fuels.

Biodiesel Blends

Transesterification converts triglycerides to esters, which is how biodiesel is made from vegetable oils or animal fats. It possesses qualities that are similar to those of fossil diesel. Bio-diesel and environmentally beneficial diesel fuel alternatives for diesel engines [2, 20]. When compared to diesel fuel, biodiesel has a higher viscosity and density, as well as a higher cetane number, pour point, and flash point. Biodiesel is a full-oxygen fuel with a 10–15 percent oxygen content by weight and no sulfur [11, 21]. When compared to petro-diesel, using biodiesel in engines reduces CO, HC, and particulate matter emissions dramatically [22]. When burned as a fuel, biodiesel is appealing because it is biodegradable, sulfur-free, non-toxic, and can greatly lower exhaust emissions and overall life cycle CO₂ emissions from the engine, depending upon biodiesel blend as per ASTM standard [23, 24].

The emissions reduction advantage is roughly proportional to the blend level; for example, B20 has a 20% pollution reduction advantage over B100 [25, 26]. The load on the engine also has an impact on emission reduction. Researchers found that at full load, the mean BTE of Biodiesel blends was around 7 to 10 percent lower than that of pure diesel, but at lower loads, this variance was as high as 17 percent, which might be ascribed to Biodiesel blends

much poorer efficiency, particularly at lower loads [11, 27, 28].

Nano Additives in Biodiesel Blend

Nano additives are nanoparticles that can improve the quality of biodiesel so that it can compete with diesel. The surface area of nanoparticles, where the heterogeneous reaction takes place, is much larger than that of bulk particles [29]. It increases the number of reaction sites available for the reaction to take place. The atoms on the surface are more unstable and reactive. This instability stems from their lattice location, which forces them to unbind their neighboring atoms or molecules [30, 31]. The instability and reactivity rise as the surface/bulk particles ratio increases. As a result, nanoparticles have a large surface-to-volume ratio [32, 33].

The use of nano-additives in biodiesel has substantial benefits, including increased lubricity, energy content, stability, and oxidation resistance [34]. These nanoparticles help to increase power output and reduce fuel consumption. Moreover, nano-additives improve heat transfer, catalytic activity, and fuel atomization, resulting in fewer NO_x, CO, and hydrocarbon emissions. [35] These combined effects lead to improved fuel characteristics, better engine performance, lower emissions, and more efficient combustion procedures [36]. The small size of nano additives allows them to easily mix with biodiesel [13]. Furthermore, the addition of nanoadditives to some previously reported studies improved stability and other tribological features dramatically. This unique feature of Nano additives also aids in the elimination of wear/tear and choking concerns in CI engine parts [34, 36, 37].

Besides that, Qaisar Manzoor et al. (2024) reported that inhaled TiO₂ nanoparticles can accumulate in vital organs and are linked to oxidative stress, inflammation, and genotoxic effects, posing long-term health threats [38]. Jain et al. (2024) found that using TiO₂ nano-additives in diesel combustion increases fine and ultrafine particle emissions, which can penetrate deep into the lungs and pose serious respiratory health risks [39].

DESIGN OF EXPERIMENT

Depending upon operational conditions, examine how changes in variables (operational conditions) affect a desired response, along with choosing the right operational conditions to meet the requirements. [1, 40]

In this study, we are considering three continuous variables, denoted as L, B, and N. To obtain a design matrix by central composite design, the total number of experiments run is obtained by using a two-level full factorial equation

$$E = 2^n + 2n + n_0 \quad (1)$$

$$E = 2^3 + (2 \times 3) + 0$$

$$E = 14$$

Where E = the total number of experiments

n = Number of Operational Variables

n0 = Center points in axial

Design Summary

Factors:	3	Replicates:	1
Base runs:	14	Total runs:	14
Base blocks:	1	Total blocks:	1
Two-level factorial: Full factorial			

Point Types

Cube points:	8	Center points in cube:	0
Axial points:	6	Center points in axial:	0

Design Table (Randomized)

A single-cylinder, four-stroke diesel engine coupled with an eddy current-type dynamometer is taken into account to perform all experiments [41]. For online performance assessment, the LabVIEW-based Engine Performance Analysis software package is also present in the setup.

The maximum load-carrying capacity of the engine is 5 kW at a maximum rpm of 1500. It's an Indian make engine, made by Bharat DX, a very renowned name in the field of diesel engines. The engine has a stroke of 110 mm, a bore diameter of 87.5 mm, a cylinder volume of 661 cc, and a compression ratio (CR) of 17.5. It is fitted on a base frame having only one balance wheel and all other provisions for cooling water and temperature measurements.

Table 1. Experiment Run-Order Matrix with Formulations

Std Order	Run Order	Pt Type	Blocks	L Load (KW)	B Bio-Diesel Blend (%)	N Nano Particles (ppm)	Formulates
01	02	03	04	05	06	07	08
5	1	1	1	1.0	0	200	B0N200
8	2	1	1	3.5	30	200	B30N200
13	3	-1	1	2.0	15	0	B15N0
9	4	-1	1	1.0	15	100	B15N100
7	5	1	1	1.0	30	200	B30N200
12	6	-1	1	2.0	30	100	B30N100
3	7	1	1	1.0	30	0	B30N0
2	8	1	1	3.5	0	0	B0N0
6	9	1	1	3.5	0	200	B0N200
14	10	-1	1	2.0	15	200	B15N200
1	11	1	1	1.0	0	0	B0N0
11	12	-1	1	2.0	0	100	B0N100
10	13	-1	1	3.5	15	100	B15N100
4	14	1	1	3.5	30	0	B30N0

Table 1 presents a total of 14 experimental runs. Each run corresponds to a specific combination of three operational variables. The engine load varies from a minimum of 1.00 kW to a maximum of 3.50 kW, while the biodiesel blend ranges from 0% to 30%. The concentration of nanoparticles is adjusted between 0 ppm and 200 ppm. For each experimental run order, all six responses were recorded. To minimize errors in experimental data principle of replication is taken into account.

EXPERIMENTAL SETUP AND WORK

There are three parts to the work.

- Setting up the test rig
- Preparation of fuel as per the requirement of the experiment
- Collecting data by performing all experiments

Bharat DX makes an eddy current-based dynamometer of 5 KVA capacity with 1500 rpm at 50Hz. The dynamometer is directly connected to the engine shaft. It is also fixed in the same frame. It is air-cooled, so easy to operate and maintain. The output of the dynamometer is directly connected to the Rheostat.

A rheostat of capacity 3.5 kW is used to create a load on the engine. There are a total of seven loads, each of 0.5 kW, arranged. Each load has an individual ON-OFF switch, and a main Power Switch is used to put the load on the engine. By operating individual ON-OFF switches load can be applied in additional order.

To measure exhaust gas AIRVISOR model number AVG-500 was used. It is interconnected to the computer system. The probe of the gas analyzer is inserted into the exhaust line of the engine. The airvisor itself has a small display

unit to display all parameters associated with exhaust gas measurement.

The blending of biodiesel and nanoparticles with pure diesel fuel has some issues, like biodiesel has high viscosity and density, leading to poor fuel atomization and combustion [11, 22]. It also has poor cold flow properties, causing starting problems in low temperatures [22], and oxidation instability, which affects long-term storage [34]. Other issues include phase separation, injector clogging, and material compatibility with rubber parts [27, 28]. To reduce these problems, transesterification is used. This chemical process converts raw oils into biodiesel with lower viscosity and better fuel properties, improving blending and engine compatibility.

For nanoparticles, the main issues are agglomeration, poor dispersion, and settling down during storage, which reduces effectiveness and may block injectors [36]. To address this, ultrasonication is used. High-frequency sound waves break up clumps and evenly disperse nanoparticles in the fuel. This method ensures better suspension, stable blending, and improved combustion. Together, transesterification and ultrasonication provide effective solutions for improving the stability, performance, and safety of biodiesel-nanoparticle blends in diesel engines.

Uncertainties Associated and Minimization

To minimize errors and variability in experimental measurements, the following steps were taken:

1. Fuel blending inconsistencies: Standardize blending procedures, implement quality control, and verify fuel composition through comparative before-use analysis.
2. Ambient condition variations: Conduct experiments in controlled environments, monitor conditions, and calibrate instruments for environmental factors.

3. Instrument calibration limitations: using regularly calibrated instruments, high-precision equipment, and validating results against reference methods.

To reduce uncertainties (errors) in experimental data measurement; the principle of replication is used to find the best experimental result. At its core, replication means repeating an experiment or measurement under the same parameters till the same response of the last two consecutive runs to assess variability, reduce random error, and increase confidence in the conclusions drawn. In the analysis phase, replication provides degrees of freedom for the estimation of the error term. That's why in ANOVA, the error term is calculated from the deviations within each replicated group with a 95% level of confidence and 5% significance factor. More replications mean a more accurate and stable estimate of the error variance.

In this work, the Central Composite Design under RSM also helps to reduce uncertainty by minimizing the number of experiments while still giving a statistically strong model.

Blended Fuel with Nanoparticle Formulation

Firstly, as per the ASTM D6751 standard, the rice bran biodiesel, supplied by a reputable Indian supplier, was blended with pure diesel sourced from a well-known Indian company's filling station, according to the design matrix [23]. TiO_2 in powder form, procured from Adnano, Karnataka, India, was then added to the rice bran biodiesel-diesel mixture, following the design matrix, to produce a stable nanofluid solution [42, 43]. To prevent nanoparticle agglomeration, the TiO_2 -blended biodiesel underwent ultrasonication using a bath-type ultrasonicator operating at a frequency of 40 kHz and a power of 80W for three hours at a temperature of 30-40°C, resulting in a clear, stable solution [15]. Following this, transesterification was carried out, and a high-shear magnetic mixer was used to

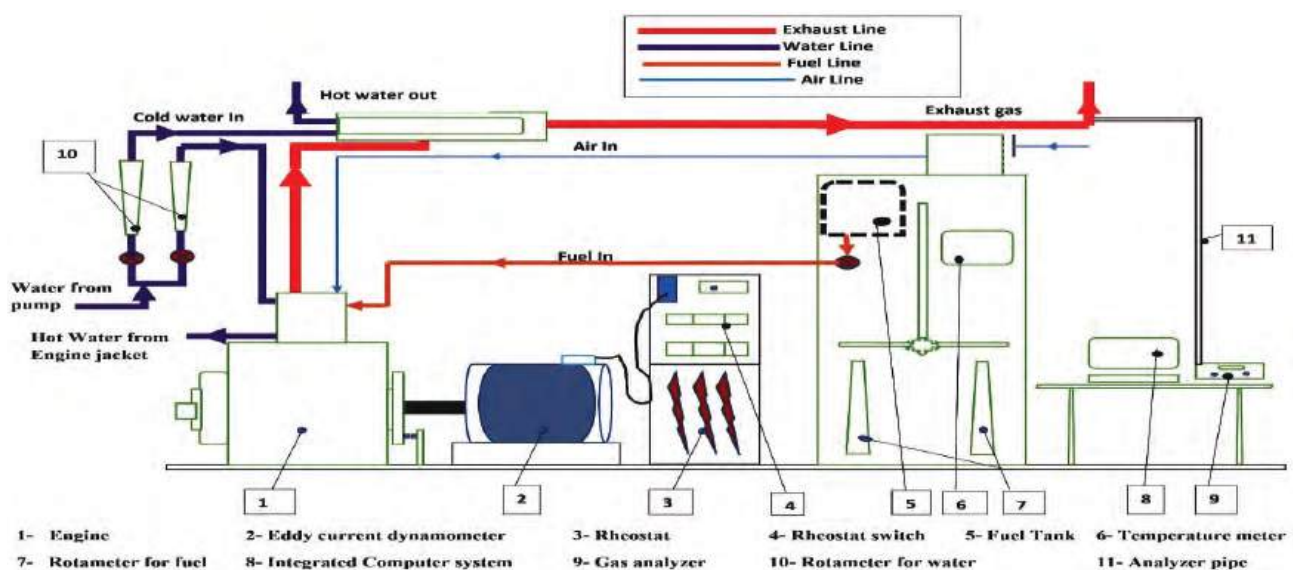


Figure 2. Schematic line diagram of test rig.

ensure thorough mixing for two hours at 40°C. This process applied mechanical force to evenly disperse the nanoparticles in the biodiesel, producing a homogeneous, stable solution free from sedimentation [44]. The formulation of fuel is given in Table 01 in column no. 08.

Observation Table for Experiment Run Order

Table 2 shows the different parameter values obtained during the operation of the diesel engine test rig. Column no. 01 shows the experiment run order no as per DOE. Column no. 02 shows the fuel used for the experiment.

Column 03 shows the load on the engine in Kw. Column no. 04 shows the time in minutes and seconds to consume 20ml of fuel. Columns no-05 to 09 show the value of different response variables.

Calculation of Performance Parameters

Total fuel consumption (TFC)

Total fuel consumption denotes the fuel consumed per unit of time. The unit of TFC is Kg/Sec.

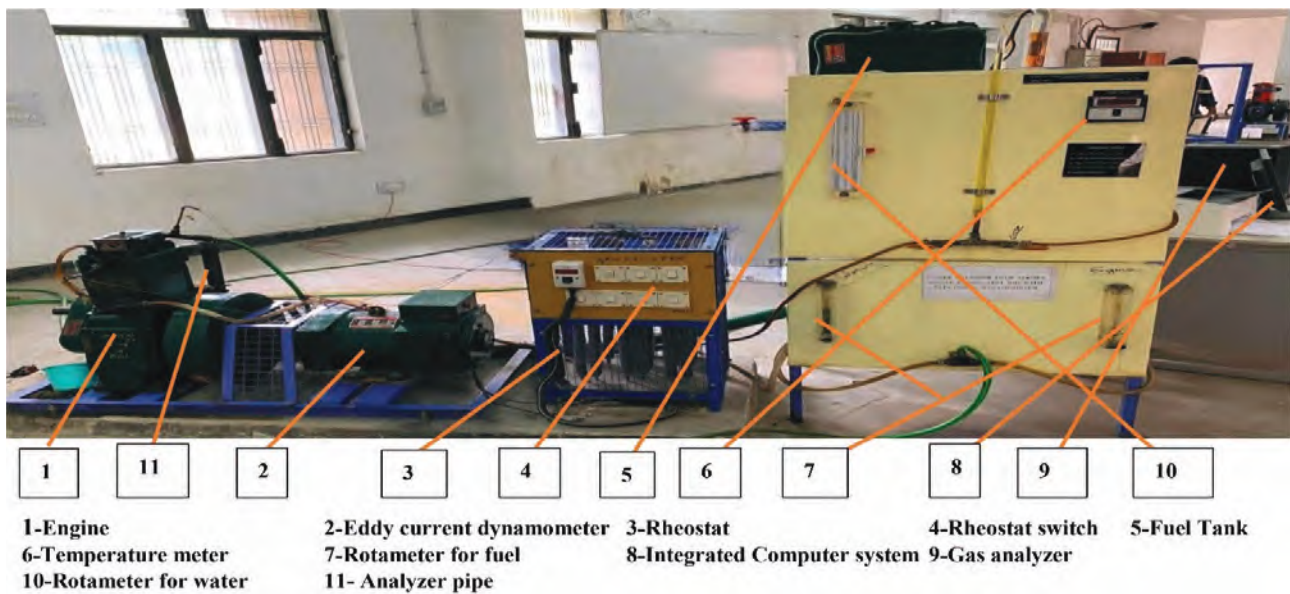


Figure 3. Actual test rig.

Table 2. Diesel Engine Test Rig Observation Data

Run Order	FUEL	Load	20ml Fuel consumption time (Sec)	HC	CO ₂	NO _x	O ₂	CO
01	02	03	04	05	06	07	08	09
1	B0N200	1.0	225.3	29	2.60	305	18.53	0.08
2	B30N200	3.5	74.3	14	4.40	552	15.87	0.06
3	B15N0	2.0	86.9	25	3.40	300	14.90	0.05
4	B15N100	1.0	180.9	30	2.50	280	18.23	0.07
5	B30N200	1.0	200.4	26	2.40	330	18.34	0.07
6	B30N100	2.0	107	16	3.20	303	16.40	0.06
7	B30N0	1.0	113.2	30	2.50	233	17.75	0.07
8	B0N0	3.5	54.4	9	4.50	556	14.35	0.04
9	B0N200	3.5	68.8	9	4.20	578	17.80	0.06
10	B15N200	2.0	124.7	25	3.20	350	16.80	0.06
11	B0N0	1.0	178.0	27	2.70	257	16.45	0.06
12	B0N100	2.0	124.2	12	3.30	332	16.20	0.06
13	B15N100	3.5	64.5	14	4.65	520	15.65	0.05
14	B30N0	3.5	47.3	19	4.70	501	13.69	0.06

$$TFC = \frac{\text{Fuel consumed}}{\text{Time}} \times \frac{\text{Specific Gravity}}{1000} \quad (2)$$

Where

Fuel consumed in ml

Time in Seconds

Brake-specific fuel consumption (BSFC)

Brake-specific fuel consumption is the total fuel consumption per unit brake power. Here, the unit of BSFC is Kg/kW Hr.

$$BSFC = \frac{TFC}{BP} \times 3600 \quad (3)$$

Where

TFC is total fuel consumption in Kg/Sec

BP is load in kW

1.1.1 Brake thermal efficiency (BTE)

It is calculated by the following equation

$$BTE = \frac{BP}{TFC \times CV} \times 100 \quad (4)$$

Where

BP is load in kW

TFC is total fuel consumption in Kg/Sec

CV is the Calorific value of fuel in KJ/Kg

RESULTS AND ANALYSIS

As per the design of the experiment, all 14 sets of operation variable tests were performed, and the 06 response variables were recorded along with the engine performance parameters. All values are given below in Table 3.

Table 6 shows that the regression models for all response variables (HC, CO₂, NO_x, O₂, CO, and BTE) are highly accurate, with R² values above 99% in most cases. The low standard error (S) indicates a good fit, and the high adjusted and predicted R² values confirm strong model performance and reliability in predictions. Overall, all models are robust and well-suited for analysis and prediction.

The confidence level of analysis is taken at 95%, so the significance factor (α) is 5%. If the significance factor of any response variable for any operational variable is more than 5% (P > 0.05), then it means the null hypothesis is satisfied and indicates no dependence on the operational variable. If it is less than 5% (P ≤ 0.05), then it means the null hypothesis is rejected and the alternate hypothesis is accepted, suggesting a significant impact of the operational variable [40].

Tables 4 and 5 show that all three operational variables (L, B, and N) significantly affect all six responses (P ≤ 0.05). For HC and CO₂, the load (L) is the most significant factor due to its lowest p-value and highest F-test value, while biodiesel (B) is the least significant. For NO_x, L is again the most significant, while nanoparticle (N) is the least. In the case of O₂, L is most significant, followed by N, while B remains the least significant.

Table 3. Result table for each run-order

Run Order NO	FUEL	Load	20ml Fuel consumption time (Sec)	HC (ppm)	CO ₂ (% of total Sample)	NO _x (ppm)	O ₂ (% of the total sample)	CO (% of the total sample)	TFC (Kg/Hr)	BSFC (Kg/Kw Hr)	BTE (%)
01	02	03	04	05	06	07	08	09	10	11	12
1	B0N200	1.0	225.3	29	2.60	305	18.53	0.08	0.2690	0.2690	30.2
2	B30N200	3.5	74.3	14	4.40	552	15.87	0.06	0.8279	0.2365	33.5
3	B15N0	2.0	86.9	29	3.40	287	14.90	0.05	0.6899	0.3449	24.7
4	B15N100	1.0	180.9	30	3.40	305	18.23	0.07	0.3342	0.3342	24.5
5	B30N200	1.0	200.4	26	2.30	330	18.34	0.08	0.3071	0.3071	25.8
6	B30N100	2.0	107	16	2.80	303	16.40	0.05	0.5718	0.2859	28.4
7	B30N0	1.0	113.2	30	2.50	202	17.75	0.065	0.5327	0.5327	15.8
8	B0N0	3.5	54.4	14	4.90	556	14.35	0.06	1.0982	0.3137	26.2
9	B0N200	3.5	68.8	9	3.50	472	17.80	0.06	0.8801	0.2514	32.3
10	B15N200	2.0	124.7	25	2.80	305	16.80	0.055	0.4894	0.2447	32.8
11	B0N0	1.0	178.0	32	4.10	357	16.45	0.08	0.3355	0.3355	24.5
12	B0N100	2.0	124.2	16	3.30	340	16.20	0.06	0.4843	0.2421	33.9
13	B15N100	3.5	64.5	15	4.90	520	15.65	0.05	0.9367	0.2676	30.6
14	B30N0	3.5	47.3	19	4.40	458	13.69	0.05	1.2754	0.3644	23.1

Table 4. ANOVA table for CO₂, NO_x, and O₂

Source	DF	HC				CO ₂				NO _x			
		Adj SS	Adj MS	F-Value	P-Value	Adj SS	Adj MS	F-Value	P-Value	Adj SS	Adj MS	F-Value	P-Value
Model	9	776.604	86.289	1363.32	0.000	10.1657	1.12952	252.90	0.000	162380	18042	1914.62	0.000
Linear	3	625.551	208.517	3294.44	0.000	6.8949	2.29831	514.59	0.000	116375	38792	4116.52	0.000
L	1	577.600	577.600	9125.72	0.000	5.1840	5.18400	1160.70	0.000	112148	112148	11901.01	0.000
B	1	3.304	3.304	52.21	0.002	0.3546	0.35459	79.39	0.001	3214	3214	341.03	0.000
N	1	44.647	44.647	705.39	0.000	1.3563	1.35634	303.69	0.000	1013	1013	107.51	0.000
Square	3	131.582	43.861	692.97	0.000	1.1623	0.38744	86.75	0.000	16984	5661	600.76	0.000
L*L	1	10.571	10.571	167.02	0.000	1.0291	1.02907	230.41	0.000	14952	14952	1586.68	0.000
B*B	1	86.779	86.779	1371.05	0.000	0.2309	0.23087	51.69	0.002	178	178	18.87	0.012
N*N	1	63.087	63.087	996.73	0.000	0.1616	0.16163	36.19	0.004	711	711	75.49	0.001
2-Way Interaction	3	29.922	9.974	157.58	0.000	1.5879	0.52929	118.51	0.000	18107	6036	640.50	0.000
L*B	1	28.651	28.651	452.66	0.000	0.6675	0.66746	149.44	0.000	1556	1556	165.07	0.000
L*N	1	1.146	1.146	18.11	0.013	0.0092	0.00917	2.05	0.225	531	531	56.35	0.002
B*N	1	0.125	0.125	1.97	0.233	0.9112	0.91125	204.03	0.000	16021	16021	1700.07	0.000
Error	4	0.253	0.063			0.0179	0.00447			38	9		
Total	13	776.857				10.1836				162418			

Table 5. ANOVA table for O₂, CO, and BTE

Source	DF	O ₂				CO				BTE			
		Adj SS	Adj MS	F-Value	P-Value	Adj SS	Adj MS	F-Value	P-Value	Adj SS	Adj MS	F-Value	P-Value
Model	9	30.0541	3.3393	151.45	0.000	0.001676	0.000186	78.39	0.000	331.000	36.778	109.04	0.000
Linear	3	25.1588	8.3863	380.34	0.000	0.001111	0.000370	155.90	0.000	264.049	88.016	260.95	0.000
L	1	14.2564	14.2564	646.56	0.000	0.000903	0.000903	379.84	0.000	62.001	62.001	183.82	0.000
B	1	0.2151	0.2151	9.75	0.035	0.000120	0.000120	50.64	0.002	39.644	39.644	117.54	0.000
N	1	10.6873	10.6873	484.69	0.000	0.000088	0.000088	37.21	0.004	162.404	162.404	481.50	0.000
Square	3	2.3353	0.7784	35.30	0.002	0.000608	0.000203	85.33	0.000	48.533	16.178	47.96	0.001
L*L	1	2.1198	2.1198	96.14	0.001	0.000327	0.000327	137.75	0.000	29.190	29.190	86.54	0.001
B*B	1	0.0071	0.0071	0.32	0.600	0.000054	0.000054	22.76	0.009	1.000	1.000	2.97	0.160
N*N	1	0.3865	0.3865	17.53	0.014	0.000012	0.000012	4.96	0.090	7.647	7.647	22.67	0.009
2-Way Interaction	3	3.8418	1.2806	58.08	0.001	0.000085	0.000028	11.91	0.018	25.666	8.555	25.36	0.005
L*B	1	1.7824	1.7824	80.84	0.001	0.000004	0.000004	1.64	0.270	16.343	16.343	48.45	0.002
L*N	1	1.1072	1.1072	50.21	0.002	0.000003	0.000003	1.20	0.334	0.078	0.078	0.23	0.656
B*N	1	0.9522	0.9522	43.18	0.003	0.000078	0.000078	32.88	0.005	9.245	9.245	27.41	0.006
Error	4	0.0882	0.0220			0.000010	0.000002			1.349	0.337		
Total	13	30.1423				0.001686				332.349			

Table 6. ANOVA model assessment

Model	S	R-sq	R-sq(adj)	R-sq(pred)
HC	0.251582	99.97%	99.89%	99.43%
CO ₂	0.0668302	99.82%	99.43%	98.07%
NO _x	3.06976	99.98%	99.92%	99.77%
O ₂	0.148491	99.71%	99.05%	96.79%
CO	0.0015414	99.44%	98.17%	91.56%
BTE	0.580766	99.59%	98.68%	95.95%

Table 7. Regression relation of variables

HC =	44.293 - 13.537L + 0.6122B - 0.11437N + 1.388L * L - 0.026389B * B + 0.000506N * N + 0.10053L * B - 0.003016L * N - 0.000083B * N
CO ₂ =	5.281 - 1.630L - 0.02876B - 0.002544N + 0.4332L * L - 0.001361B * B - 0.000026N * N + 0.01534L * B + 0.000270L * N + 0.000225B * N
NO _x =	459.14 - 154.86L - 6.980B + 0.1393N + 52.21L * L + 0.03778B * B - 0.001700N * N + 0.7407L * B - 0.06492L * N + 0.029833B * N
O ₂ =	19.510 - 3.673L + 0.0625B + 0.01505N + 0.6217L * L + 0.000239B * B - 0.000040N * N - 0.02507L * B + 0.002964L * N - 0.000230B * N
CO =	0.11481 - 0.04244L - 0.001148B - 0.000035N + 0.007725L * L + 0.000021B * B + 0.00000N * N + 0.000037L * B - 0.000005L * N + 0.000002B * N
BTE =	15.64 + 11.16L - 0.4604B + 0.06307N - 2.307L * L + 0.00283B * B - 0.000176N * N + 0.0759L * B + 0.00079L * N + 0.000717B * N

For CO, L is the most significant operational variable, with B being the second most significant, and N the least. In BTE analysis, N is the most significant, followed by L, with B being the least. These trends are further supported by the sum of squares (SS) data for each operational variable.

The ANOVA model assessment is presented in Table 6, showing that values of R-sq, R-sq (adj), and R-sq (pred) are more than 90% for each response. All values of R² are very high, so it means the model is highly matched with the experimental data. The equations presented in Table 7 employ the least-squares method to derive second-order polynomial equations corresponding to each distinct engine response.

Response Optimization: BTE, CO, NO_x, CO₂, HC

Response Optimizer Function of Minitab software with 95 % of the lower bound confidence level of all intervals is used to optimize the responses with the following inputs-

Parameters

The goal of minimization was set for four emissive responses: HC, CO₂, NO_x, and CO; out of five emissive responses, except O₂. Since O₂ is not a harmful emissive, so does not need to be optimized. BTE is set for maximum. All responses have the same weightage and responses as given in Table 8.

Variable Ranges

The range of operation variables is set as per the given Table 9.

Table 8. Table of parameters of responses to rsm

Response	Goal	Lower	Target	Upper	Weight	Importance
BTE(%)	Maximum	15.8	33.90		1	1
CO	Minimum		0.05	0.08	1	1
NO _x	Minimum		202.00	556.00	1	1
CO ₂	Minimum		2.30	4.90	1	1
HC	Minimum		9.00	32.00	1	1

Table 9. Table of variable range for RSM

Variable	Values
L	(1, 3.5)
B	(0, 30)
N	(0, 200)

Solution

The optimization results show a fair balance between emissions (CO, NO_x, CO₂, HC) and efficiency (BTE), with a desirability score of 0.737499. The Brake Thermal Efficiency is optimized to 32.51%, while emissions are

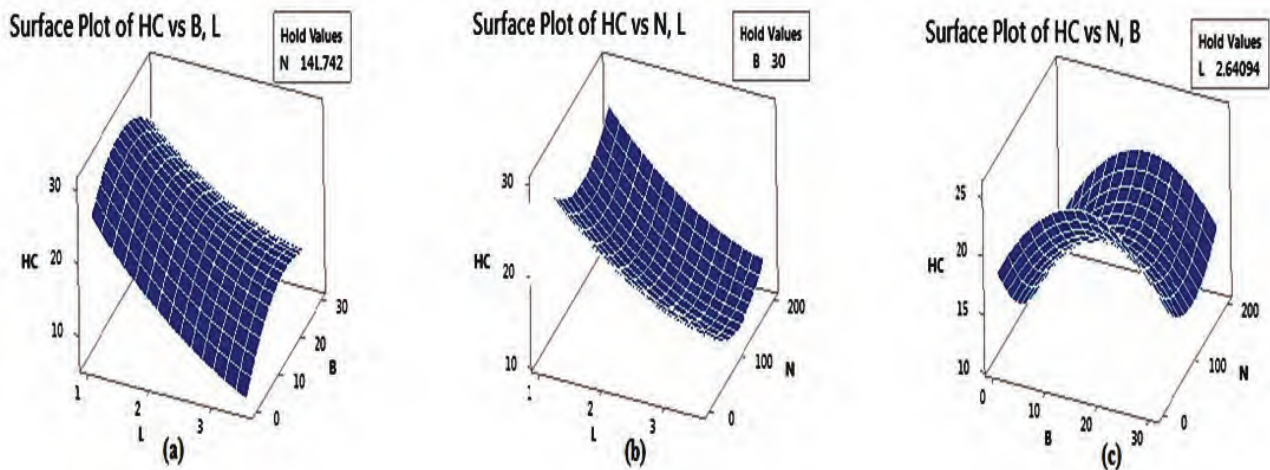
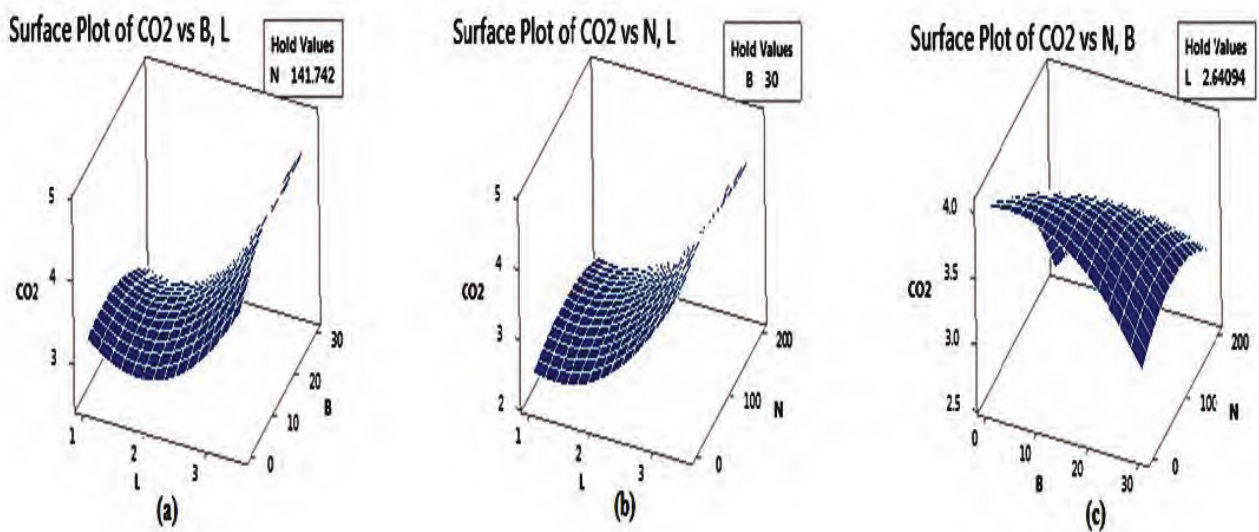
minimized, indicating a reasonably effective balance between efficiency and emissions.

Figure 10 shows the nature and variations of each response concerning each operation variable for the optimized result, the solution obtained in Table No. 10.

Surface Plots of Optimized Responses

The set of figures (Figures 4 to 9) illustrates various surface plots of different emissions and parameters, each plotted against combinations of variables B & L, N & L, and N & B.

Starting with the HC plots, the first graph shows that B and L parameters are influencing in a coordinated way, leading to an interplay that can result in varying HC concentrations depending on their respective values. This

**Figure 4.** Surface Plot of HC with respect to (a) B&L, (b) N&L, (c) N&B.**Figure 5.** Surface Plot of CO₂ with respect to (a) B&L, (b) N&L, (c) N&B.

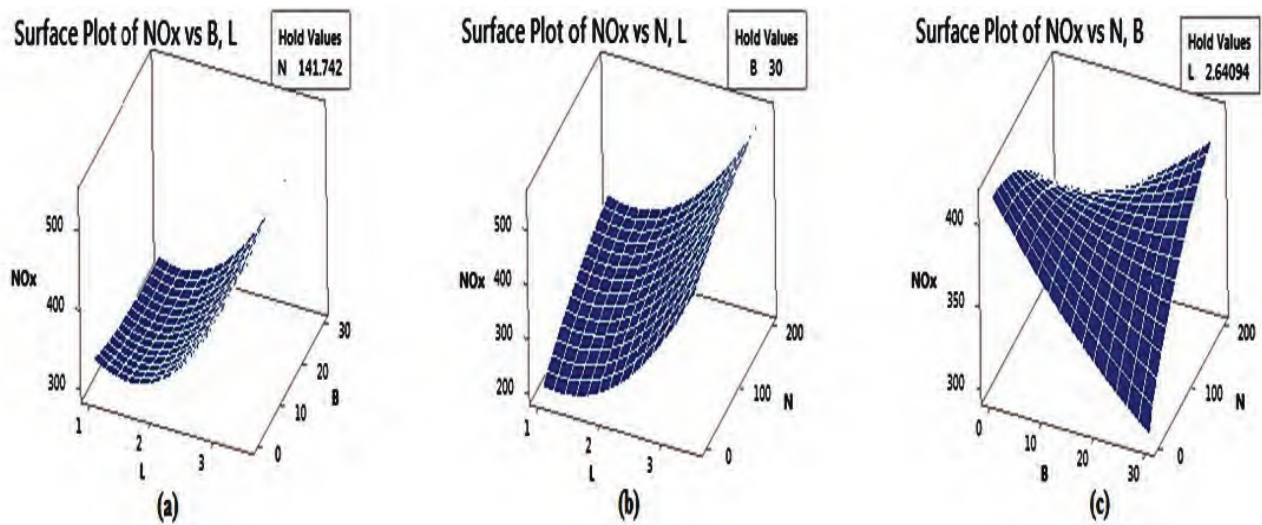


Figure 6. Surface Plot of NOX with respect to (a) B&L, (b) N&L, (c) N&B.

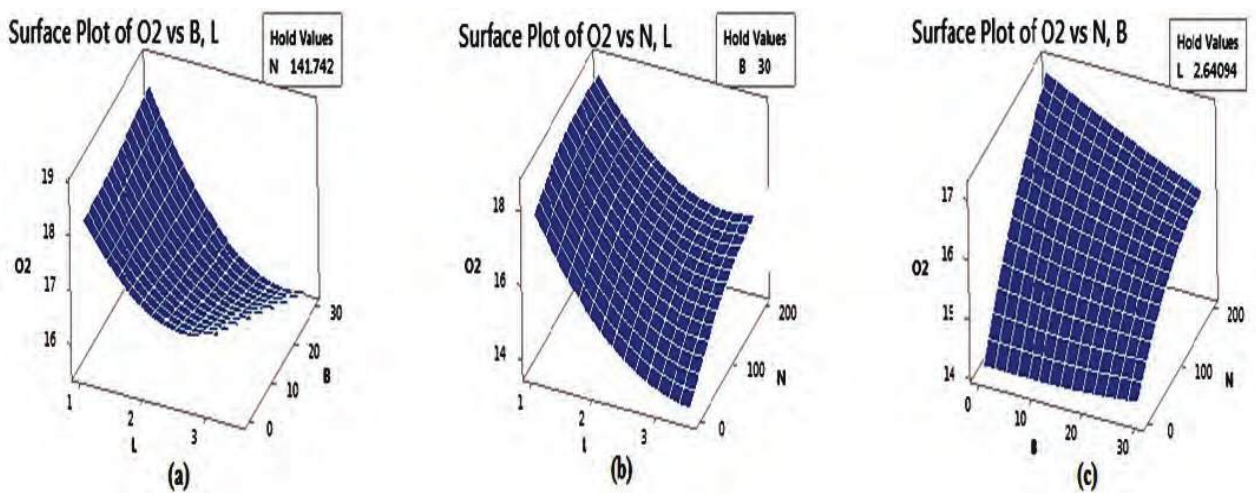


Figure 7. Surface Plot of O₂ with respect to (a) B&L, (b) N&L, (c) N&B.

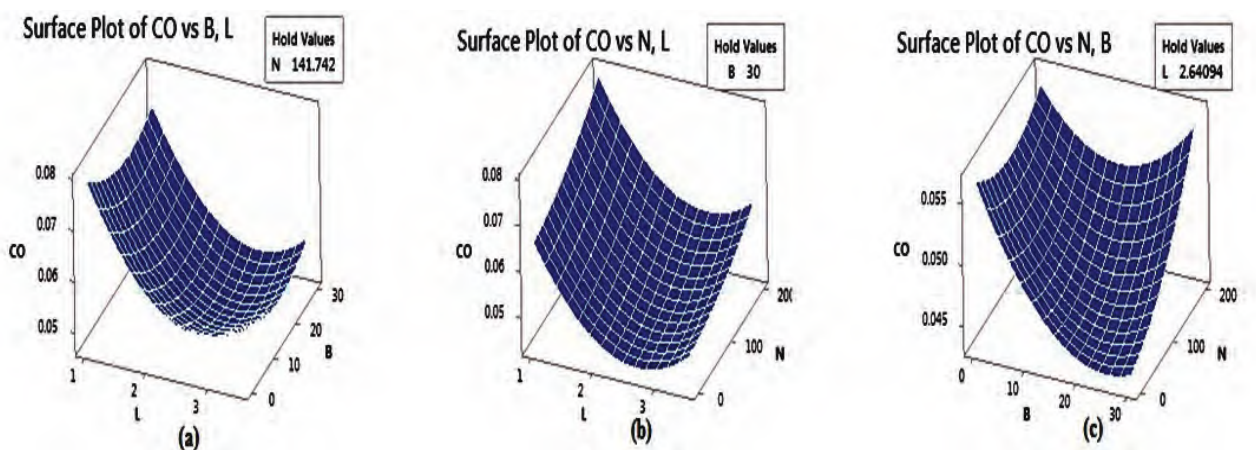


Figure 8. Surface Plot of CO with respect to (a) B&L, (b) N&L, (c) N&B.

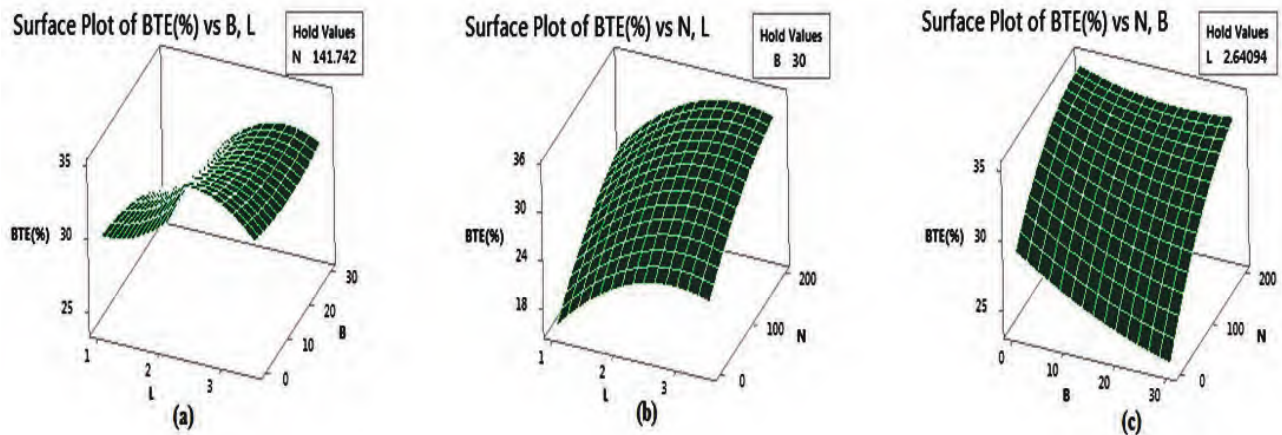


Figure 9. Surface Plot of BTE with respect to (a) B&L, (b) N&L, (c) N&B.

Table 10. Optimize solution by RSM

Variables	Solution
L	2.64094
B	30.00
N	141.742
BTE (%) Fit	32.5088
CO Fit	0.0504219
NO _x Fit	385.780
CO ₂ Fit	3.30655
HC Fit	13.2912
Composite Desirability	0.737499

suggests that biodiesel, possibly due to its cleaner combustion properties, leads to lower HC emissions, especially at higher loads. In the second plot of HC versus N and L, the non-linear behavior suggests a more complex relationship between these two, where increasing one or both has a significant and potentially non-additive impact on HC levels, with nanoparticles contributing to a further reduction in HC emissions as load increases. The third HC plot, which examines N and B, indicating strong nonlinear interactions, implies that when varied together, they have a pronounced effect on HC emissions, mostly magnifying or reducing emissions. This might be due to the way nanoparticles enhance combustion or act as catalysts.

The first plot of CO₂ shows a surface that gradually rises. This indicates that both variables, B and L, contribute to increasing CO₂ emissions, as would be expected in a combustion process where higher loads require more fuel. Biodiesel shows a modest effect on CO₂ levels, which might be due to the chemical composition of biodiesel itself. In the second CO₂ plot, the surface still exhibits a slight curvature,

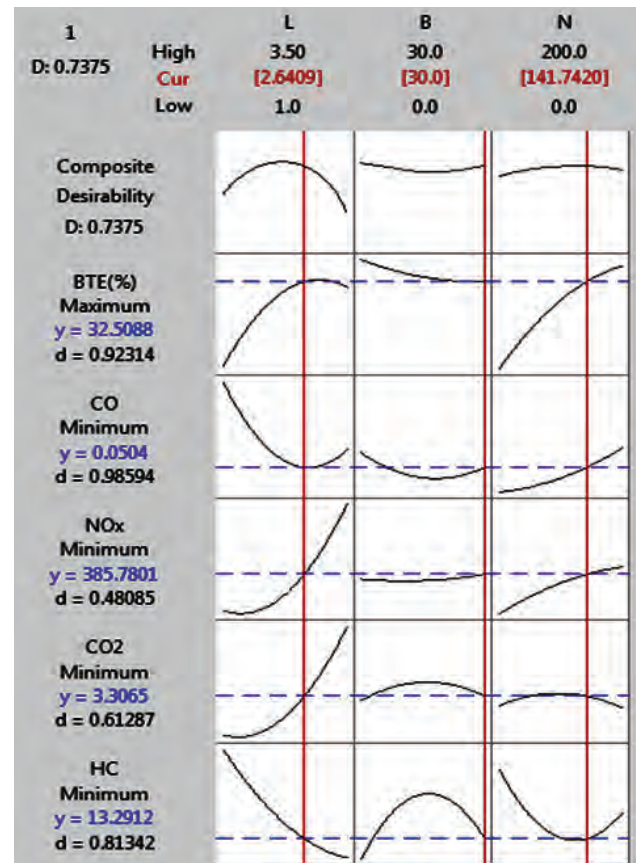


Figure 10. Optimization of responses.

indicating that variables do interact, although the effect seems more gradual [30]. The third CO₂ plot shows more curvature than the first two, suggesting that N and B together have a more dynamic effect on CO₂ emissions.

The first NO_x plot shows a gentle slope, suggesting that as B and L increase, NO_x levels rise, but the surface

suggests that this increase is smoother compared to the HC and CO₂ cases. In the second NO_x plot, a stronger upward curve is observed, indicating that as either N or L increases, NO_x levels rise more sharply, particularly when both variables are at higher levels due to a non-linear relationship. The third NO_x plot reveals the most complex surface of the three, with the possibility of both variables either amplifying or mitigating each other's impact on the system.

The first plot of O₂ shows that oxygen levels decrease as both load and bio-diesel concentration increase. This suggests that higher loads and more biodiesel content in the blend result in a richer combustion process, which consumes more oxygen. In the second plot, as the load increases and the nanoparticle concentration rises, oxygen levels tend to decrease. This trend implies that the introduction of nanoparticles, along with an increased load, intensifies the combustion process, leading to more oxygen being consumed. The third plot of O₂ examination shows that Oxygen concentration decreases as both nanoparticle and bio-diesel content increase, again indicating more complete combustion with higher bio-diesel and nanoparticle concentrations [45].

The first plot of CO, which focuses on the relationship between B and load L, shows a slight increase in CO emissions as both variables increase. This suggests that under higher loads and with higher bio-diesel content, incomplete combustion becomes more likely, producing more CO. The second plot shows that as load increases, CO levels tend to decrease slightly, while increasing nanoparticle content results in higher CO emissions. The nanoparticles might enhance combustion efficiency [29]. The third plot of CO shows that CO emissions decrease as nanoparticle concentration rises, but at the same time, CO increases

slightly as bio-diesel concentration increases. This behavior could be due to the complex interaction between bio-diesel properties and nanoparticle effects on the combustion process.

At the last the very first surface plots of BTE, the brake thermal efficiency tends to increase with moderate loads and bio-diesel concentrations, but starts to decline when either load or bio-diesel concentration becomes too high. This suggests that the combustion process is most efficient at moderate levels of load and bio-diesel, but higher loads and bio-diesel concentrations lead to inefficiencies, possibly due to incomplete combustion or higher exhaust losses. In the second plot, N and L affect BTE, with B held constant. Here, BTE generally increases as nanoparticle concentration rises, indicating that nanoparticles help improve the combustion process by enhancing the heat release and improving the overall efficiency [46]. However, higher loads seem to reduce efficiency after a certain point. The third plot, which explores the effect of nanoparticle concentration and bio-diesel content with a load constant, reveals that brake thermal efficiency increases with nanoparticle concentration, especially at higher bio-diesel levels. This suggests that the addition of nanoparticles into the bio-diesel blend helps to counteract some of the efficiency losses typically associated with using bio-diesel by improving combustion stability and heat transfer.

Validation

The optimization results obtained by RSM had a desirability of 0.737 for engine load 2.64 KW, rice bran bio-diesel blend 30 Ppm, and TiO₂ nanoparticle 141.74 Ppm. To fulfill the objective of the current work, the validation of these new findings is necessary, so a confirmation experiment is done by taking operational variables:

Table 11. Properties of conformation test fuel [47, 48, 49]

Fuel	Time in Sec (20ml consumption of fuel)	Specific gravity	Viscosity (mm ² /Sec)	Calorific value (KJ/Kg)	Load (Kw)	TFC (Kg/Hr)	BSFC (Kg/Kw Hr)
B30N150	98.5	0.852	4.58	44880	2.5	0.6227	0.24908

Table 12. Verification experiment test of responses

Responses	Predicted value	Experimental value	Error (%)
BTE	32.56	32.2	1.11
CO	0.051	0.05	2.0
O ₂	15.71	15.25	3.01
NO _x	385.78	375	2.87
CO ₂	3.30	3.2	3.12
HC	13.29	13	2.23

Load (L) = 2.5 Kw

Rice bran bio-diesel (B) = 30 Ppm

TiO₂ nanoparticle (N) = 150 Ppm

Here, the operational variable's value is taken to the nearest complete factors of optimized results due to the ease of setting up of experiment on the engine. Here, the principle of replication is used to find the best experimental result. The properties of the fuel are:-

In addition, projected values identified with the help of the model were utilized for validation, together with experimental values. The percentage of errors displayed in Table 12, demonstrates that the proposed models, which incorporate RSM for BTE, CO, NO_x, CO₂, HC, and O₂, are reasonably capable of predicting the impact of 2.5 Kw engine load, 30 Ppm rice bran biodiesel blend, and 150 Ppm TiO₂ nanoparticle on diesel engine combustion. Since the expected error is less than 4%, it is acceptable and should be approved.

These findings demonstrate the potential of rice bran biodiesel blended with TiO₂ nanoparticles as a clean, efficient fuel for CI engines. The optimized parameters reduce emissions and improve engine performance, making this approach suitable for practical applications. Future research can focus on scaling, exploring alternative nanoparticles, long-term engine impact, and integration with smart engine control systems to advance sustainable fuel technologies.

CONCLUSION

As per the objective, the present work is associated with synergistic optimization and the effect of three operational variables, namely Load on the engine, rice bran biodiesel Blend, and TiO₂ Nanoparticle for emission attenuation of HC, CO₂, NO_x, O₂, and CO along with BTE amelioration of diesel engine sustainability. Because of the criticalness of these responses to estimate the suitability of fuel under the influence of operational variables. The main observations are given below:-

1. The utilization of transesterification and ultrasonic techniques to produce a blend of rice bran biodiesel and TiO₂ nanoparticles has resulted in a fuel that meets the specifications outlined in the ASTM biodiesel standard, making it a viable option for use in CI engines.
2. By utilizing Central Composite Design-based Response Surface Modeling, a Design of Experiments approach can potentially identify the optimal parameter settings necessary to achieve the desired level of performance. This can be accomplished through the use of the composite desirability method, which can provide a reasonably accurate estimation of the required parameter values.
3. The optimum combination of three operation parameters is determined and reported as 2.64094 kW engine Load, 30 ppm Rice bran Bio-Diesel blend, and 141.742 ppm TiO₂ nanoparticles.

4. Additionally, the desired outcomes at the previously mentioned optimal combination included 0.050% of total sample CO, 3.30% of total sample CO₂, 13.29 ppm of HC, 385.78 ppm of NO_x, and a BTE of 32.50%, resulting in a desirability effect of 73.74%.
5. The experimental data were used to validate the results, and the validation showed an error of 1.11% for BTE, 2.0% for CO, 3.01% for O₂, 2.87% for NO_x, 3.12% for CO₂, and 2.23% for HC. These errors were within a range of below 4%.
6. Compared to earlier studies, the current work shows significant improvements. Vijayakumar et al. (2016) achieved only a 2.19% BTE increase and limited emission reductions using 20% mahua biodiesel with 50 ppm CuO. Padmanaba Sunder et al. (2021) reported a 2.64% BTE gain with 200 ppm TiO₂ in plastic oil. In contrast, our study using B30N150 at a 2.5 kW load outperforms pure diesel in both terms, of emissions and efficiency. It reduces HC by 59.4% at low load and 7.1% at high load. CO₂ emissions drop by 21.95% at low load and 34.7% at high load. CO is significantly reduced by 69.7% under low load and 16.7% under high load. Under low load, NO_x increases by 5.0 percent. But under high load, it is reduced by 32.5 percent. The blend of 30% rice bran-biodiesel and 150 ppm TiO₂ nanoparticles improves BTE by 31.4% compared to pure diesel at low load and 22.9% at high load. The B30N150 improves combustion, which leads to remarkable BTE amelioration and emission attenuation.

This study provides valuable insights into the optimization of combustion parameters for rice bran biodiesel blends with TiO₂ nanoparticles. This means the fuel is cleaner and more efficient overall. Since it works well in regular diesel engines without major changes, it could help meet strict pollution rules around the world. Also, because it uses non-edible oil and helps burn fuel better, it's a smart and eco-friendly option for the future. Stakeholders can utilize this information to enhance engine performance and combustion behavior. However, the use of nanoparticles in fuels, particularly metal oxide nanoparticles, presents new challenges concerning particulate matter emissions. Additional research is needed to understand and address the impact of these nanoparticles on emissions and develop appropriate measures for emission control.

ABBREVIATIONS

ANOVA	Analysis of variance
DOE	Design of the experiment
BTE	Brake thermal efficiency
BSFC	Brake specific fuel consumption (Kg/Kw Hr)
TFC	Total fuel consumption (Kg/Hr)
TiO ₂	Titanium Dioxide
CO	Carbon monoxide
CO ₂	Carbon dioxide
HC	Hydrocarbons

NO _x	Nitrogen oxides
SO _x	Sulfur oxide
RSM	Response Surface Modeling
L	Load on Engine
B	Rice Bran Bio-Diesel blend
N	Nanoparticle, α - significance factor
CI	Compression Ignition

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Singh Pali H, Kumar M, Nguyen NV, Singh Y, Deepanraj B, Quy P, et al. Enhancement of combustion characteristics of waste cooking oil biodiesel using TiO₂ nanofluid blends through RSM. *Fuel* 2023;331:125681–125681. [Crossref]
- [2] Balajii M, Niju S. Esterification optimization of underutilized Ceiba pentandra oil using response surface methodology. *Biofuels* 2021;12:495–502. [Crossref]
- [3] Rajak R, Chattopadhyay A. Short and Long Term Exposure to Ambient Air Pollution and Impact on Health in India: A Systematic Review. *Int J Environ Health Res* 2020;30:593–617. [Crossref]
- [4] Walsh MP. Mobile source related air pollution: Effects on health and the environment. *Encycl Environ Heal* 2019;436–442. [Crossref]
- [5] International Energy Agency. *World Energy Outlook 2021*. Available at: <https://www.iea.org/reports/world-energy-outlook-2021> Accessed on Sep 5, 2025
- [6] US Energy Information Administration (EIA). *International Energy Outlook 2017 Overview*. Available at: [https://www.eia.gov/outlooks/ieo/pdf/0484\(2017\).pdf](https://www.eia.gov/outlooks/ieo/pdf/0484(2017).pdf). Accessed on Sep 5, 2025
- [7] Sustainable T, Project M. *Mobility 2030*. World Bussiness Counc Sustain Dev 2014.
- [8] ROOT MH. *International Energy Agency 2012*. Available at <https://www.iea.org/reports/world-energy-outlook-2012>
- [9] Dye C, Reeder JC, Terry RF. Research for universal health coverage. *Sci Transl Med*. 2013;5:199. [Crossref]
- [10] S. L. C. Richard PK, Allan, Arias PA, Berger S, Canadell JG, Cassou C, Chen D, Cherchi A. *Climate Change 2021*. Available at: <https://www.ipcc.ch/report/ar6/wg1/>. Accessed on Sep 5, 2025
- [11] Raheman H, Phadatare AG. Performance of compression ignition engine with mahua (*Madhuca indica*) biodiesel. *Fuel* 2007;86:2568–2573. [Crossref]
- [12] Makepa DC, Chihobo CH, Musademba D. Advances in sustainable biofuel production from fast pyrolysis of lignocellulosic biomass. *Biofuels*. 2023;14:529–550. [Crossref]
- [13] Chandrasekaran V, Arthanarisamy M, Nachiappan P, Dhanakotti S, Moorthy B. The role of nano additives for biodiesel and diesel blended transportation fuels. *Transp Res Part D Transp Environ*. 2016;46:145–156. [Crossref]
- [14] Sundar SP, P Vijayabalan, Hemalatha D, Ramani B, Chamkha JA, Sathyamurthy R, et al. Feasibility study of neat plastic oil with TiO₂ nanoadditive as an alternative fuel in internal combustion engine. *J Therm Anal Calorim* 2022;147:2567–2578. [Crossref]
- [15] Fasogbon SK, Oyedepo SO. Nano-additive blends examination of performance and emission profile of CI engines fuelled with waste cooking oil based-biodiesel. *J Therm Eng*. 2025;11:1–15. [Crossref]
- [16] Khankal D, Hole S, Gadekar H, Jagtap A, Pandhare A. Comparative analysis of performance and emission from single cylinder diesel engine fuelled with mango kernel biodiesel. *J Therm Eng* 2024;10:1632–1646. [Crossref]
- [17] Yamini K, Kishore PS, Dhana Raju V. Effect of diethyl ether and isobutanol as fuel additives on the diesel engine attributes fueled with subabul seed biodiesel. *J Therm Eng* 2025;11:215–225. [Crossref]
- [18] Sharma P, Le MP, Chhillar A, Said Z, Deepanraj B, Cao DN, et al. Using response surface methodology approach for optimizing performance and emission parameters of diesel engine powered with ternary blend of Solketal-biodiesel-diesel. *Sustain Energy Technol Assessments* 2022;52. [Crossref]
- [19] I Veza, Karaoglan AD, Ileri E, Afzal A, Hoang AT, Tamaldin N, et al. Multi-objective optimization of diesel engine performance and emission using grasshopper optimization algorithm. *Fuel* 2022;323. [Crossref]

- [20] Gbadeyan OJ, Muthivhi J, Liganiso LZ, Mpongwana N, Dziike F, Deenadayalu N. Recent improvements to ensure sustainability of biodiesel production. *Biofuels* 2024;15:1063–1077. [\[Crossref\]](#)
- [21] Rajan K, Prabhahar M, Senthilkumar KR. Experimental studies on the performance, emission and combustion characteristics of a biodiesel-fuelled (Pongamia methyl ester) diesel engine with diethyl ether as an oxygenated fuel additive. *Int J Ambient Energy* 2016;37:439–445. [\[Crossref\]](#)
- [22] Kumar S, Dinesha P, Bran I. Influence of nanoparticles on the performance and emission characteristics of a biodiesel fuelled engine: An experimental analysis. *Energy* 2017;140:98–105. [\[Crossref\]](#)
- [23] ASTM D6751-15c. Standard Specification for Biodiesel Fuel Blend Stock (B100) for Middle Distillate Fuels. *ASTM Int* 2010;i:1–11.
- [24] Prasad A, Sivanraju R, Teklemariam A, Tafesse D, Tufa M, Bejaxhin BH. Influence of nano additives on performance and emissions characteristics of a diesel engine fueled with watermelon methyl ester. *J Therm Eng* 2023;9:395–400. [\[Crossref\]](#)
- [25] Machado Corrêa S, Arbilla G. Carbonyl emissions in diesel and biodiesel exhaust. *Atmos Environ* 2008;42:769–775. [\[Crossref\]](#)
- [26] Raheman H, Phadatare AG. Diesel engine emissions and performance from blends of karanja methyl ester and diesel. *Biomass Bioenergy* 2004;27:393–397. [\[Crossref\]](#)
- [27] Venkata Subbaiah G, Raja Gopal K. An experimental investigation on the performance and emission characteristics of a diesel engine fuelled with rice bran biodiesel and ethanol blends. *Int J Green Energy* 2011;8:197–208. [\[Crossref\]](#)
- [28] Alam A, Kalam MA, Habibullah M, Hossain MM, Masjuki HH, Rashedul HK, et al. Impact of edible and non-edible biodiesel fuel properties and engine operation condition on the performance and emission characteristics of unmodified DI diesel engine. *Biofuels* 2016;7:219–232. [\[Crossref\]](#)
- [29] Çalhan R, Kaskun Ergani S. The impacts of nano fuels containing Fe-Ni-TiO₂/activated carbon nanoparticles on diesel engine performance and emission characteristics. *Biofuels* 2023;14:661–671. [\[Crossref\]](#)
- [30] Kumar S, Dinesha P, Bran I. Experimental investigation of the effects of nanoparticles as an additive in diesel and biodiesel fuelled engines: a review. *Biofuels* 2019;10:615–622. [\[Crossref\]](#)
- [31] Jaikumar S, Srinivas V, Rajasekhar M. Influence of dispersant added nanoparticle additives with diesel-biodiesel blend on direct injection compression ignition engine: Combustion, engine performance, and exhaust emissions approach. *Energy* 2021;224:120197. [\[Crossref\]](#)
- [32] Mahalingam S, Ganesan S. Effect of nano-fuel additive on performance and emission characteristics of the diesel engine using biodiesel blends with diesel fuel. *Int J Ambient Energy* 2020;41:316–321. [\[Crossref\]](#)
- [33] Soudagar MEM, Nik-Ghazali NN, Abul Kalam M, Badruddin IA, Banapurmath NR, Akram N. The effect of nano-additives in diesel-biodiesel fuel blends: A comprehensive review on stability, engine performance and emission characteristics. *Energy Convers Manag* 2018;178:146–177. [\[Crossref\]](#)
- [34] Yusof SNA, Sidik NAC, Asako Y, Japar WMAA, Mohamed SB, az Muhammad NM. A comprehensive review of the influences of nanoparticles as a fuel additive in an internal combustion engine (ICE). *Nanotechnol Rev* 2021;9:1326–1349. [\[Crossref\]](#)
- [35] Giwa SO, Aasa SA, Shote AS, Sharifpur M. Nanoparticles-suspended biodiesel and its blends in compression ignition engines: a bibliometric analysis of research trend and future outlook. *Biofuels* 2023;14:673–686. [\[Crossref\]](#)
- [36] Dehhaghi M, Kazemi Shariat Panahi H, Aghbashlo M, Lam SS, Tabatabaei M. The effects of nanoadditives on the performance and emission characteristics of spark-ignition gasoline engines: A critical review with a focus on health impacts. *Energy* 2021;225:120259. [\[Crossref\]](#)
- [37] Hosseini SH, Taghizadeh-Alisaraei A, Ghobadian B, Abbaszadeh-Mayvan A. Effect of added alumina as nano-catalyst to diesel-biodiesel blends on performance and emission characteristics of CI engine. *Energy* 2017;124:543–552. [\[Crossref\]](#)
- [38] Manzoor Q, et al. Toxicity spectrum and detrimental effects of titanium dioxide nanoparticles as an emerging pollutant: A review. *Desalin Water Treat* 2024;317:100025. [\[Crossref\]](#)
- [39] Jain A, Ambekar A, Thajudeen T. Effect of Titania Nano-additives on Fine and Ultrafine Carbonaceous Emissions during Flame Combustion of Diesel. *Aerosol Air Qual Res* 2024;24. [\[Crossref\]](#)
- [40] Yamin J, Hdaib II, Eh Sheet EA, Abu Mushref AJ. RSM analysis of heat balance of direct injection 4-stroke diesel engine using biodiesel fuel. *Biofuels* 2021;12:777–787. [\[Crossref\]](#)
- [41] Chinmaya M, Anuj P, Singh TV, Naveen K. Combustion, Emission and Performance Characteristics of a Light Duty Diesel Engine Fuelled with Methanol Diesel Blends. *Int J Mech Mechatronics Eng* 2013;7:40–50.
- [42] Jayaraman J, Alagu K, Venu H, Appavu P, Joy N, Jayaram P, Mariadhas A. Enzymatic production of rice bran biodiesel and testing of its diesel blends in a four-stroke CI engine. *Energy Sources Part A Recover Util Environ Eff* 2023;45:5340–5351. [\[Crossref\]](#)
- [43] Hotti SR, Hebbal OD. Biodiesel production and fuel properties from non-edible champaca (*Michelia champaca*) seed oil for use in diesel engine. *J Therm Eng* 2015;1:330–336. [\[Crossref\]](#)

- [44] Kayode B, Hart A. An overview of transesterification methods for producing biodiesel from waste vegetable oils. *Biofuels* 2019;10:419–437. [\[Crossref\]](#)
- [45] Bohlouli A, Mahdavian L. Catalysts used in biodiesel production: a review. *Biofuels*. 2021;12:885–898. [\[Crossref\]](#)
- [46] Ghanati SG, Doğan B, Yeşilyurt MK. The effects of the usage of silicon dioxide (SiO_2) and titanium dioxide (TiO_2) as nano-sized fuel additives on the engine characteristics in diesel engines: a review. *Biofuels* 2024;15:229–243. [\[Crossref\]](#)
- [47] Drews A. Standard Test Method for Kinematic Viscosity of Transparent and Opaque Liquids (the Calculation of Dynamic Viscosity). In *Manual on Hydrocarbon Analysis*. 6th ed. Pennsylvania, USA: ASTM Manuals; 2008. p. 126–128. [\[Crossref\]](#)
- [48] Tomar M, Kumar N. Influence of nanoadditives on the performance and emission characteristics of a CI engine fuelled with diesel, biodiesel, and blends—a review. *Energy Sources Part A Recover Util Environ Eff* 2020;42:2944–2961. [\[Crossref\]](#)
- [49] Drews A. Standard Practice for Density, Relative Density (Specific Gravity), or API Gravity of Crude Petroleum and Liquid Petroleum Products by Hydrometer Method. In *Manual on Hydrocarbon Analysis*. 6th ed. Pennsylvania, USA: ASTM Manuals 2008. p. 252–255. [\[Crossref\]](#)



Research Article

On the role of biomimetics shark skin flow control in the aerodynamic characteristics of leading-edge protuberanced wing section

S. SMRITHIKA¹, S. ARUNVINTHAN^{1,*}

¹Department of Aerospace Engineering, SASTRA Deemed University, Thanjavur, Tamil Nadu, 600001, India

ARTICLE INFO

Article history

Received: 28 May 2017

Accepted: 08 August 2017

Keywords:

Aerodynamic Force Coefficients;
Biomimetics; Flow Control;
Leading-Edge Protuberances;
Surface Pressure Distribution;
Time-Series Data; Wind Tunnel
Testing

ABSTRACT

An experimental investigation of the effect of shark skin flow control structures on the aerodynamic characteristics of novel bio-inspired Leading-Edge Protuberanced wing section is presented in this paper. NACA 63(4)-021 airfoil based leading-edge protuberanced wing featuring an Amplitude of 0.12c and wavelength 0.5c is utilized in this study. Short-fin Mako's scale structure as outlined in the previous literature were 3D Printed and they act as the base geometry of the shark scale structure. Two different sets of shark scale geometries, varying in chord length, span, amplitude, and wavelength were chosen based on the literature. Additionally, to assess the significance of the alignment of these 3D Printed shark scale structures on the aerodynamic characteristics, different patterns like Staggered non-overlapped, linear non-overlapped, linear overlapped were considered. This study merges two biomimetic ideas such as leading-edge protuberances and shark scales which renders aerodynamic benefits. The focus of this study is to assess the influence of the shark scale structures as an effective flow control means for biomimetic Leading-Edge Protuberanced Wing section. Further, all the works related to shark scale were only carried out over conventional straight wing airfoil sections. The present study is the first of its kind to investigate the influence of shark scale structures on the aerodynamic characteristics of Leading-Edge Protuberanced wings. All the series of experiments were conducted at wide range of angles of attack ranging from $0^\circ \leq \alpha \leq 70^\circ$ in an increment of 5° at two different Reynolds number 32066 and 69488. Surface pressure measurements were obtained over the test models with the help of Miniature Pressure Scanner 4264 Scanivalve pressure scanner pneumatically connected to the 50 pressure taps equi-distributed over the upper and the lower surface of the test model. Results reveal that the use of shark scale structures as a means of flow control renders aerodynamic benefit in terms of lift increment, drag reduction and stall delay. The modified LEP model with Shark scale structures exhibit 34.6% increment in lift coefficient, 29% decrement in drag coefficient and 42.8% stall delay. The modified shark scale model fitted Leading-Edge Protuberanced wing is effective and outperforms conventional Leading-Edge Protuberanced wing especially at low Reynolds number and hence could be used as a viable solution for Micro-Aerial Vehicles and Nano-Aerial Vehicles operating in this regime.

Cite this article as: Smrithika S, Arunvinthan S. On the role of biomimetics shark skin flow control in the aerodynamic characteristics of leading-edge protuberanced wing section. J Ther Eng 2025;11(5):1293–1311.

*Corresponding author.

*E-mail address: smrithika613@gmail.com, sarunvinthan@gmail.com

This paper was recommended for publication in revised form by
Regional Editor Ahmed Selim Dalkilic



INTRODUCTION

Airfoils invention dates back to 19th century and over the years it has become omnipresent. Airfoils have now emerged as a crucial component of several man-made Engineering structures like propellers, marine rudders, wind turbine blades, high-lift devices of airplanes, control surfaces of missiles etc., as illustrated in Fig. 1. Despite the application field, it remains common that the flow over the airfoil must remain attached to the surface as long as possible even at greater angles of attack for increased performance in the airfoil. Flow separation is one of the major issues which plagues the aerodynamic performance of the airfoils. To mitigate this risk and reduce its influence, Flow control techniques have emerged and several researchers and engineers adopted several flow control strategies over the years. Therefore, it becomes clear that the flow control technology has emerged as a promising solution to conquer the terrible aerodynamic performance triggered by the boundary layer separation on airfoil's suction surface as airfoils are ubiquitous [1]. The flow separation over the airfoils can be controlled by three methodologies namely active, passive and hybrid. As we know that the when both the active and passive flow control techniques are incorporated together, they give rise to a hybrid flow control technique. Based on the literature, it is evident that the passive flow control techniques are popular as they require minimum energy intervention as they don't require any active energy source. In addition, because of its intrinsic simplicity accompanied with economy, it is one of the most viable solutions as a flow control means. Recently, researchers have started focusing their attention on optimizing the Unmanned Aerial Vehicles (UAV), Micro-Aerial Vehicles (MAV) and Nano-Aerial Vehicles (NAV). Micro Aerial

Vehicles and Nano Aerial Vehicles are recently utilized as the First-Person View Drones in the real-world combat scenarios. Due to its small size and operating Reynolds number (Re), without an effective flow control method, the MAV/NAVs face aerodynamic challenges which can lead to unstable and unpredictable airflow. An effective flow control means like shark scale structures will help stabilize the airflow around the vehicle, improving overall stability and control thus rendering more accurate control of lift, drag and other aerodynamic forces. In this way, it can reduce the power required for the flight, thus increasing the endurance i.e. Extended flight time / Range translating to longer flight times. FPV operating in closed building will experience turbulence mitigation and increased agility allowing for quick response to changes in the environment which is vital. As flow control effectively reduces the acoustics signature, it can also lead to quieter operation. Given the prevalence of airfoils in aircraft, UAVs, MAVs, and NAVs, wings, control surfaces etc., it is thought to be essential to predict their aerodynamic performance under various wind characteristics to withstand and lessen the adverse consequences.

With the advancements in the technology, biomimetics can be considered as an option to effectively control the flow over the airfoils. One such intellectual way is to have a look into shark's denticular shape as a form of passive vortex generator over the airfoil. Shark skin has dermal denticles (scales) that impede body separation by providing local flow separation control, thus helping the shark to swim faster in water with least drag. The shark scale is replicated as vortex generator in the previous research of Arunvinthan et al. [2]. Based on the results reported by Arunvinthan et al. [2], the use of Shark scale-based vortex generators (SSVGs) has improved the coefficient of lift by 3.8%, contributing to

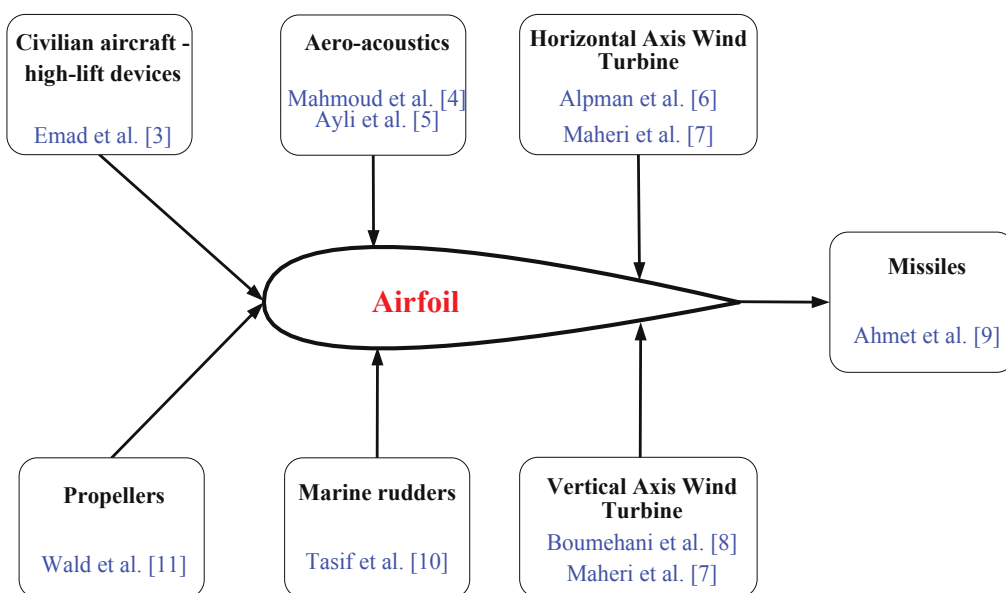


Figure 1. Few applications of airfoil [Created by authors].

drag reduction and delays the stall by regulating the flow separation. It was discovered that bristled denticles on the shark skin act as a vortex generator which is a passive flow control mechanism.

Lang et al. [12] investigated the microgeometry of the shark scale and reported that the cavity like structure formed over the denticles plays a vital role in suppressing the wake formation. Lang et al. [12] investigated the complex 3d embedded cavity model and compared its result with the flat base (i.e. square cavity) model and revealed that the shark skin structure forms a series of interlocking web of vorticity thus promoting drag reduction. Bechert et al. [13] claimed that the shark scale structures behave in a similar way as riblets based on the SEM analysis. Bechert et al. further continued his research [13] by replicating 800 plastic scales with compliant anchoring over a flat plate resembling shark scales and tested whether they can be used as a viable alternative for vortex generators to suppress the flow separation. From the results, it has been identified that the vortices created along their axis in the streamwise direction by shark scales induces momentum in to the flow and promotes exchange of momentum thus leading to an increased flow velocity near the wall. Motta et al. [14], identified that the shark skin denticles are composed of tooth-like layers of enameloid and dentine with a central pulp cavity, and have an expanded base that anchors individual denticles into the skin as shown in Fig. 2. Utilizing similar morphology as found by Motta et al. [14], Li Wen et al. [15] designed and manufactured flexible biomimetic shark skin models using 3D Printing. Li Wen et al. [15] focused his attention on the effect of shark scale pattern over the flat plate in water tunnel. Aiming at identifying the influence of the patterns on the drag characteristics, three different alignment patterns were utilized. Based on the research findings, Li Wen et al. [15] reported that the staggered-overlapped pattern offers exceptional swimming speed with reduced drag at low speeds when compared against the other 2 alignment patterns. This clearly shows that the

pattern of shark scale also plays an important role in determining the drag characteristics. However, to quantify that, Bechert et al. [13], established a numerical relation using a dimensionless parameter S^+ to represent the effective Re based on the spacing between the denticle top ridges. S^+ can be mathematically represented as shown in Eqn. (1) where, S is the spacing between the adjacent denticle ridges, ρ the fluid density, ν the kinematic viscosity and τ_w being the average shear stress.

$$S^+ = \frac{S}{\nu} \sqrt{\frac{\tau_w}{\rho}} \quad (1)$$

The spacing between the adjacent denticle ridges (s) can be seen from the zoomed-in view of the sagittal section as shown in the Fig. 2. With the advent of latest digital manufacturing techniques, researchers adopted the methodology of 3D printing shark scale structures to mimic them and obtain the aerodynamic benefit over airfoils. Some of the recent literatures like Arunvinthan et al. [2], Li wen et al. [15-16], Yuji Yasuda et al. [17], Wen Chien et al. [18] etc., has utilized 3D printed shark scales in their research. Li wen et al. [15-16] have mimicked the real-time shark scale and 3D printed them to experimentally investigate the hydrodynamic characteristics of the shark skin membrane. The shark scales were scaled up around 12.4 times due to the technical limitations involving degradation of shark scale denticles when printed at real-time shark scale dimensions are associated with 3D printing technology. Yuji Yasuda et al. [17] mentions about the challenges in manufacturing due to scaling and cost. Furthermore, Wen Chien et al. [18] indicated that the shark scale structures manufactured using 3D printing lack surface hydrophobicity and could potentially create biological settlement. To overcome which, their report suggested anti fouling coating using enzymes (which includes a combined chemical effect) which makes the 3D Printing a lot difficult when we make it in microscopic level. Despite several researchers focusing their attention on such 3D Printed

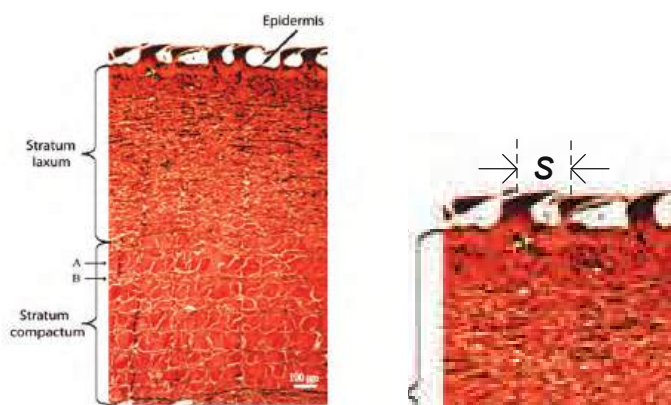


Figure 2. Sagittal section through the skin of a female shortfin mako shark *Isurus oxyrinchus* and its zoomed-in view [Bechert et al. [14], with permission from John Wiley].

shark scale structures, it is worth noting that the least sizing which a 3D printer can print is of no match to the actual real-world shark scale which is one of the primary challenges involved. Another challenge involved in designing and manufacturing the shark scale is because of its complicated design geometry. Domel et al. [19] morphologically investigated the shark skin and identified that shark skin is covered with rigid bony denticles that with ridges which narrows to a thin neck that anchors into the skin and contribute to drag reduction effect and claims that the shark scales play a significant role in the lift enhancement providing high lift to drag ratio. Their research results revealed that the boundary layer formed near the shark scale helps in generating a short separation bubble which in turn creates a low-pressure region (i.e., suction) over the upper surface of the airfoil causing the flow to reattach thus improvising the lift and rendering aerodynamic benefits. In addition, the streamwise vortices generated by the shark scales or denticles prevent loss of lift at high angles of attack and increment in drag. Similarly, C. Lee et al. [20] numerically investigated about the rectangular sharklet patterned membrane under two flow conditions which confirmed that the sharklet pattern is favourable in flow re-attachment. Following which Fagla J. Mawignon et al. [21] investigated riblets orientation and the arrangement patterns to optimize the results. To characterize the alignment of the riblets in both the laminar and turbulent flow two patterns

namely aligned and staggered pattern were evaluated. The investigation reveals that aligned and staggered arrangements of the simplified shark scale structures reduce the drag effectively about 11.3% in laminar state and 6% in turbulent state. From the results, it becomes clear that the shark scale structure effectively alters the flow characteristics based on their sizing and alignment pattern. Shuai Li et al. [22] also investigated the drag reduction mechanism of bionic skin made of scale structures and reported that the bionic skin generates vortex structure which favours the drag reduction by flow re-attachment. However, to identify any influence of the rigidity of the scale, Dengke Chen et al. [23-25] have experimentally investigated the effect of bionic gradient flexible fish skin on the drag characteristics and revealed that it also reduces the drag effectively. Furthermore, D. Chen et al. [23-25] carried forward the research to evaluate the effect of multi-scale and multi-level riblet on the drag characteristics by considering various fish skin like shark skin, dolphin skin, tuna skin etc. The multi-scale and multi-level riblet which was inspired from the shark skin has been fabricated and the results conclude that multi-scale and multi-level riblet surface provides a maximum drag reduction ratio of 6.35% at the Reynolds number in the order of 10^5 . Dengke Chen et al. [23-25] also has identified and reported intricacies like fabrication of shark skin due to the complex topography of the shark skin in their literature. Recent literatures claim that the

Table 1. Literature review and its relation with the experimental findings

Author	Experimental / Numerical	Description	Characteristics	Relation with the findings
Bechert et al. [13]	Experiment	Vortices by the shark scales infuse momentum in to the flow resulting in drag reduction.	Hydrodynamics	Drag reduction
Lang et al. [12]	Experiment	Shark scale structures promote drag reduction by suppressing the wake formation.	Hydrodynamics	Drag reduction
Li Wen et al. [15-16]	Experiment	staggered-overlapped pattern offers high speed with reduced drag.	Hydrodynamics	Drag reduction
Domel et al. [19]	Experiment	Shark scale generates separation bubble causing the flow to reattach, thus improvising the lift coefficient	Aerodynamics	Lift increment
Arunvinthan et al. [2]	Experiment	SSVGs improved lift coefficient by 3.8%, contributes to drag reduction and stall delay	Aerodynamics	Lift increment, drag reduction and stall delay
Lee et al. [20]	Numerical	Sharklet patterns promote flow-reattachment	Hydrodynamics	Extended flow attachment
Mawignon et al. [21]	Numerical	Staggered outperforms the aligned pattern of riblet surface.	Hydrodynamics	Drag reduction.
Shuai Li et al. [22]	Numerical	Delays the flow separation and reduces drag.	Hydrodynamics	Extended flow attachment and Drag reduction.
Dengke Chen et al. [23-25]	Experiment	Multi-level riblets provide maximum drag reduction ratio	Hydrodynamics	Drag reduction.

shark-skin inspired thin film attached to the airplane exhibits a reduced drag and reduction in fuel consumption which necessitates the study to be developed further in the domain of space and aviation.

The present study aims at investigating the effectiveness of shark scale structures as a flow control method over airfoils by experimentally evaluating two different shark scale structures under three different alignment patterns for a wide range of angle of attack ranging from $0^\circ \leq \alpha \leq 70^\circ$ in an increment of 5° at two different $Re = 32066$ and 69488 . This will act as a pilot study for administering shark scale structures as an effective flow control method for airfoils. Most of the previous literature explores the feasibility of shark scale structures as an effective means of drag reduction methodology to augment the hydrodynamic characteristics. Even though the use of shark scale in hydrodynamics has a strong foundation, limited number of studies were conducted in aerodynamics. To the extent of author(s) knowledge, the knowledge of shark scale structures on the aerodynamic characteristics remains unclear, and is yet to be explored. It is worth noting that this domain is less explored because of the challenges associated with the manufacturing (3D printing) of complicated shark-scale geometries, fixing them over aerofoils and in obtaining real-world Reynolds number scenarios. Further, all the works related to shark scale were only carried out over conventional straight wing airfoil sections. The present study is the first of its kind to investigate the influence of shark scale structures on the aerodynamic characteristics of Leading-Edge Protuberanced wings (inspired from the flippers of the

Humpback whales featuring varying chord along the span). Additionally, the present study acts as a bridge between two successful biological inspirations from nature. Therefore, the present study will help aerodynamic designers in identifying the optimum denticle pattern, size and shaping at low Reynolds number for the possible application over UAV/MAV and NAV with immediate effect. Table 1 represents the relation between the literature and experimental findings of the present study.

EXPERIMENTAL METHODOLOGY

The effect of shark-scale structure and its alignment on the aerodynamic characteristics of NACA 63(4)-021 airfoil based biomimetic leading-edge Protuberanced wing was experimentally evaluated in this paper at $Re=32066$ and 69488 .

Wind tunnel: All the experimental evaluations carried out in this study were conducted at a low-speed subsonic wind tunnel facility located at SASTRA Deemed University. This wind tunnel encompasses a rectangular test section with cross section of $300 \times 300 \times 1500$ mm. The tunnel is operated by a fan powered by a 10HP motor and may attain a maximum wind speed of around 60 m/s. The free stream turbulence intensity is 0.51%. A schematic representation of wind tunnel facility with necessary equipment set up is shown in Fig. 3 and the real images of Low-speed subsonic wind tunnel facility is shown in the Fig. 4.

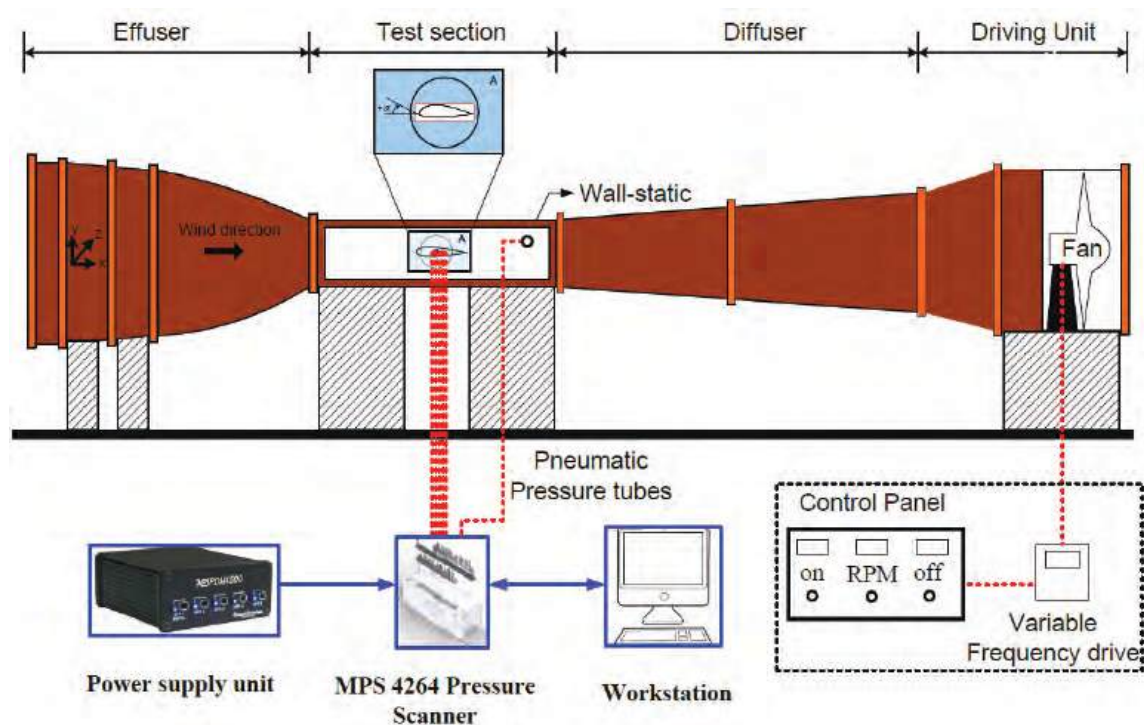


Figure 3. Wind tunnel facility equipped with MPS 4264 Scanivalve pressure scanner.



Figure 3. Low-speed subsonic wind tunnel facility

Table 1
Geometrical parameter of the test model.

A/c	λ/c	Label
1	1	Baseline
0.025	0.5	Small LEP
0.1	0.25	Medium LEP
0.12	0.5	High LEP

Test model: NACA 63(4)-021 airfoil model was chosen as the base model as it closely resembles the cross section of the humpback whale flipper which acts as a bio-inspiration for the LEP wing. Since the present study aims at investigating the effect of shark-scale surface and its alignment pattern on the aerodynamic characteristics of NACA 63(4)-021 airfoil based biomimetic leading-edge Protuberanced wing, the baseline model is a LEP Wing. Based on the framework of the previous researchers [26], it becomes clear that the LEP wing features a sinusoidal leading-edge profile influenced by two geometrical parameters named Amplitude (A) and wavelength (λ). It should however be noted that as the present study focusses its attention on investigating the effect of shark scale surface and its alignment pattern alone, the amplitude and the wavelength of the LEP is kept constant throughout this paper.

Based on their experimental evaluation, it has been identified that the $A=0.12c$ and $\lambda=0.5c$ renders optimum aerodynamic benefit and outperforms the other two test cases. Therefore, the same $A=0.12c$ and $\lambda=0.5c$ has been chosen as the baseline model for this present study. The airfoil considered in this study is assumed to have a mean chord length of 100 mm and span 300 mm which signifies that the test model is an infinite one. The full-span model was then fabricated by 3d printing using polylactic acid (PLA) material at the resolution of 100 microns. There are special recommendations for distributing the surface pressure taps over the LEP airfoil. As the test model is a non-constant chord model, the surface pressure distribution needs to be measured at two different locations namely peak (chord maxima) and the trough (chord minima) which is one of the special requirements of the LEP wing. A total of 50 pressure tapings were made over the

surface of the model. 20 pressure taps were equi-distributed along the peak and 16 taps over the trough region while the remaining taps were distributed along the span. The diameter of each pressure tapping is approximately 1mm with the spacing between each pressure tapings as 9 mm [26]. The airfoil model considered in the present study is the same considered in the previous study by Arunvinthan et al. [26]. Henceforth, the results are validated against the previous research paper [26] as shown in the Fig. 5 and the results reveal that the validation holds in good agreement with the paper with a minimum error of 0.68% and maximum deviation of 12%. The difference in the values can be owing to the increase in the number of pressure ports on the present study from 36 to 50. Furthermore, since the present study is the pilot study in incorporating 3D Printed shark scales over the LEP airfoil, shark scale affixed structure cannot be validated. The dimensions of the test model and the schematics of the baseline LEP model and modified LEP model is represented in the Fig. 6 & 7.

Shark-Scale Surface: Based on the framework of the previous researchers, it has been reported that the morphometric measurements of shortfin mako scales were found to be of height 0.03 mm, spacing $\sim 0.04 - 0.08$ mm, chord length of 0.169-0.214 mm, span length of 0.14 – 0.244 mm as digitized by Motta et al. [15].

As we know that the denticle shapes and size are highly dynamic and is often subjected to change based on the location of the shark scale on the body, type of the fish etc., it becomes important to assess the denticles of different shapes and sizes. However, as the present study tries to mimic the short fin mako, the denticle shape becomes constant. Therefore, in this study, 2 shark-scale test models featuring

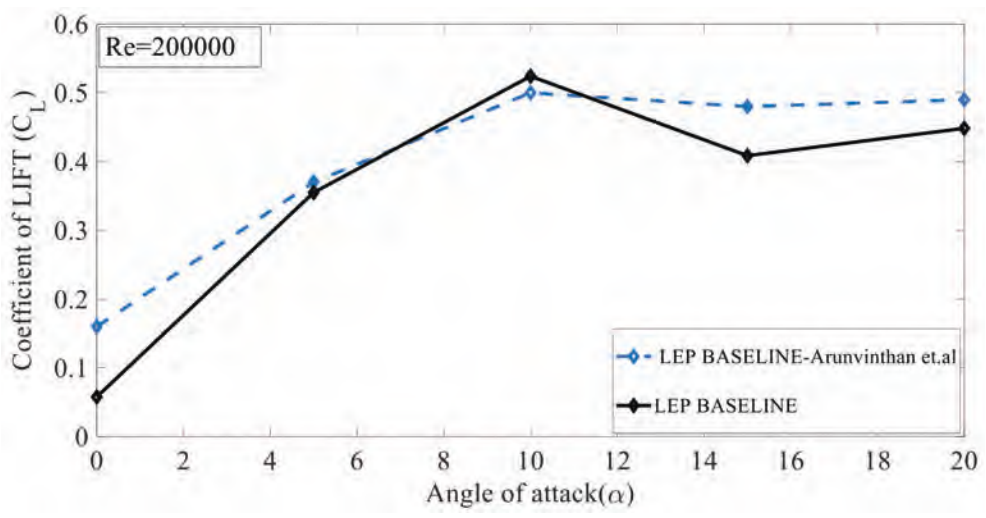


Figure 5. Validation of LEP baseline with previous literature.

Table 2. Geometric dimensions of 3D printed shark scales

Attributes	Small Shark Scale	Large Shark Scale
Chord length	10 mm	20 mm
Span	6 mm	12 mm
Leading-Edge Amplitude	2 mm	3 mm

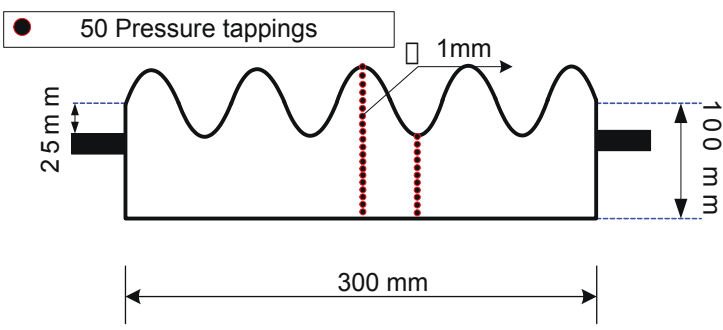


Figure 6. Dimensions of the test model.

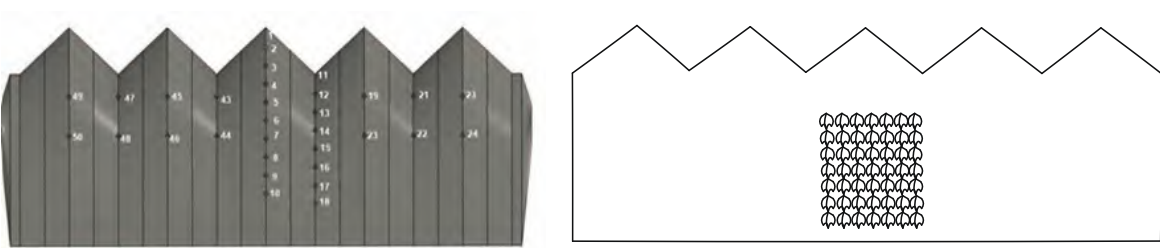


Figure 7. Schematics of baseline LEP model and modified LEP model.

chord length of 10 mm, span 6 mm with leading edge amplitude of 2 mm (hereafter named as small shark scale) and the other model has a chord length of 20 mm, span 12 mm with

leading edge amplitude of 3 mm (hereafter named as large shark scale) is considered. The geometric dimensions of the 3D printed shark scales are represented in the Table 2. One

could immediately note the difference in the size existing between the original morphometric measurements and the test model considered. This drastic difference in the sizing could possibly be explained by the limitations of the currently available 3D printers. In other words, due to the limitations in 3D printing resolution, the model was scaled up

to the above stated dimensions. Yuji Yasuda et al. [17], Wen Chien et al. [18] etc. confirmed this limitation of sizing in using 3D printing in their corresponding research papers. In addition to this limitation, Yoseph Bar-Cohen et al. [27] emphasizes that nature's designs are optimized for specific environments and conditions that may not match the needs

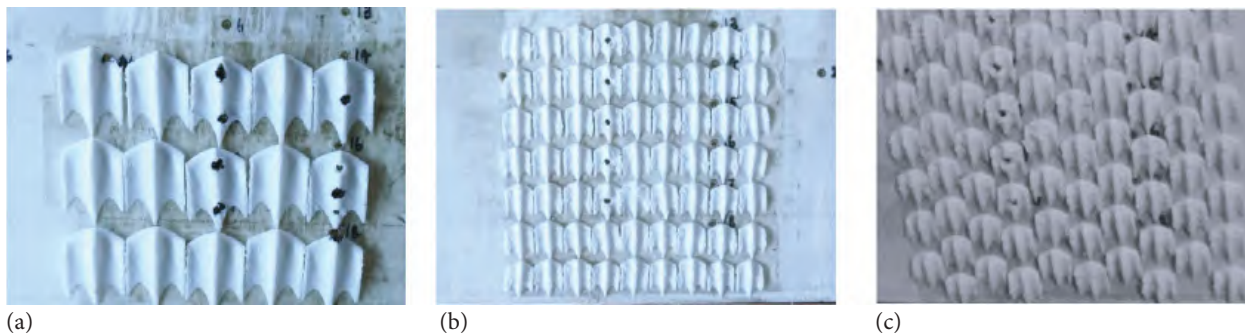


Figure 8. (a) large shark scale structure arranged in linear non-overlapped pattern (b) small shark scale structure arranged in linear overlapped pattern (c) small shark scale arranged in staggered non-overlapped pattern.

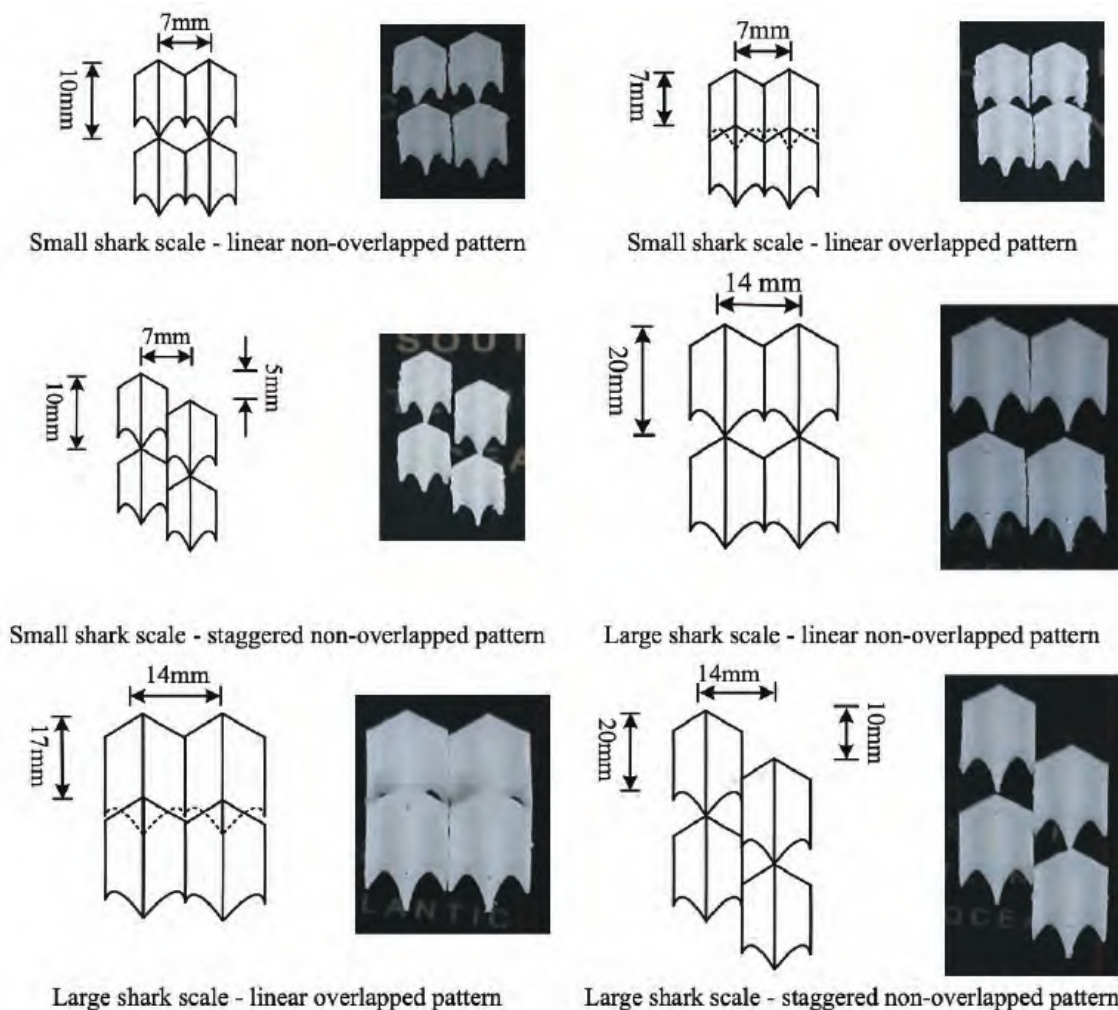


Figure 9. Schematics of alignment patterns for both large and shark scales.

of human applications. However, by making modifications and adapting these natural designs, it is possible to achieve functional and efficient solutions for technological challenges. This concept is central to biomimetics, where engineers and scientists take inspiration from nature's designs but modify them to fit specific human needs or constraints. Further, Li wen et al. [15] reported that although there is a great variety of denticle shapes and size, its intricate flow mechanism greatly depends on its complex patterns of alignment. To assess the significance of such complex alignment pattern in addition to the denticle size, three different patterns namely Staggered non-overlapped, linear non-overlapped, linear overlapped were inspired from the previous studies. A schematic representation is shown in Fig. 8. In this study, shark scale structures of different sizing were 3D printed using PLA material at a resolution of 100 microns per mm with ultra-marble finish. The 3D printed shark scale structures of different sizes were then blended over the leading-edge protuberanced wing with the help of commercially available non-residue double sided tissue tape having a thickness of 60 μ . Schematics of alignment patterns for both large and shark scales are shown in Fig. 9.

Simultaneous pressure scanner: The surface pressure of the model is measured through the pressure taps on the surface which is pneumatically connected to MPS4264 Scanivalve pressure scanner. It has a range of 10" of H₂O water column with $\pm 0.06\%$ Full-scale, long-term accuracy as per the calibration certificate provided by the original equipment manufacturer (OEM). The time-series data obtained from the Scanivalve pressure scanner were collected at a sampling frequency of 700Hz corresponding to 10000 data samples. The aerodynamic lift and drag force acting over the biomimetic LEP Wing with and without shark scale structures were then estimated using pressure integration technique [26, 28-31] to yield the coefficient of lift and coefficient of drag.

$$F_D = \sum (\Delta P) \times S_i \times \cos(\alpha + \frac{\pi}{180}\theta) \quad (2)$$

$$F_L = \sum (\Delta P) \times S_i \times \sin(\alpha + \frac{\pi}{180}\theta) \quad (3)$$

$$C_D = F_D / 0.5\rho v^2 S \quad (4)$$

$$C_L = F_L / 0.5\rho v^2 S \quad (5)$$

Generally, the uncertainties in the experiments are due to the dispersion of the data or by the measuring equipment and instruments involved. Uncertainties associated with the wind tunnel like Buoyancy, solid blockage, wake blockage etc. and the instrumental uncertainties are discussed as follows.

Buoyancy correction: It is well known fact that boundary layer thickens as the flow travels downstream towards the exit cone thereby reduction in jet area and thus, inducing static pressure variation without the presence of test models. This could give rise to additional drag force and hence a suitable correction needs to be implemented on the experimental data. Glauert's method of buoyancy correction was implemented in the present study. According to Glauert, the total drag increment due to the buoyancy for a two-dimensional infinite model is:

$$D_B = \frac{1}{2} \pi \lambda_2 t^2 P' \quad (6)$$

where λ_2 is the body shape factor, t is the body thickness and P' is the slope of longitudinal static pressure gradient curve. The value for λ_2 for NACA 63(4)-021 airfoil used in this study is 1.4 as outlined in Barlow et al. [32]. The drag due to buoyancy was estimated as 0.009693 for this paper.

Solid blockage correction: The test-section walls reduce the area through which air flows when compared to free-stream conditions. Therefore, the velocity of the air increases as it squeezes through the reduced area in the vicinity of the model. This blockage is a function of model thickness, model distribution and the model size. This paper utilizes Thom's method of solid blockage correction for two dimensional tunnels:

$$\varepsilon_{sb} = \frac{K_1 (Model Volume)}{C^{3/2}} \quad (7)$$

where K_1 equals 0.74 for wing spanning the tunnel breadth and C is test-section area. The model volume can be approximated as follows:

$$Model Volume = 0.7 \times model thickness \times model chord \times model span \quad (8)$$

The model volume was estimated to be 0.441 and the blockage correction was estimated as 0.01208 for this paper.

Wake blockage correction: The two-dimensional wake blockage correction provided by Allen and Vincentti was utilized in this paper. The wake blockage correction factor for two-dimensional case is:

$$\Delta C_{d,wb} = \Lambda \sigma \quad (9)$$

where Λ is body shape factor equals 0.36 for NACA 63(4)-021 airfoil as stated in Barlow et al. [32]. σ can be estimated from the formula

$$\sigma = \left(\frac{\pi^2}{48} \right) \left(\frac{c}{h} \right)^2 \quad (10)$$

and estimated as 0.0616225. The overall wake blockage correction was estimated to be 0.0221841 which is already implemented in the present manuscript.

Table 3. Pressure scanner accuracy

Full scale ranges	Accuracy	Over pressure capability	Operating Temperature
1 psid	0.06%FS	15x	0° -70° C

Table 4. Experimental dimensions table

Component	Dimension(s)	Component	Dimension(s)
Test section Length	1500 mm	Space between pressure taps	9 mm
Test section Width	300mm	Angles of attack	0°-70°
Test section Height	300 mm	Small scale chord length	10 mm
Maximum wind speed	60 m/s	Small scale span	6 mm
Mean chord length	100 mm	Small scale leading-edge amplitude	2 mm
Large scale leading-edge amplitude	3 mm	Large scale chord length	20 mm
Pressure taps diameter	1 mm	Large scale span	12 mm
Span length	300 mm		

Instrument and Data Error: The MPS4264 miniature Scanivalve pressure scanner utilized in the pressure measurement was precise with a full-scale error of $\pm 0.06\%$, the correction affected only the third decimal of the uncorrected value. The accuracy of the pressure scanner is shown in the Table 3. The time-series data acquired from the MPS4264 Scanivalve pressure scanner shows a stationary data which indicates that the outliers or extreme values which can disproportionately affect the mean are absent. Also, since the process is automated using MATLAB, mistakes made during manual data entry, such as typographical errors capable of skewing the data is eliminated. Use of Scanivalve proved mettle in terms of calibration errors and drift in values. In short, the author(s) strongly believe that the data errors are kept minimum.

The drag due to Buoyancy is estimated as 0.009693 using Glauert's method; Solid Blockage is 0.01208 based on Thom's rule; Wake blockage correction is estimated as 0.0221841 utilizing Allen & Vincenti correction factor; Full-scale error of Instrument is $\pm 0.06\%$ based on the OEM Manual; Tolerance of 3D printer is ± 0.1 mm. The detailed insights of the uncertainty measurements have been elaborated in the previous study [31-32]. Dimensions of all the components utilized in this study has been tabulated as shown in Table 4.

RESULTS AND DISCUSSION

The shark scale model affixed over the baseline LEP model was primarily examined to identify the influence of size and patterns on the aerodynamic characteristics of the LEP Wing at two different $Re=32066$ and 69488 and the corresponding results were presented and discussed in this section. The results were compared against the baseline

LEP model to gain more insight about the effect of size and pattern of the shark scale structures on the biomimetic LEP Wing.

a) $Re=32066$:

The coefficient of lift for the clean baseline LEP wing and the modified airfoils affixed with two different shark scale geometries (i.e., small, and large shark scale structure) organized in three distinct alignment patterns operating at $Re = 32066$ is shown in Fig. 10. It can be observed from the figure that the unmodified model as well as modified model affixed with shark scale structures does not follow the conventional lift characteristics curve. It is speculated that this might be plausible due to the Reynolds number regime which is not conventional as well. Generally, it is known from the previous literatures that the clean baseline LEP wing tends to exhibit two-step stall characteristics. However, on the contrary, the clean baseline LEP wing did not exhibit two-step stall characteristics and follows an unconventional trendline of lift characteristics curve. Since the Re is very low, it is believed that the flow does not possess enough energy to follow the curvature prevailing at the nose region of the airfoil itself at $\alpha=0^\circ$. Furthermore, with the further increase in the angles of attack making it more difficult for the flow to remain attached over the airfoil might give an insight for this unconventional lift characteristics curve. Despite producing relatively lower lift coefficients, the modified models affixed with shark scales produce higher lift when compared against the baseline LEP wing. Based on the results, it has been identified that the incorporation of shark scale structures on the airfoil tends to transition the flow in to the turbulent regime thereby energizing the flow resulting in better flow characteristics. For instance, the lift coefficient for the 3D Printed

small shark scale model with linear non-overlapped alignment pattern exhibits two-step stall characteristics with the primary stall occurring at $\alpha = 30^\circ$ and secondary stall happening at $\alpha = 50^\circ$. The two step stall characteristics which is reported in the LEP wing prevails with the addition of the shark scale structure. This clearly shows that the addition of shark scale structures on the LEP airfoils tends to augment its lift characteristics at the Re considered in this study. Likewise, a similar trendline of C_L is also seen in the small shark scale model with linear overlapped alignment pattern with primary stall occurring at $\alpha = 15^\circ$ and secondary stall happening at $\alpha = 50^\circ$ having a maximum C_L of 0.22 at $\alpha = 50^\circ$.

Aiming at identifying the influence of staggered pattern, the small shark scale structure with staggered non-overlapped alignment pattern were also experimentally evaluated and the results show that the primary stall occurs at $\alpha = 10^\circ$ and the secondary stall happening at $\alpha = 40^\circ$. Following which, to gain some more insight on the small shark scale structures, further analysis were carried out. It has been observed from the results that the presence of small shark scale structures over the LEP test model irrespective of the alignment pattern energizes the flow over the airfoil. This helps the modified model with shark scale structures exhibit delay in stall characteristics. For instance, it can be observed that a stall delay of 10° could be seen for modified small shark scale structure affixed LEP model in staggered non-overlapped alignment pattern than the linear overlapped alignment pattern. Based on these results, it can be concluded that the usage of the shark scale surface on the airfoil renders aerodynamic benefit in terms of stall delay. At the same time, it is worth noting that an abnormal trendline of C_L is seen for the large scale arranged in staggered non-overlapped pattern. This pattern produces a maximum lift coefficient of 1.2 at $\alpha = 15^\circ$. Beyond which with the further increase in the angle of attack, the lift coefficient gradually decreases. The lift coefficient for large scale

arranged in linear overlapped pattern performs poorer than its counterparts. It is speculated that when these large shark scale structures are linearly overlapped, the first set of shark scales are aligned with zero degree of incidence to the flow. However, with the subsequent overlapping, from the second line of shark scale structures to the last line of shark scale structures, they experience a significant alteration in the incidence angle owing to overlapping. Therefore, the vortices induced from the first line of scale can be easily disturbed and trapped by the second line of shark scale structures inclined at an angle. Studies suggests that it resembles the bristling angle of the shark scale with one difference that in a conventional fish they are non-overlapped, but in this case, it gets overlapped which results in the detrimental result. As the momentum is not fully imparted in to the flow and gets trapped inside the incidence angle of the subsequent shark scale structures the net pressure gradient reduces. This gives the negative lift coefficient in comparison against the lift increment provided by other models. To further ascertain this behavior, it can be observed that, when the alignment pattern is changed from linear overlapped to staggered non overlapped, the energized flow arising out of shark scale structures moves over the surface of the airfoil without much hindrance, thus enabling better aerodynamic performance showing peak lift coefficient. Based on the results, it has been decided to eliminate the discussions on the linear overlapped pattern in further section because of its adverse nature.

The effect of shark scale on the drag characteristics is shown in the fig. 11. It is worth noting that the mean coefficient of drag plotted here represents only the pressure drag which is obtained from the airfoil surface pressure measurement and does not involve skin friction drag. Further, it can be inferred from the figure that the drag coefficient increases gradually till stall angle. Following that, a rapid increase in the C_D can be seen. The high turbulent mixing

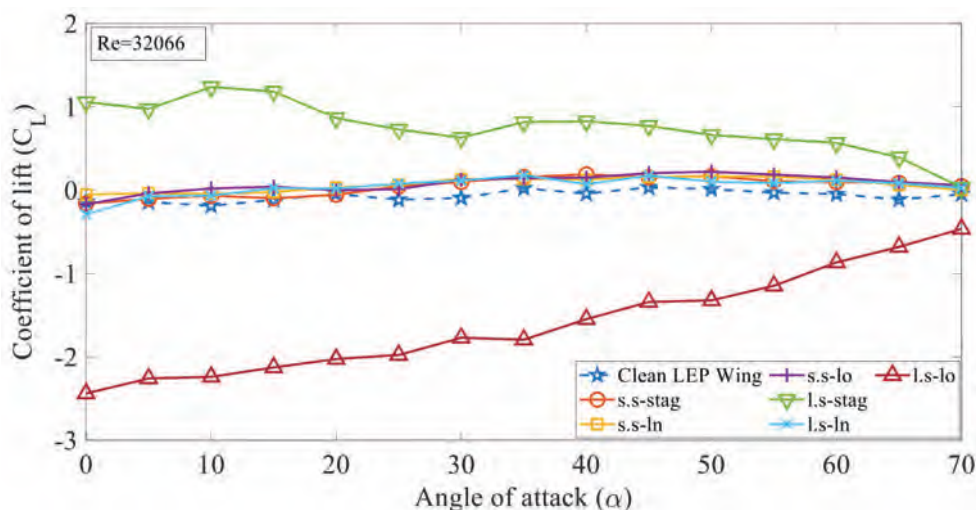


Figure 10. Coefficient of lift (C_L) vs Angle of attack (α).

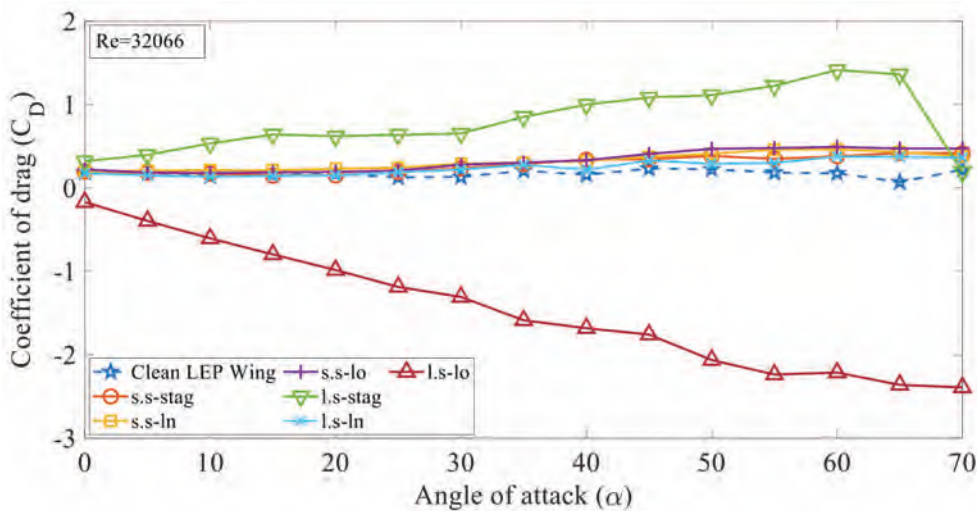


Figure 11. Coefficient of drag (C_D) vs Angle of attack (α).

in the boundary layer by large shark scale with staggered non-overlapped alignment pattern might be a possible explanation for the unusual increment in the drag coefficient. Despite the increased drag, lift by drag ratio or aerodynamic efficiency is high and hence, aerodynamic performance is not compromised for the same. The baseline LEP wing has a very much lower drag when compared to the shark scale models at post-stall angles. However, at the pre-stall angles, it can be inferred from the Fig. 11 that the large shark scale models arranged in linear non-overlapped alignment pattern exhibits the lowest drag. Figure 12 demonstrates the comparison of lift coefficient over baseline and shark scale models. From Fig. 12 it is observed as the small shark scale structure reduces the flow acceleration over the airfoil in the pre-stall angles, it causes a reduction in the overall net pressure difference between the suction and the pressure side of the model thus resulting in lesser lift coefficient. However, at the same time, during post-stall angles, the same small shark scale structure enhances the turbulent mixing of the separated shear layers thus rendering the shark scale model advantageous. It can be easily seen from the Fig. 12 that beyond $\alpha=20^\circ$, LEP model with small shark scale structures exhibit higher lift coefficient in comparison against the baseline LEP model without shark scale structures. Evidently, large shark scale structures arranged in the staggered non-overlapped pattern exhibits a higher lift in comparison with other models even at greater angles of attack. The negative lift coefficient produced by model incorporating large shark scale structures arranged in linearly overlapped configuration can be seen clearly from the Fig. 12(a). This confirms the poor aerodynamic behaviour of the large shark scale structures arranged in linearly overlapped configuration as stated earlier. Figure 13 represents the comparison of drag coefficient over the baseline and shark scale models. Apparently, a higher-pressure drag is

produced by the large shark scale structures arranged in the staggered non-overlapped pattern than other models which might be caused due to perturbation over the airfoil model. It can be concluded from the observations that the turbulence mixing induced by the shark scale structures creates perturbation over the airfoil thus increasing the pressure drag resulting in a higher drag coefficient than its counterpart unmodified equivalent. Experimental results revealed that each shark scale size and pattern have its own unique effects on aerodynamics of the airfoil in terms of maximum lift coefficient, minimum drag coefficient, stall delay characteristics etc. Hence, understanding the fundamental flow physics is therefore seen to be essential.

b) $Re=69488$:

The time-averaged lift coefficient as a function of angle of attack (α) for small and large shark scale geometries aligned in three different patterns are shown in Fig. 14. It is evident from the graphs that there is significant difference in the coefficient of lift between the three patterns model of both small and large shark scale structures and baseline LEP wing. This clearly shows that with the change in the alignment pattern, the surface flow shifts appreciably over the modified LEP airfoil. It can be inferred from the figure that the (C_L) of baseline LEP wing increases from $\alpha = 0^\circ$ to 45° with a maximum C_L of 0.8650 at $\alpha=45^\circ$. Beyond which with the further increase in the angle of attack, stall phenomenon is initiated.

Aiming at identifying the aerodynamic lift for the modified models, initially, it was observed that the trendline of small shark scale structure aligned in staggered non-overlapped pattern, the C_L for 3D printed shark scale model increases as α increases to 40° , with $(C_L)_{\max} = 0.8541$. However, the lift coefficient for the same at pre-stall angles, say 15° , is 33.2% higher than its unmodified counterpart. Secondly, the C_L for small scale structure arranged in

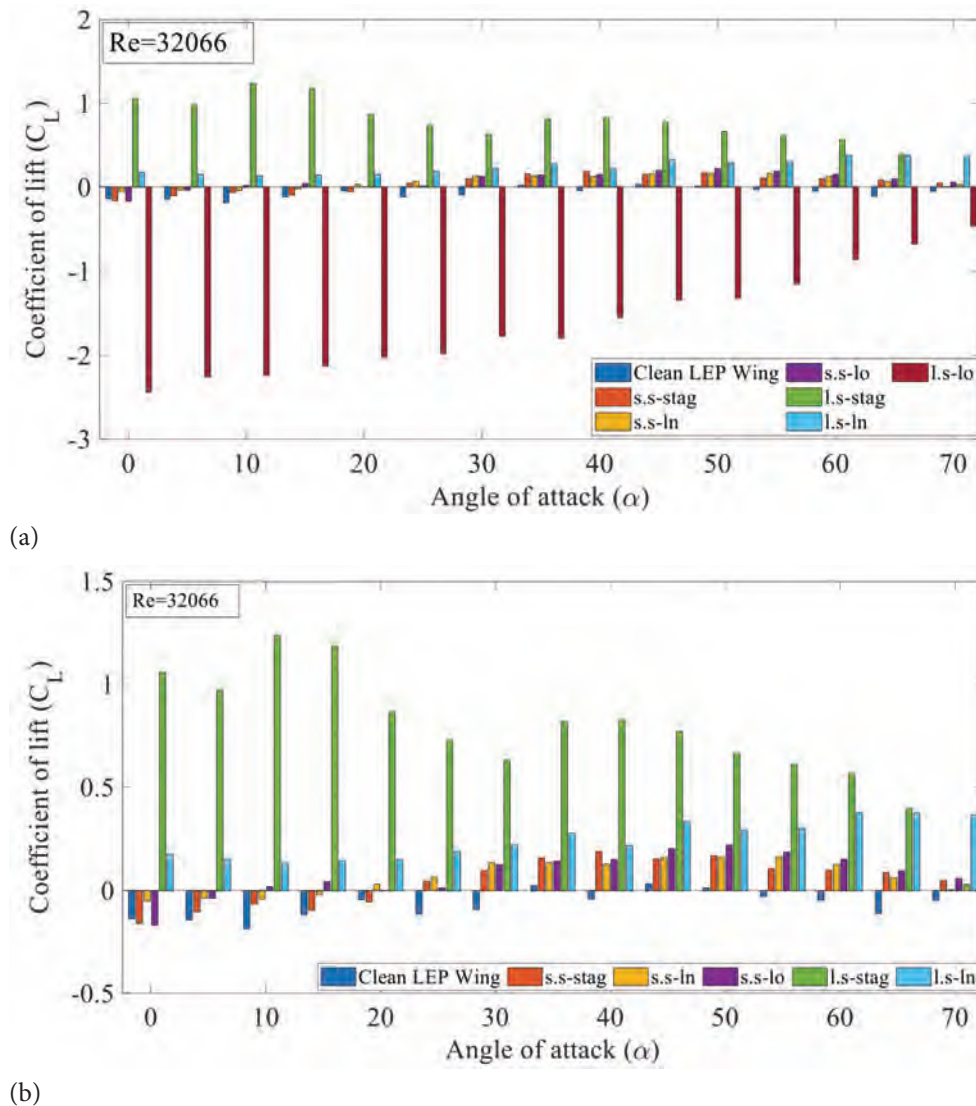


Figure 12. (a) Comparison of the coefficient of lift (C_L) over baseline and shark scale models (b) Comparison of the coefficient of lift (C_L) over baseline and shark scale models (better resolution)

linear non-overlapped structure exhibited a maximum C_L of 0.8823 which stalls at 50° offering significant stall delay. Subsequently, the lift coefficient for small scale structure arranged in linear overlapped structure increases till $\alpha = 20^\circ$ and exhibits a small dip in the lift coefficient. It regains lift with a maximum lift coefficient of 0.8567 at $\alpha = 45^\circ$. Upon examining the lift coefficient curve for large shark scale structure aligned in staggered non-overlapped pattern, the C_L increases till 45° angle of attack with a maximum lift coefficient of 0.9425. The lift coefficient for large shark scale structure aligned in staggered non-overlapped pattern at post-stall regime, say 55° , has improved by 23.8% when compared to baseline LEP wing. The C_L for large shark scale structure aligned in linear non-overlapped pattern, represents increment as a function of an angle of attack in the increasing direction of angles of attack from $0^\circ < \alpha < 40^\circ$

with a maximum lift coefficient of 0.8598 at $\alpha = 40^\circ$. It is also observed that all the shark scale models produce negative lift at initial angles of attack. A possible explanation to this observation could be, at $\alpha = 0^\circ$ the baseline LEP wing section itself have attached flow. Moreover, the incorporation of the shark scales over the LEP wing section could acts as an obstruction leading to accumulation of pressure over suction side and hence, negative lift. From the above observations, it can be reported that the staggered non-overlapped pattern for small shark scale structure exhibit an enhanced aerodynamic performance in terms of lift coefficient in the pre-stall regime while staggered non-overlapped pattern for small shark scale structure enhances the performance in post-stall regime. Therefore, based on the results, it can be claimed that whatsoever pattern and size of shark scale structures, incorporation of the same plays

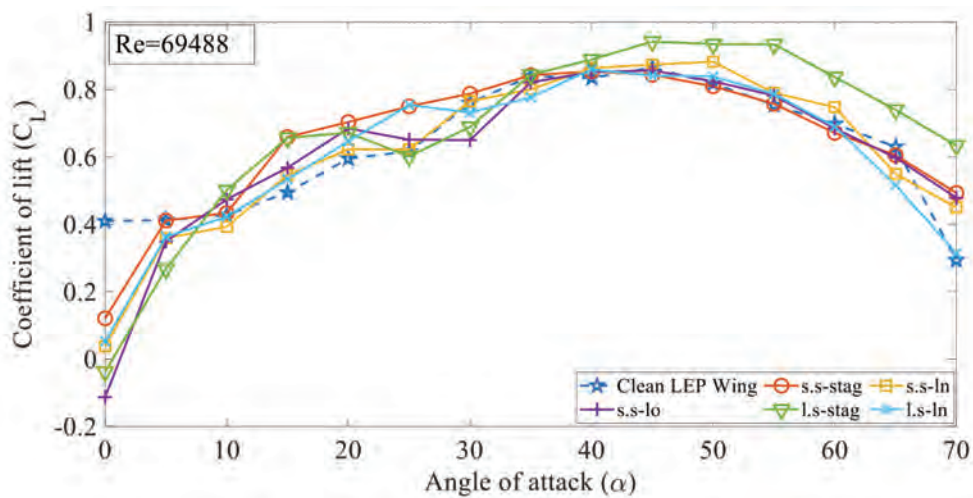


Figure 13. Coefficient of lift (C_L) vs Angle of attack (α).

a significant role in altering flow characteristics and renders better aerodynamic performance than its unmodified equivalent. Fig. 15 illustrates the variation of drag coefficient with angle of attack for all the test models. It can be noted from the figure that the C_D increases with the increase in angle of attack. Aiming at identifying the aerodynamic drag coefficient trendline for different arrangement of shark scale models, the drag coefficient for the large shark scale structures aligned in staggered non-overlapped pattern is lesser in the pre-stall angles than the baseline LEP wing. The modified airfoil with large shark scale structure staggered maintains a thinner, more energized boundary layer, keeping the flow attached over a greater portion of the surface. It is speculated that the staggered pattern infuses more momentum in to the flow thereby resulting in a smoother pressure distribution and reduced wake size, leading to a

lower overall drag coefficient at pre-stalling angles. While, the C_D trendline for large shark scale structure arranged in linear non-overlapped pattern was similar to the trendline of baseline LEP wing. But, drag coefficient is lesser at pre-stalling angles in comparison with the baseline model. It can be inferred as the difference in the net pressure between leading-edge and trailing edge is less at pre-stall angles in the model integrated with shark scale structure arranged in linear non-overlapped configuration. The drag coefficient for small shark scale arranged in linear overlapped pattern is slightly higher when compared to baseline LEP wing. The higher drag might be associated with the trapping induced by the incidence angle of the overlapping shark scale structure which enhances the turbulence that eventually leads to increased drag. Overall, it is believed that the disturbance induced by the intensity of vortices generated from the

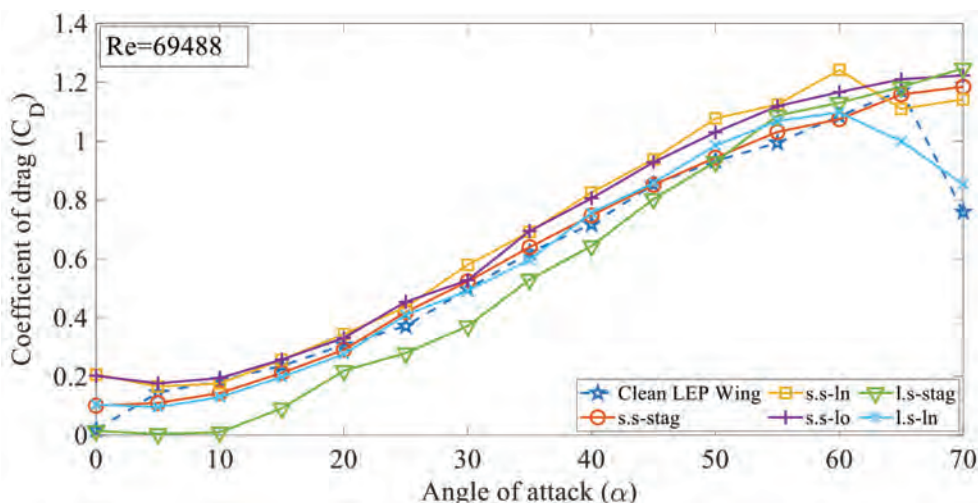


Figure 14. Coefficient of drag (C_D) vs Angle of attack (α).

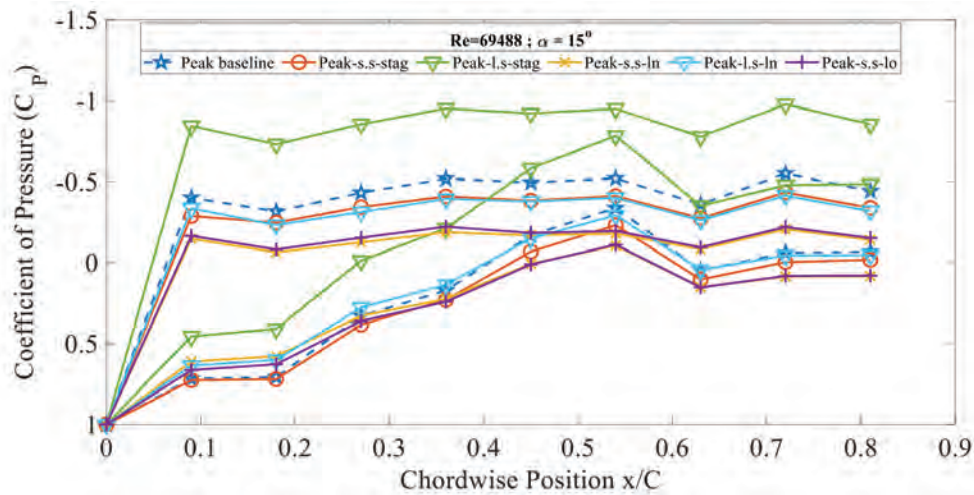


Figure 15. Pressure coefficient vs x/c at $Re = 69488$ at $\alpha = 15^\circ$ (peak region).

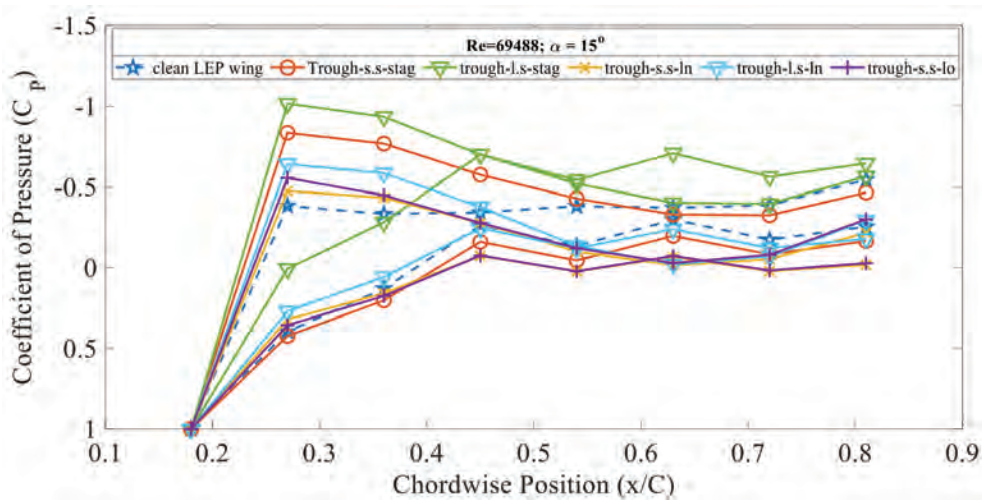


Figure 16. Pressure coefficient vs x/c at $Re = 69488$ at $\alpha = 15^\circ$ (trough region).

shark scale structures might be the reason for the increase in overall drag coefficient.

Figure 16 & 17 depicts the coefficient of pressure (C_p) vs. chordwise position (x/C) for peak and trough section of all the test models at 15° angle of attack. It is evident that pressure distribution of the clean baseline LEP wing exhibits less negative pressure on the upper or suction surface of peak region till 40% of the chordwise location. This indicates the attached flow over the first 40% of the chordwise location and eventually the flow gets separated over the remaining chordwise location. On the other hand, the pressure distribution for other models except large shark scale aligned in staggered non-overlapped pattern exhibit a negative suction pressure in the vicinity of leading-edge which can be inferred from the Fig. 16. On investigating the large shark scale aligned in staggered non-overlapped pattern, the

suction pressure decreases till 54% of the chordwise location indicating more attached flow over the peak section. Thus, it can be understood that the flow over the modified model incorporating large shark scales aligned in staggered non-overlapped pattern is accelerated effectively indicating delayed flow separation. It is worth noting that the pressure difference between the upper and lower surface of the modified test cases is higher than the baseline LEP wing representing higher lift coefficient. This can be confirmed from the Fig. 14 and can be plausibly explained as the shark scales infuses energy into the boundary layer to the effectively accelerate flow at $\alpha = 15^\circ$ in the peak section of the model. Further investigating the pressure distribution graph, it can be observed from the Fig. 16 that a low negative suction pressure is seen in the leading-edge region varying from -0.2 to -0.4 for all the models except for large shark scale

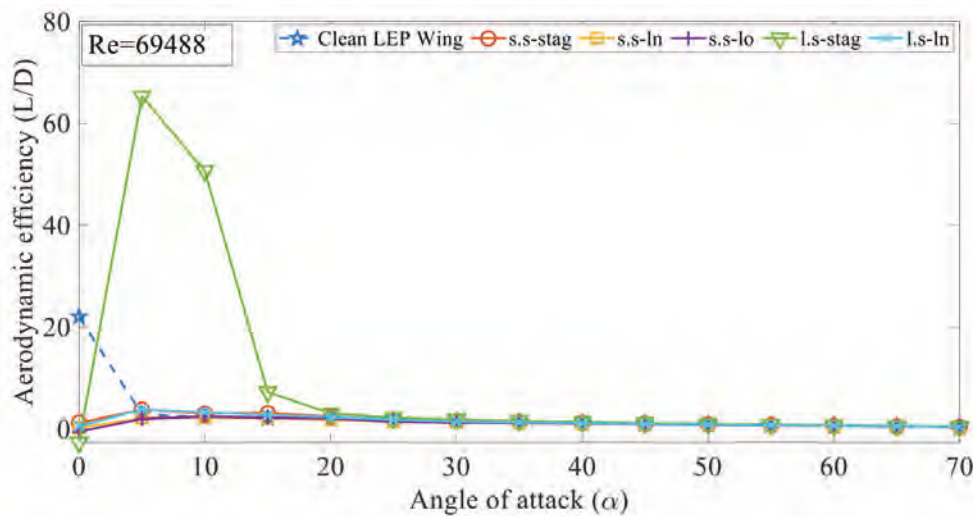


Figure 17. Aerodynamic efficiency (L/D) Vs angle of attack (α).

Table 5. Comparison of Results

Label	Parameter	Baseline	Modified	Benefits
s.s-stag	Lift Coefficient	0.49	0.66	34.6% increment in the C_L
l.s-stag	Drag Coefficient	0.31	0.22	29% decrement in the C_D
s.s-ln	Stall	35°	50°	Stall delay by 42.8%

model aligned in staggered non-overlapped pattern. In the case of large shark scale model arranged in staggered non-overlapped pattern, negative suction pressure is maximum than all the other test cases and present over the first 40%-50% of the chordwise location. It is observed that pressure varies linearly but fails to reach ambient pressure at the trailing edge vicinity. The pressure difference between the rear and front of the airfoil increases along the chordwise distance when flow over the airfoil separates, resulting in increasing in pressure drag. On examining Fig.17, it is identified that the pressure distribution in the trough section for the modified large shark scale model aligned in staggered non-overlapped pattern has a higher suction pressure which indicates the flow is accelerated over the trough section of the airfoil. From these observations, it becomes clear that the modified shark scale models effectively delay the flow separation in addition with increased lift.

Figure 17 represents the aerodynamic efficiency as a function of angle of attack. From the figure, it becomes clear that the efficiency of the modified model affixed with shark scale structure is relatively higher than the unmodified equivalent especially in the pre-stall regime. Though the modified models enhance the performance, linearly overlapped pattern of large shark scale structure outperforms the conventional LEP baseline model by injecting momentum from the free-stream into the boundary layer

effectively which improves the aerodynamic efficiency. Moreover, the large shark-scale structure arranged in staggered non-overlapped pattern renders even more high performance than the linearly overlapped shark scale pattern. The linear overlapped pattern tends to behave like a riblet inducing momentum in to the flow along the line, whereas in the case of the staggered non-overlapped pattern, the vortices induced from the first line of shark scale gets more infusion of momentum due to the formation of the turbulent boundary layer. The small shark-scale structures also exhibit better performance than the baseline model. However, the amount of momentum induced by such small-scale structures are not sufficient enough to alter the flow characteristics favourably resulting in lesser performance characteristics than the large shark scale structures. Therefore, based on the results, it can be concluded that the staggered pattern of the large shark-scale structures offers more aerodynamic benefits and delay the flow separation more effectively. The summary of the comparison of results are tabulated in the Table 5.

CONCLUSION

A series of wind tunnel test were performed to investigate the effect of two different sized shark scale arranged in three different configurations over the test airfoil at multiple

Reynolds number. Based on the experimental evaluation the following conclusions were made as follows:

- Knowledge on the effect of shark scale flow control structures over the aerodynamic characteristics of a biologically inspired leading-edge protuberanced test model were highlighted and presented.
- Modified airfoil models fitted with shark scale structures performs significantly better than its unmodified equivalent by suitably altering the flow behavior over the airfoil.
- In terms of lift coefficient at pre-stall angles, the modified airfoil model exhibits better performance characteristics. Out of different shark scale patterns and two different Reynolds number tested it has been identified that the staggered non-overlapped pattern exhibits a peak lift increment of about 33.2% at $\alpha = 15^\circ$, when compared against the baseline Leading-Edge Protuberanced model.
- At post-stall angles, the shark scale structures featuring staggered non-overlapped pattern exhibits an enhanced aerodynamic lift performance with the maximum increment of 23.8% than the baseline model.
- At low Reynolds number conditions, the modified airfoil models outperform the conventional Leading-Edge Protuberanced model, making it as a better choice for applications, such as micro- and nano-aerial vehicles, shark scales are preferred.
- Incorporation of linearly non-overlapped pattern of small shark scale structured airfoil eliminates the two step stall characteristics behaviour present in the conventional Leading-Edge Protuberanced models and offers the aerodynamic benefit in terms of stall delay by stalling at $\alpha=50^\circ$.
- If drag reduction is of primary concern, then large-scale staggered shark scale structured airfoil can be employed for a peak drag reduction of about 28.7% in pre-stall angles at $\alpha=20^\circ$.
- Results reveal that the baseline Leading-Edge Protuberanced airfoils performs apparently poorer at the low-Reynolds number tested in this experiment. Moreover, with the addition of the shark scale structures it renders the aerodynamic benefit. Comparing the overall results, it has been observed that the large-scale staggered shark scale structure exhibits a peak 65.31% of aerodynamic efficiency increment than the baseline Leading-Edge Protuberanced model.

The experimental results proved that different pattern and sizes have unique features in altering the aerodynamics of flow. Since, the modified shark scale model is effective at low Reynolds number and at low angle of attack, it can be applied to Lighter Than Air technologies like airships where drastic change in angles of attack will not be very frequently utilized, Micro-Aerial Vehicles and Nano-Aerial Vehicles which fly at low Reynolds numbers. Since the 3D Printed shark scale geometries were pasted on the Leading-Edge Protuberanced wing section, the wind tunnel testing

at high Reynolds number were carefully ignored to avoid dislodging of such 3D Printed structures which could be resolved in the future study with some other means. It is worth noting that the shark scale structures can be 3D printed directly on the wing model itself to avoid dislodging at higher Reynolds numbers, but at the price of more cost per model. Bristling angle and different patterns and sizes at higher Reynolds number testing along with the attempts to decode the underlying flow physics will be made in the near future to potentially utilize this technology in real-time.

NOMENCLATURE

LEP	Leading-edge protuberance
GAMBIT	Geometry And Meshing Built-In Tool
MAV	Micro-Aerial Vehicle
NAV	Nano-Aerial Vehicle
UAV	Unmanned-Aerial Vehicle
VG	Vortex Generator
SSVG	Shark Scale Vortex Generator
SEM	Scanning Electron Microscopy
λ	Wavelength
A	Amplitude
C	Chord length
PLA	Polylactic acid
F_D	Drag force
F_L	Lift force
ΔP	Net Pressure
S+	Effective Re based on shark scale spacing
S_i	Area
α	Angle of attack
θ	Angle of incidence on the i^{th} port
C_D	Coefficient of drag
C_L	Coefficient of lift
C_P	Coefficient of pressure
X/C	Chordwise location
Re	Reynolds Number
s.s-stag	Small shark scale - staggered non-overlapped pattern
s.s-ln	Small shark scale - linear non-overlapped pattern
s.s-lo	Small shark scale - linearly overlapped pattern
l.s-stag	Large shark scale - staggered non-overlapped pattern
l.s-ln	Large shark scale - linear non-overlapped pattern
l.s-lo	Large shark scale - linearly overlapped pattern

ACKNOWLEDGEMENTS

The authors would like to thank SASTRA University for allowing us to carry out the wind tunnel experiments.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

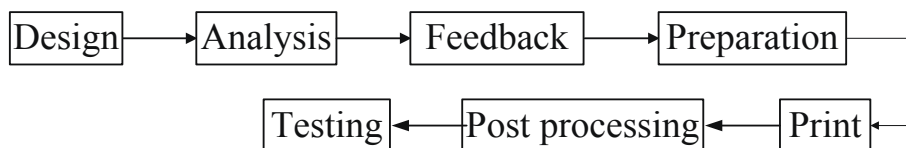
REFERENCES

- [1] Choi H, Park H, Sagong W, Lee SI. Biomimetic flow control based on morphological features of living creatures. *Phys Fluids* 2012;24:121302. [\[CrossRef\]](#)
- [2] Arunvinthan S, Raatan VS, Nadaraja Pillai S, Pasha AA, Rahman MM, Juhany KA. Aerodynamic characteristics of shark scale-based vortex generators upon symmetrical airfoil. *Energies* 2021;14:1808. [\[CrossRef\]](#)
- [3] Hussein E, Azziz H, Rashid F. Aerodynamic study of slotted flap for NACA 24012 airfoil by dynamic mesh techniques and visualization flow. *J Therm Eng* 2021;7:230–9. [\[CrossRef\]](#)
- [4] Mahmoud H. Stability of turbine blades, aircraft wings and their acoustic radiation. *J Therm Eng* 2015;1:6. [\[CrossRef\]](#)
- [5] Ayli E, Koçak E, Türkoğlu H. Numerical investigation of rod-airfoil configuration aeroacoustic characteristics using Ffowcs-Williams-Hawkings equations. *J Therm Eng* 2021;7:58–70. [\[CrossRef\]](#)
- [6] Alpman E. Aerodynamic performance of small-scale horizontal axis wind turbines under two different extreme wind conditions. *J Therm Eng* 2015;1:420–32. [\[CrossRef\]](#)
- [7] Maheri A. Simulation of wind turbines utilising smart blades. *J Therm Eng* 2016;2:557–65. [\[CrossRef\]](#)
- [8] Boumehani A, Noura B, Kerfah R, Khelladi S, Dobrev I. Numerical investigation of the blade profile effect on the aerodynamic performance of a vertical-axis wind turbine Darrieus H-rotor. *J Therm Eng* 2020;6:388–402. [\[CrossRef\]](#)
- [9] Şumnu A, Güzelbey İ. The effects of different wing configurations on missile aerodynamics. *J Therm Eng* 2023;9:1260–71. [\[CrossRef\]](#)
- [10] Tasif TH, Rahman MH, Fazle AB, Karim MM. Numerical prediction of flow past a marine rudder. *Procedia Eng* 2017;194:59–66. [\[CrossRef\]](#)
- [11] Zanolli A, Menini L, Savino A, Grassi D, Riccobene L. Experimental investigation of wing-propeller aerodynamic interaction in eVTOL configurations. *Aerosp Sci Technol* 2024;152:109348. [\[CrossRef\]](#)
- [12] Lang A, Hidalgo P. Cavity flow characterization of the bristled shark skin microgeometry. In: 47th AIAA Aerospace Sciences Meeting including the New Horizons Forum and Aerospace Exposition. 2009;1107. [\[CrossRef\]](#)
- [13] Bechert DW, Bruse M, Hage W. Experiments with three-dimensional riblets as an idealized model of shark skin. *Exp Fluids* 2000;28:403–12. [\[CrossRef\]](#)
- [14] Motta P, Habegger ML, Lang A, Hueter R, Davis J. Scale morphology and flexibility in the short-fin mako *Isurus oxyrinchus* and the blacktip shark *Carcharhinus limbatus*. *J Morphol* 2012;273:1096–110. [\[CrossRef\]](#)
- [15] Wen L, Weaver JC, Thornycroft PJ, Lauder GV. Hydrodynamic function of biomimetic shark skin: Effect of denticle pattern and spacing. *Bioinspir Biomim* 2015;10:066010. [\[CrossRef\]](#)
- [16] Wen L, Weaver JC, Lauder GV. Biomimetic shark skin: Design, fabrication and hydrodynamic function. *J Exp Biol* 2014;217:1656–66. [\[CrossRef\]](#)
- [17] Yasuda Y, Zhang K, Sasaki O, Tomita M, Rival D, Galipon J. Manufacturing of biomimetic silicone rubber films for experimental fluid mechanics: 3D printed shark skin molds. *J Electrochem Soc* 2019;166:B3302–8. [\[CrossRef\]](#)
- [18] Chien HW, Chen XY, Tsai WP, Lee M. Inhibition of biofilm formation by rough shark skin-patterned surfaces. *Colloids Surf B Biointerfaces* 2020;186:110738. [\[CrossRef\]](#)
- [19] Domel AG, Saadat M, Weaver JC, Haj-Hariri H, Bertoldi K, Lauder GV. Shark skin-inspired designs that improve aerodynamic performance. *J R Soc Interface* 2018;15:20170828. [\[CrossRef\]](#)
- [20] Lee C, Lee GW, Choi W, Yoo CH, Chun B, Lee JS, et al. Pattern flow dynamics over rectangular Sharklet patterned membrane surfaces. *Appl Surf Sci* 2020;514:145961. [\[CrossRef\]](#)
- [21] Mawignon FJ, Liu J, Qin L, Kouediatouka AN, Ma Z, Lv B, et al. The optimization of biomimetic shark-skin riblet for the adaptation of drag reduction. *Ocean Eng* 2023;275:114135. [\[CrossRef\]](#)
- [22] Li S, Liu S, Zhao D, Dong L, Jiao H. Drag reduction characteristics of the placoid scale array skin supported by micro Stewart mechanism based on penalty immersed boundary method. *Appl Ocean Res* 2024;149:104049. [\[CrossRef\]](#)
- [23] Chen D, Cui X, Liu X, Chen H. Bionic gradient flexible fish skin acts as a passive dynamic micro-roughness to drag reduction. *Surf Coat Technol* 2023;457:129337. [\[CrossRef\]](#)
- [24] Chen D, Liu X, Cui X, Zhang L, Chen H. Research progress and development trend of the drag reduction inspired by fish skin. *Prog Org Coatings* 2023;182:107613. [\[CrossRef\]](#)
- [25] Chen D, Li W, Zhao Y, Liu J, Cui X, Zhao Z, et al. Drag reduction capacity of multi-scale and

- multi-level riblet in turbulent flow. *Biosurf Biotribol* 2024;10:7–15. [\[CrossRef\]](#)
- [26] Arunvinthan S, Pillai SN, Cao S. Aerodynamic characteristics of variously modified leading-edge protuberanced (LEP) wind turbine blades under various turbulent intensities. *J Wind Eng Ind Aerodyn* 2020;202:104188. [\[CrossRef\]](#)
- [27] Bar-Cohen Y. *Biomimetics: Biologically inspired technologies*. Boca Raton (FL): CRC Press; 2005. [\[CrossRef\]](#)
- [28] Anderson JD Jr. *Fundamentals of aerodynamics*. New York: McGraw-Hill; 2016.
- [29] Li QA, Kamada Y, Maeda T, Murata J, Nishida Y. Effect of turbulent inflows on airfoil performance for a horizontal axis wind turbine at low Reynolds numbers (Part I: Static pressure measurement). *Energy* 2016;111:701–12. [\[CrossRef\]](#)
- [30] Arunvinthan S, Pillai SN. Aerodynamic characteristics of unsymmetrical aerofoil at various turbulence intensities. *Chin J Aeronaut* 2019;32:2395–407. [\[CrossRef\]](#)
- [31] Arunvinthan S, Gouri P, Divysha S, Devadharshini RK, Nithya Sree R. Effect of trough incidence angle on the aerodynamic characteristics of a biomimetic leading-edge protuberanced (LEP) wing at various turbulence intensities. *Biomimetics* 2024;9:354. [\[CrossRef\]](#)
- [32] Barlow JB, Rae WH, Pope A. *Low-speed wind tunnel testing*. New York: John Wiley & Sons; 1999.

APPENDIX

The shark scales and baseline models were 3D printed using Neptune 4 Plus FDM 3D printer with the tolerance of ± 0.1 mm. The primary design of base model and shark scale models were made using gambit which was exported as .iges file. The same has been converted to .stl file and analysed the models in Klipper software which was provided along with the OEM of 3D printer to test for the wall thickness parameter and location of pressure taps etc. The feasibility check was then run through the klipper software. Following which, the models were then 3D printed using Neptune 4 Plus FDM 3D printer using PLA Material. The 3D printed models were then post-processed using a series of abrasive sheets to obtain ultra-marble finish. Later, the specimens were blown with a compressed air to remove any mechanical burrs etc before testing.



Basic flow chart depicting the 3D printing process



Research Article

Highly viable nanoparticle based Thermal Interface Materials (TIM) for electronics device cooling applications

S Uma MAHESHWARI^{1,*}, A. Brusly SOLOMON², G. THILAGAVATHI³,
Madhukar HEMAMALINI⁴, D. ILLAKKIAM⁵

¹Department of Physics, Mother Teresa Women's University, Kodaikanal, 624101, India

²Micro and Nano Heat Transfer Lab, Centre for research in material science and thermal management, Division of Mechanical Engineering, Karunya Institute of Technology and Sciences, Karunya Nagar, Coimbatore, 641001, India

³Dr. M.G.R Govt. Arts and Science College for Women, Villupuram, 605401, India

⁴Department of Chemistry, Mother Teresa Women's University, Kodaikanal, 624101, India

⁵Department of Biotechnology, Mother Teresa Women's University, Kodaikanal, 624101, India

ARTICLE INFO

Article history

Received: 10 August 2024

Accepted: 13 September 2024

Keywords:

Bond Line Thickness; Effective Thermal Contact Resistance; Heat Transfer Applications; Nanoparticles; Thermal Interface Material

ABSTRACT

High performance interface materials (TIMs) were developed using various nanoparticles and paraffin wax. An indigenous test set up was fabricated using aluminium heat sink and copper plates as heating plate for measuring effective thermal contact resistance ($R_{th, eff}$) of prepared TIMs to evaluate their thermal performance. Chrome-Alumel thermocouples were used for measuring the temperature. Furthermore a relative study of effective thermal contact resistance of prepared (TIMs) was carried out to evaluate their performance. The nanoparticle chosen for present distinctive study were SiO_2 , Al_2O_3 , CuO, GO, rGO. The TIM samples were synthesized by blending 6 wt% of assorted nanoparticles in 5 ml base fluid of paraffin wax. They were uniformly mixed using ultra sonicator to create a smooth and fine paste based TIM. The resulting paste (TIMs) was applied as an ultra-thin layer between the copper heater and aluminium sink of the indigenously designed and fabricated test rig. Power inputs for experiments were 25, 50 and 75W. Experimental studies were done at reduced pressure (RP) and full pressure (FP) applied by completely loosening and tightening the screws of the aluminium heat sink. Addition of (SiO_2 , Al_2O_3 , CuO, GO and rGO) nanoparticles to the base fluid paraffin wax significantly decreases the effective thermal resistance to a large extent as the added nano particles helps in better heat conduction due to their enhanced total surface area leading to more effective heat evacuation. Enhanced results are observed at full pressure, at reduced pressure drastic increment in ($R_{th, eff}$) occurs due to improper gap filling. These results significantly depicts the importance of bondline thickness in performance of TIM. Interestingly silica based TIM gives the best performance of heat transference to ambience at 50 W power input ($R_{th, eff} = 0.46^\circ C/W$) and slightly becomes inferior to GO at higher power input. GO based TIM shows the best results at higher power of 75 W, its $R_{th, eff}$ being $0.51^\circ C/W$.

Cite this article as: Maheshwari SU, Solomon AB, Thilagavathi G, Hemamalini M, Illakkiam D. Highly viable nanoparticle based Thermal Interface Materials (TIM) for electronics device cooling applications. J Ther Eng 2025;11(5):1312–1326.

*Corresponding author.

*E-mail address: phduma13@gmail.com

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkılıç



INTRODUCTION

With the advent of low dimensional physics, new generation of electronic products are in nano dimensions packed with more power and enhanced performance in lesser packaged space, hence the greater necessity of thermal management within the overall products design becomes important. It is already known that when two rough surfaces meet, uneven interfaces are formed. These uneven surfaces are not suave and flat which causes diminished contact area. Now the actual interface only has distorted micro-contacts. The size of the impact region depends on different factors, including the roughness of the structures, the internal properties and the impact pressure. Therefore, elastic, flatness of solids and plastic properties and physical strain regulate the distribution of impact spots. As result of this micro and macro gaps and voids that are sealed up by the materials, such as air, are created. Technically the conventional heat transfer methods like conduction, convection, radiation happening through a solid-solid interface is dependent on interface type [1]. Hence, arises the true need for Thermal Interface materials. The specification of suitable thermal interface material (TIM) for a particular product application is an important part of this thermal engineering work. The emerging survey comprises a wide range of heat transfer technologies which are mechanisms for the conductive cooling which cooling of an electrical system and thermal dissipation material, such as a heat sink or a circuit board. More precisely, the innovation involves a stand-alone, form-stable film that is melted or softens within the temperature range or spectrum of the electronic part in order to properly adhere to thermal interfaces for the improved transfer of heat from the electronic to thermal dissipating components [2]. The Phase Change Materials (PCM) exhibits the good thermal behaviour of grease and possess flexibility of an elastomer pad. PCMs undergo phase change with temperature shifts and convert into liquid state with increasing temperature which is feasible for TIM fabrication. These include paraffin waxes or silicon-based materials. By adding metal nano filler particles, the thermal performance of PCM gets escalated [3, 4].

Scientist reported the higher efficacy of thermal transmission of aluminium coated spherical particles of Al_2O_3 dispersed in paraffin wax as a phase-change thermal interface material [5]. Promising research in the field of elastomers based TIM was carried out by a team of researchers; they synthesized copper nanowire- filled soft elastomer composite as thermal interface materials. [6]. Carbon based materials are becoming the choice in myriad of applications due to their novel properties, graphene being becoming the most researched one. In this league scientists incorporated a thin commercially available paper with water dispersible graphene paste to fabricate a novel TIM. The prepared TIM exhibited good thermal and mechanical properties [7]. Using liquid-phase exfoliation technique eminent researchers prepared graphene-multilayer graphene nanocomposite

for thermal efficient TIM applications. The results strongly suggested the exponential increase in thermal conductivity K of the composite [8]. Further to capitalize the maximum benefits of excellent thermal properties of graphene a group of scientist developed few-layer Graphene (FLG) composite, where FLG was prepared by the interlayer catalytic exfoliation (ICE) method. The thermal and mechanical characterization studies demonstrated feasibility of FLG composites as commercially reliable TIMs. The thermal interface resistance between FLG composite TIMs and copper was found to be marginally low in comparison to many commercial TIMs [9]. Researchers fabricated epoxy resin (ER) modified nano-series rGO based TIM and investigated the influence of types of rGO nano-series functional groups on the obtained thermal conductivity of TIM. The thermal conductivity and calculated thermal boundary resistance results proved the greater role of carboxyl groups in enhancing phonon transport at the interface between the graphene base plane and ER.[10]. Scientists designed a simulation method for creating a perfect system design for effective thermal management. An experimental set-up for static measurements is presented that evaluates the thermal conductivity and phase separation resistance of thermal materials (eg, glue, solder, washers or pastes) [11].

Scientist working in the field of thermal science investigated densely stacked vertically multi-walled carbon nanotube arrays as thermal interface materials. The results show that the physically bonded interface via van der Waals adhesion had a conductivity of around $105 \text{ W / m}^2\text{K}$. It also shows that by bonding the free end CNTs to the target surface with a thin layer of indium weld, the conductivity can be increased to a greater extent [12]. Sreekanth narumanchi and group extensively investigated the thermal performance of selected thermoplastics and phase change materials using the ASTM stabilization method and transient laser flash method. The experimental results on thermal resistance were carried out in the context of cooling automotive electronics. The results of the numerical finite element modelling greatly emphasize the importance of the thermal resistance of the TIM layer on the maximum temperature in the IGBT packet [13]. Eminent researchers investigated interfacial thermal resistance (ITR) between the TIM and aluminium surface by using the time domain thermo reflectance method. The ITR dramatically escalated due the depletion of filler layers present adjacent to the aluminium surface. This implies the importance to prevent formation of Filler Depletion Layer in the future thermal management of integrated devices [14]. Electrospun polyamide nanocomposite incorporated with rGO-Ag hetero structured nanoparticles were fabricated. The composites showed excellent thermal conductivity behaviour due the superior thermal conductivity of graphene fillers. All the thermal parameters including thermal conductivity and glass transition temperature of (Ag/rGO)/PI nanocomposites increased tremendously with increment of loading of Ag/rGO fillers [15]. TIMs are used in melange applications,

including machines, electronic electronics, telephone systems, LED lighting goods, green energy products, vehicles, military and manufacturing equipment and medical instruments. LEDs, thin film photovoltaic, automotive and medical devices, with slower development in mature electronics, military and automobile industries, are anticipated for the fastest rising areas. New innovations like phase change and metallic TIMs are leading in particular for usage at high temperatures, accompanied by adhesives in more conventional forms. The LED framework is expected to dominate the lighting market and over the next 10 years the TIM application is projected to be the fastest increasing and influential project. In recent times elastomeric pads are used anticipated to be the dominant type of TIM [16]. The literature explored above is represented in tabular manner concisely to explore the rationale of carrying out this present study.

Scope of the Present Study

According to the literature reported in our paper, various work involving filler materials like Aluminium, Go, rGO and copper nanowires have been studied for thermal conductivity. The materials have been prepared using mélange of methods including electrospinning and vertical aligned nano arrays. Thermal conductivity studies have been extensively studied. Not much importance was given in the study of effective thermal contact resistance, bond line thickness, PCMs as base fluid and pressure exerted by the TIM to optimize bond line thickness. In this league a

deep cut-through study was carried out to highlight the importance of temperature, pressure, the nature of filler particle, and bond line thickness which further affects the effective thermal contact resistance. Table 1 presents the quantitative exploration of literature study.

EXPERIMENTAL METHODS

Preparation of Thermal Interface Material

The silicon dioxide nanoparticle was prepared by chemical sol-gel method. The aluminium dioxide, copper oxide, graphene oxide and Silver- reduced graphene oxide nanoparticles were purchased from Techonano Taiwan to be used as filler particles. The size of all nanoparticles was in the range of 50-100 nm. The paraffin wax was supplied by Loba Chemical, India. Various samples of TIM were prepared by mixing 6 wt% of SiO_2 , Al_2O_3 , CuO, GO, rGO nanoparticles individually mixed in 5ml of paraffin wax and the samples were magnetically stirred first [17]. Then the mixture was sonicated for 30 minutes using ultra sonicator for uniform dispersion of nanoparticles in liquid Paraffin wax.. The optimal wt% used in the present study is chosen based on our previous experimental results reported in “Development of Low Resistance Thermal Interface Material (TIM) Using Nanomaterials” published in the International Journal of Engineering Science Invention. The formed fine paste was spread as a thin layer over the copper heater and sandwiched between aluminium sink

Table 1. Development of Thermal Interface Materials in Recent Years

S. No	Author	Materials used	Interpretation
1.	Mao D, Xie J, Sheng G, Ye H, Yuen MM, Fu XZ, Sun R and Wong CP [5]	Paraffin wax as base fluid and alumina and Al_2O_3 as filler particle	Alumina as filler particle shows the best thermal conductivity of 1.6743 W/mK compared to only paraffin wax which has thermal conductivity of 0.1504 W/mK
2.	Bhanushali S, Ghosh PC, Simon GP and Cheng W [6]	Flexible elastomer composite embedded copper nanowires.	The matrix showed a thermal conductivity of 3.1 W mK ⁻¹ which decreased with increasing temperature.
3.	Shahil KMF, Balandin AA [8]	Natural graphite as filler particle and aqueous sodium cholate solution mixed with epoxy as base fluid	The thermal conductivity of the base fluid increased from an initial value of ~5.8 W/mK to K = 14 W/mK at the small loading f = 2%.
4	Park W, Guo Y, Li X, Hu J, Liu L, Ruan X and Chen YP [9]	Graphene as filler particle and epoxy as base material.	The thermal conductivity increases as the filler concentration increased from 0 to 10 %. The highest thermal conductivity is measured to 3.87 ± 0.28 W/mK at 10 vol % at room temperature
5.	Sun Y, He Y, Tang B, Tao C, Ban J and Jiang L [10]	RGO as filler particle and epoxy as base material.	The thermal conductivity of the prepared TIMs was found to be $5.9\text{Wm}^{-1}\text{K}^{-1}$.
6.	Tong T, Zhao Y, Delzeit L, Kashani A [12]	Multiwalled CNT arrays grown over Si substrate.	Thermal conductivity was measured around 10^6 W/m ² K
7.	Guo Y, Yang X, Ruan K, Kong J, Dong M, Zhang J, Gu J, and Guo Z [15]	Electrospun Ag/rGO/ Polyimide nanocomposite as TIM	(Ag/rGO)/PI nanocomposite showed a maximum thermal conductivity of 2.12 W/(m K)

of the inbuilt test rig as shown in figure 1. The thickness of TIM spread at complete tightened screws was measured around 30 μm , as measured by a micrometre. At reduced pressure (RP) the thickness was observed to be 42 μm . Experiments were carried out at reduced pressure (RP) and full pressure (FP) which was created by completely loosening and tightening the screws of the aluminium sink. Power inputs were varied as 25, 50 and 75W. the figure 2 represent prepared nanofluids.

To improve the uniform dispersion of the nanoparticles in the base fluid, the mixture was subjected to ultrasonic agitation up to 30 minutes and dispersion was ensured using SEM images of the dried paste at periodic intervals. The stability of nanoparticles was checked using visual observation method. The data in days for which the nanofluids was stable is reported below.

Most of the nanoparticles showed high stability when stored as nanofluids in room temperature before getting precipitated. In our observation aluminium and copper

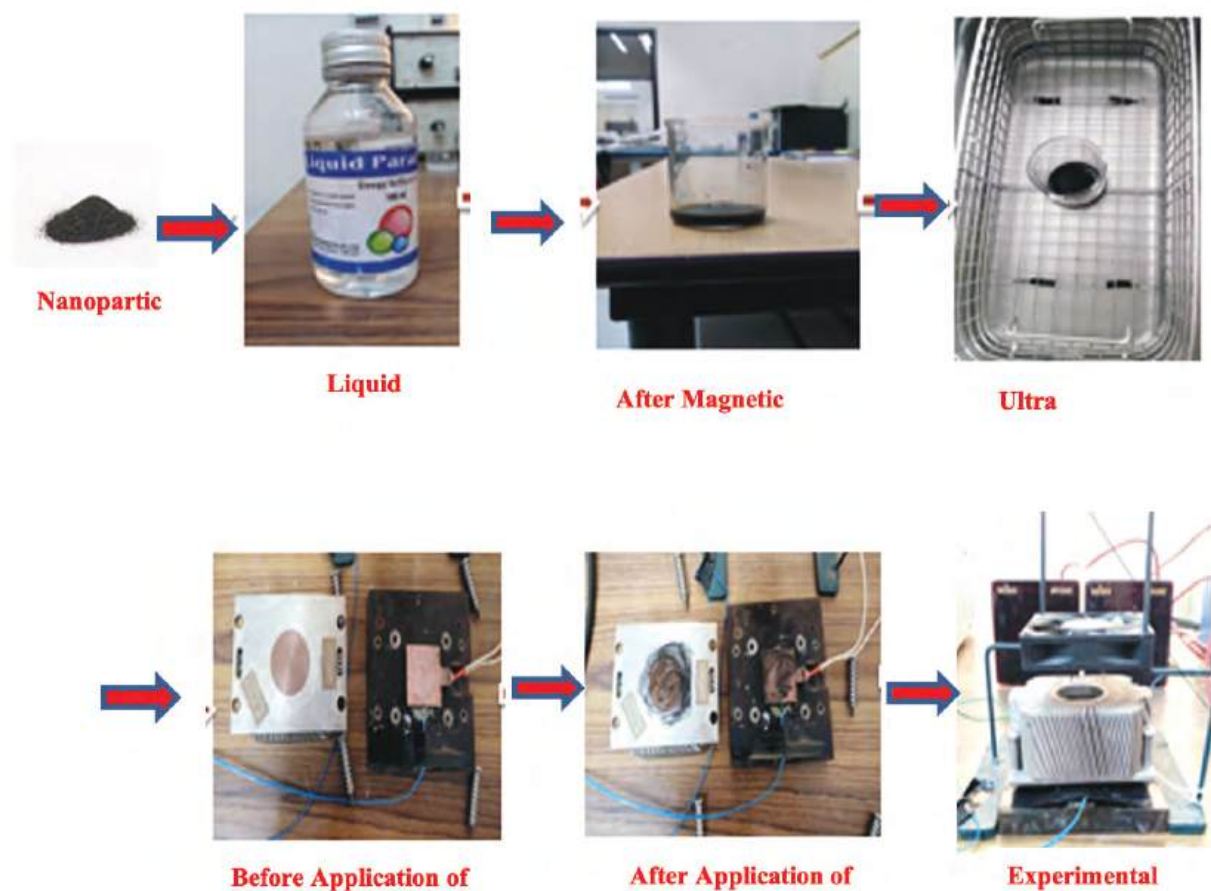


Figure 1. Preparation of TIM and Its Experimental Set Up.

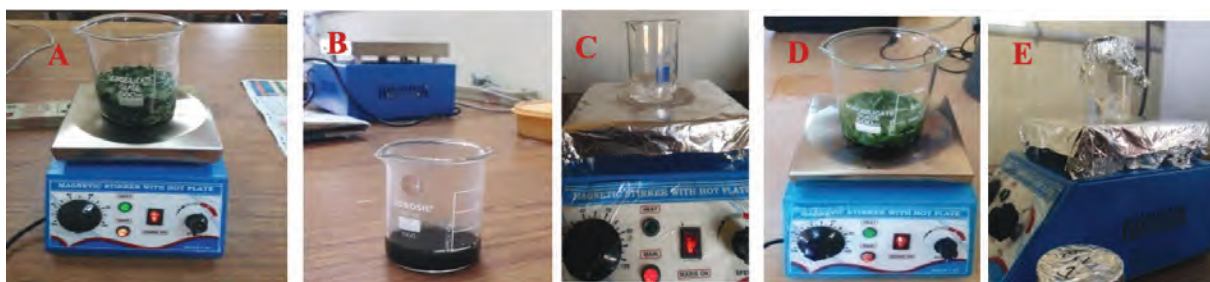


Figure 2. Shows prepared nanofluids of (A) CuO, (B) SiO₂, (C) GO, (D) rGO, (E) Al₂O₃.

Table 2. The Stability of Prepared Nanofluids in Days

S. No	Nanofluid	Stability in Days
1	SiO ₂	30
2	Al ₂ O ₃	72
3	CuO	70
4	rGO	55
5	GO	62

oxide nanofluids were the most stable nanoparticles and rGO and GO nanofluids represented similar and high stability. SiO₂ nanofluids showed least stability comparatively. The SEM images taken shows that the nanoparticles are dispersed uniformly even after ten days which can be witnessed in figure 3.

Test Set Up

Thermal experiments were carried to determine effective thermal contact resistance ($R_{th, eff}$) of various samples. Figure 4 shows schematic of the test rig.

Effective thermal contact resistance ($R_{th, eff}$) is calculated for prepared TIM using given equation.

$$R_{th, eff} = R_{th} + R_{th0, Cu-TIM} + R_{th0, Al-Air}$$

Where,

R_{th} = Thermal resistance of TIM material

$R_{th0, Cu-TIM}$ = Thermal interface resistance between the copper heater block and thermal interface material

$R_{th0, TIM-Al}$ = Thermal interface resistance between the aluminium heat sink and thermal interface material

$R_{th0, Al-Air}$ = Thermal interface resistance between the aluminium heat sink and air

In Figure the details of the experimental set up for measuring $R_{th, eff}$ is given.

The above given test rig (Fig. 5) represents the schematic of heat path established between the heater and sink and thermal interface materials bridging the rough surface existing between them. The alternating current (AC) system provides power to the heater cartridge (CH) which in turn raises the temperature of the copper heater

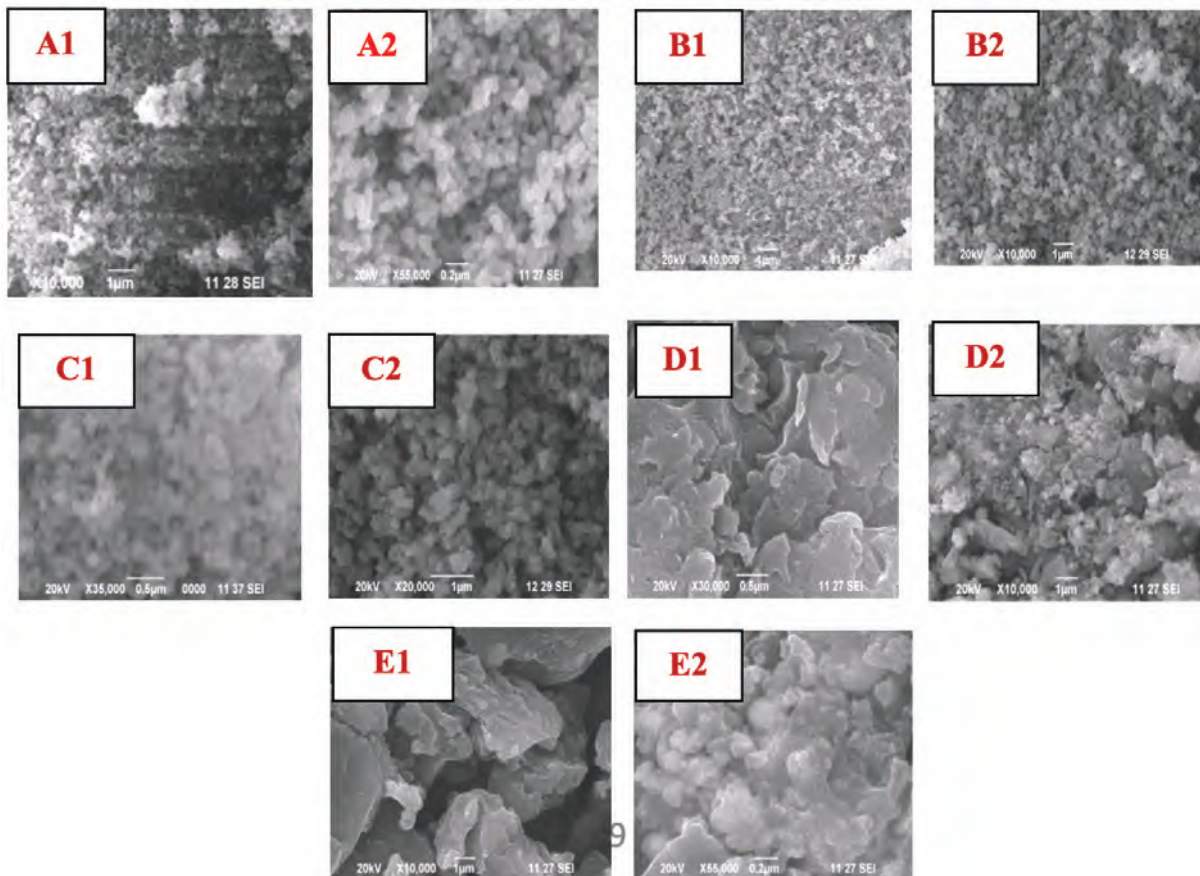


Figure 3. Shows SEM images of dried samples of (A1&A2) SEM images of dried paste of Al₂O₃ after day 1 and day 10 (B1&B2) CuO after day 1 and day 10 (C1&C2) SEM images of dried samples of SiO₂ after day 1 and day 10 (D1&D2) SEM images of dried samples of GO after day 1 and day 10 (E1&E2) SEM images of dried samples of rGO after day 1 and day 10.

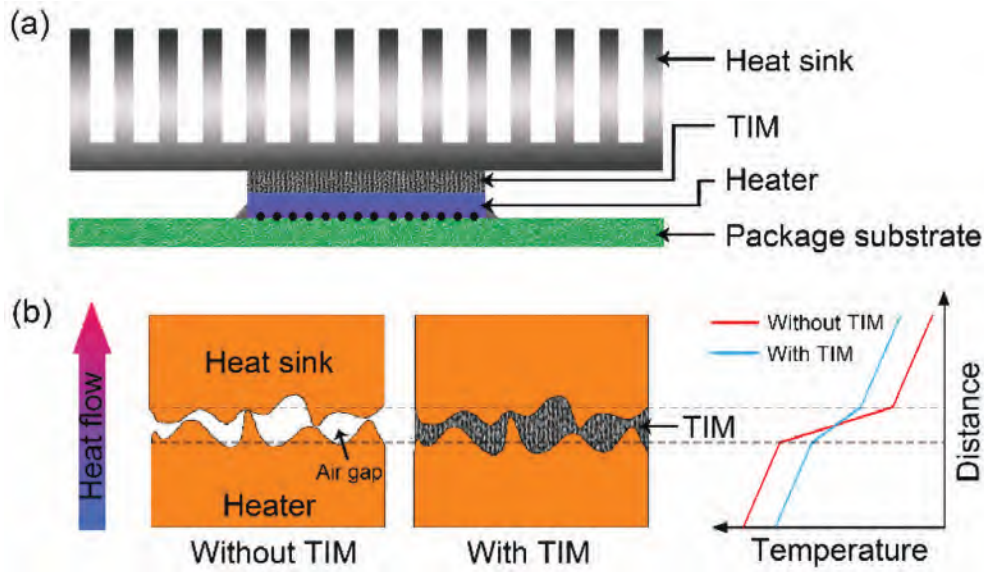


Figure 4. Schematic for Test Set Up.

EXPERIMENTAL SETUP OF PRELIMINARY TESTS FOR THERMAL INTERFACE MATERIALS

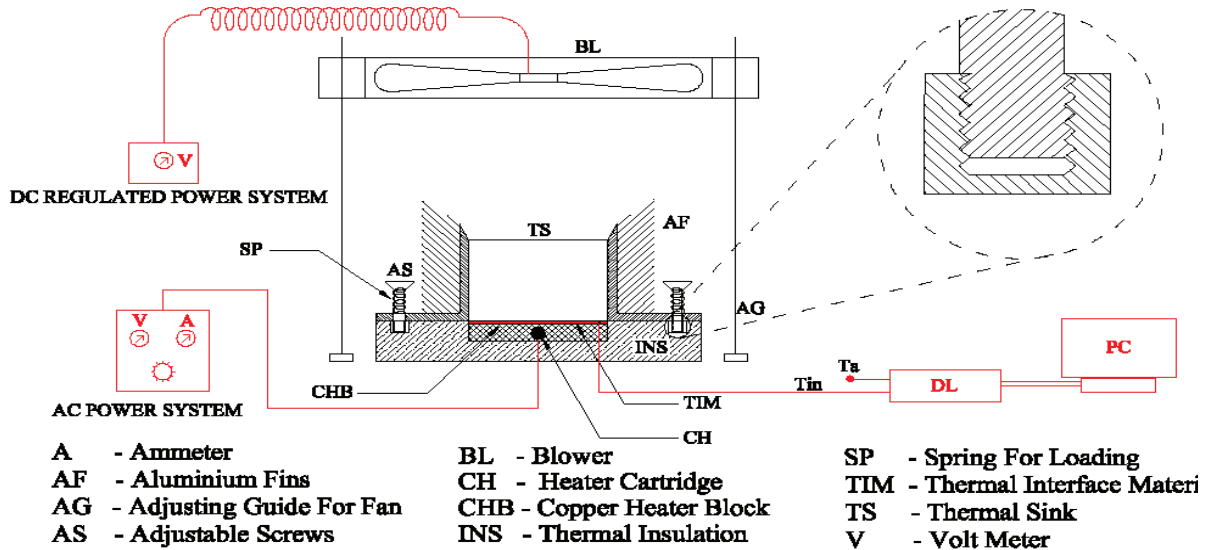


Figure 5. Experimental Setup of Preliminary Tests for Thermal Interface Material.

block (CHB), the elevated temperature being sensed and recorded using the Chrome-Alumel thermocouple system. The uncertainty of the temperature measurement is ± 0.2 Degree Celsius including the uncertainty in the voltmeter ± 0.1 Degree Celsius.

Uncertainty in power and resistance are calculated using following equations and the result is tabulated in table 3

$$\frac{\Delta p}{p} = \sqrt{(\Delta V/V)^2 + (\Delta I/I)^2}$$

$$\frac{\Delta R}{R} = \sqrt{(\Delta T/T)^2 + (\Delta P/P)^2}$$

The sensitivity of temperature sensors was measured at ± 0.2 °C. A strong aluminium block with multiple close

Table 3. Uncertainty Calculation of Power and Resistance

S. No	Heat Input In Watts	Uncertainty In Power	Uncertainty In Resistance
1	25	2.8%	3.2%
2	50	2.2%	2.6%
3	75	1.6%	1.9%

Table 4. Sensitivity of Devices

S. No	Device	Sensitivity
1.	Temperature sensor	$\pm 0.2\text{ }^{\circ}\text{C}$.
2.	Voltmeter	$\pm 0.1\text{ }^{\circ}\text{C}$.

**Figure 6.** Shows the Instrumentation of Thermal Interface Material Set-up.

stacked fins precisely noted as Aluminium fins (AF) was used as thermal sink (TS). It is fixed to the copper heater block using adjustable screws (AS) and loading spring (SP). TIM is applied over the appropriate area on CHB and aluminium sink is fixed over it. By completely loosening and tightening the screws, applied pressure on thermal interface material is varied. An air blower (BL) is fixed over the heat sink with the help of adjusting guide (AG) to dispel the heat absorbed by the sink to the atmosphere. Difference between CHB temperature and room temperature is recorded during experimentation at decided power inputs. Rotational speed of the air blower was measured using anemometer and found to be 6.8 m/s. The recorded data (DL) is interfaced to PC for further simulations. Figure 6 Shows

the indigenously fabricated instrumentation of Thermal Interface Material Set-up.

RESULTS AND DISCUSSION

Effective thermal contact resistance ($R_{th, eff}$) based on the fabricated set up is calculated as

$$(R_{th, eff}) = (T_2 - T_1) / P \text{ (watts)}$$

Where T_2 and T_1 represent the heater temperature and ambient temperature and p is the applied power.

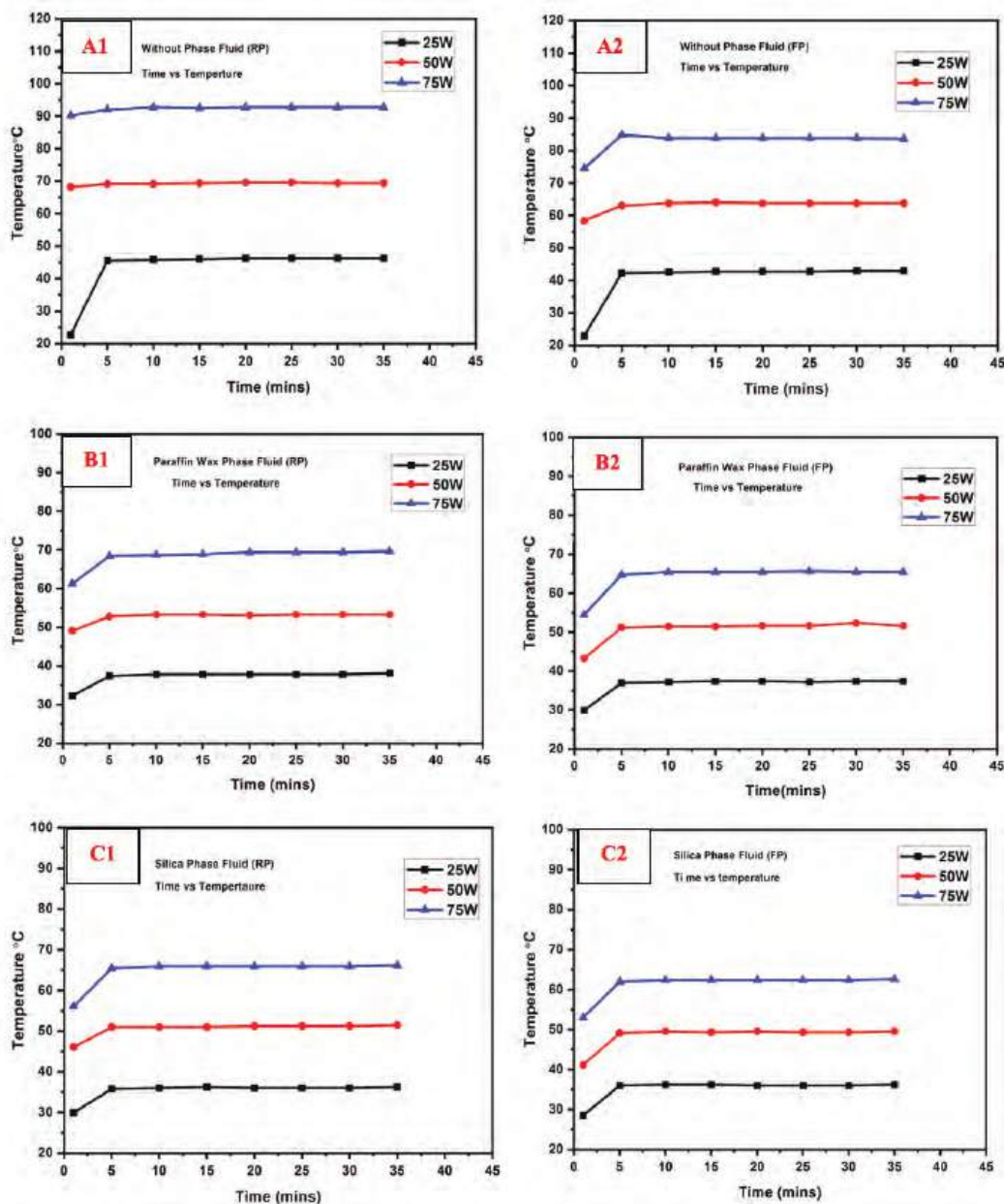
Variation of Heater with Time and Pressure

Firstly, the temperature of CHB was measured on appropriate time intervals without thermal interface material

applied between the interfacing surfaces as depicted in figure 7 (A1-A2) to particularly determine the effective thermal contact resistance of CHB. Secondly, in the similar experimental conditions temperature variation of only base fluid was determined as given in figure 7 (B1-B2). Figure 7 (C1-C2) to 7 (G1-G2) gives the variation of the CHB temperature at appropriate time interval for different TIM at (RP) and (FP) which includes only base fluid and composites of base fluid and individual nanoparticles (SiO_2 , Al_2O_3 , CuO , GO and rGO) based TIM. Power inputs were changed as 25W, 50W and 75W for study. Effective thermal contact resistance of all the samples were compared using graphs at full pressure and reduced pressure which is depicted from figure 8 to 13. ($R_{th, eff}$) is significantly less for nanoparticle

and base fluid solution compared to base fluids alone. A rather significant observation is made that at full pressure ($R_{th, eff}$) for all materials is ver low as compared to the value at reduced pressure reinforcing the importance of bond line thickness in assessing thermal performance of TIM.

The graphs for without any interface material for both full pressure and reduced pressure depicts very less transmission of heat from heater to sink at various temperature levels and time depicting the need for interface material as the gap between heat and sink is prominent and increases the effective thermal contact resistance. The presence of base fluid does make a small difference in the reduction of heater temperature over the time compared to the reading of without any interface material. The reason can be



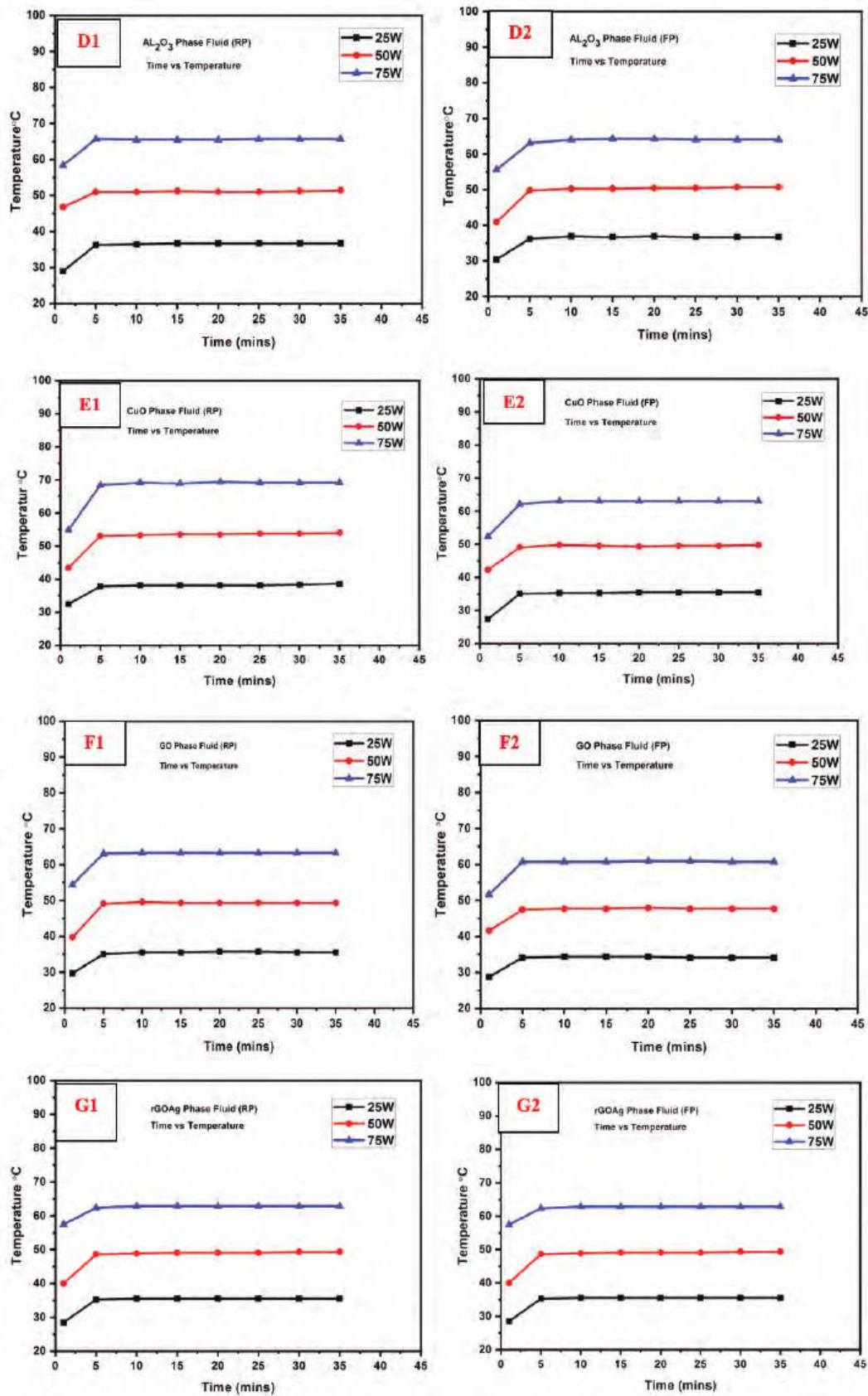


Figure 7. (A1-A2) , (B1-B2), (C1-C2), (D1-D2), (E1-E2), (F1-F2) and (G1-G2) represents Graph of time Vs heater temperature for without any interfcae material, paraffin wax, SiO_2 , Al_2O_3 , CuO, GO, rGO at difference power inputs of 25W, 50W and 75W and at full pressure (FP) and reduced (RP).

attributed to filling of microgaps and the contact between heater and sink, which aids in better flow of heat and hence reduction of effective thermal contact resistance to some extent. Here now the challenge is to improve the thermal conductivity of interface material for better results.

It is clearly observable from all the above graphs that heat dissipating capability is highest for all the nanoparticles based TIM at full pressure. The decreasing trend of ($R_{th, eff}$) at full pressure can be attributed to decreasing bond line thickness and enhanced contact between TIM and aluminium sink which helps in good evacuation of heat.

Effective Thermal Contact Resistance of Nanoparticles Based TIM

Effective thermal contact resistance for optimized 6 wt% of (SiO_2 , Al_2O_3 , CuO , GO and rGO) nanoparticles

mixed in 5ml of paraffin wax was determined at the power inputs of 25W, 50W and 75W using the experimental set up given above.

From figure 8 to 13 represents the graphical depiction of effective thermal contact resistance of various TIM at different power inputs of (25 W, 50 W, 75 W), secondly at uniformly increasing time and at reduced pressure and full pressure. After the keen interpretation of the graphs it is deduced that time plays a mere role as not much difference is observed in the $R_{th, eff}$. After the initial escalation as the time increases the value almost stabilizes at a saturated point for all TIM and for all the varied power inputs. A major contribution to the steady raise of $R_{th, eff}$ in all TIM was the increasing power input as it heats up the cartridge according to the strength of applied power. It becomes an incumbent

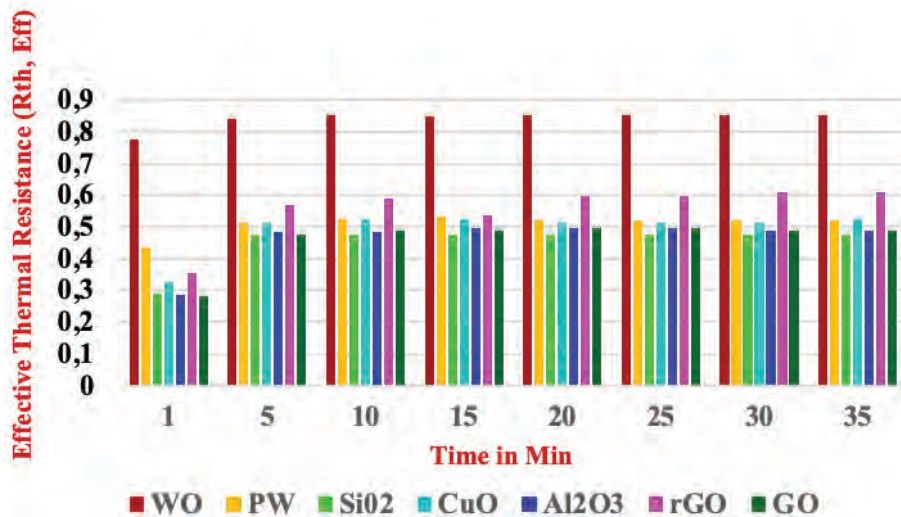


Figure 8. Comparison of different TIM at 25W input power at reduced pressure.

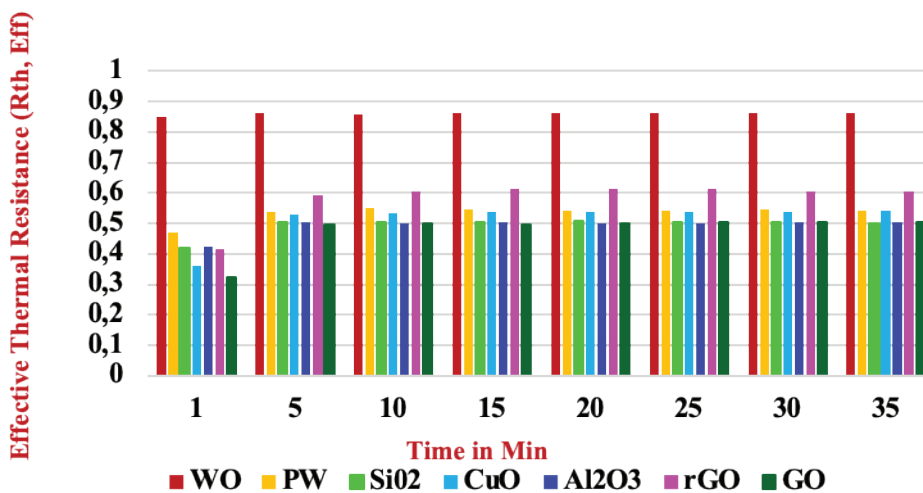


Figure 9. Comparison of different TIM at 50 W input power at reduced pressure.

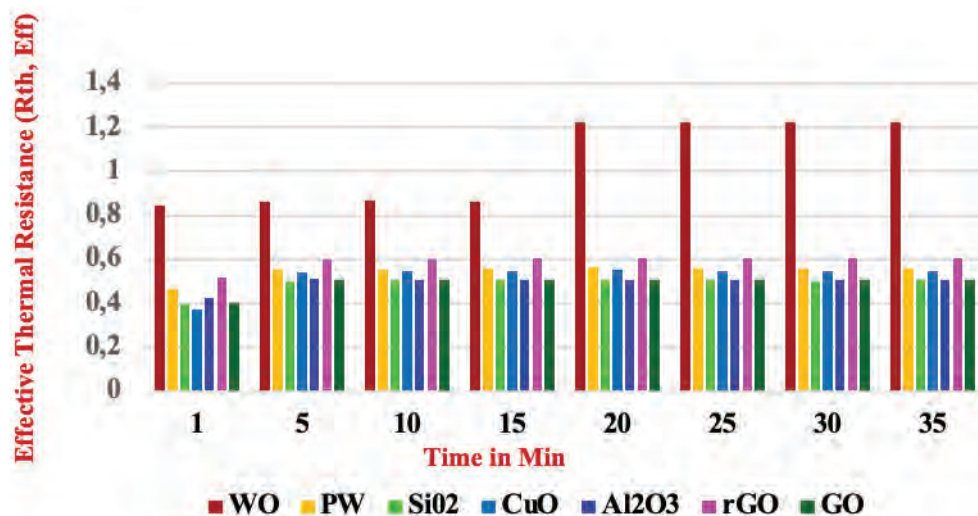


Figure 10. Comparison of different TIM at 75W input power at reduced pressure.

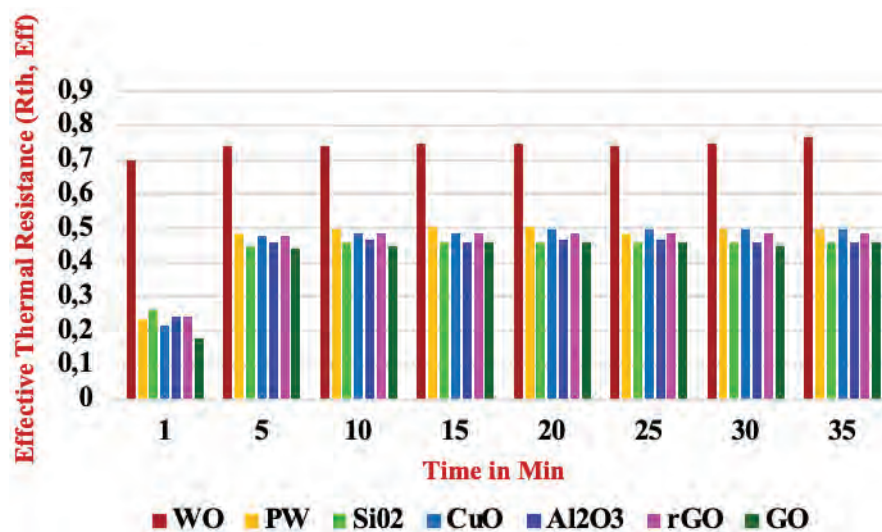


Figure 11. Comparison of different TIM at 25W input power at full pressure.

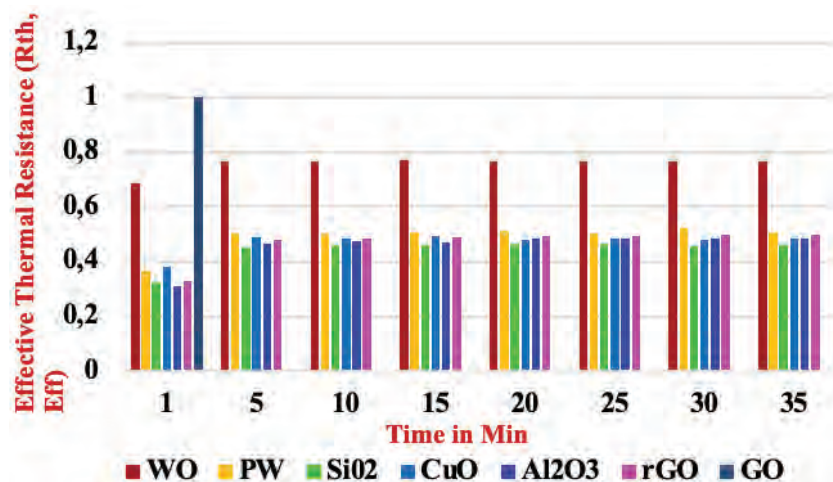


Figure 12. Comparison of different TIM at 50 W input power at full pressure.

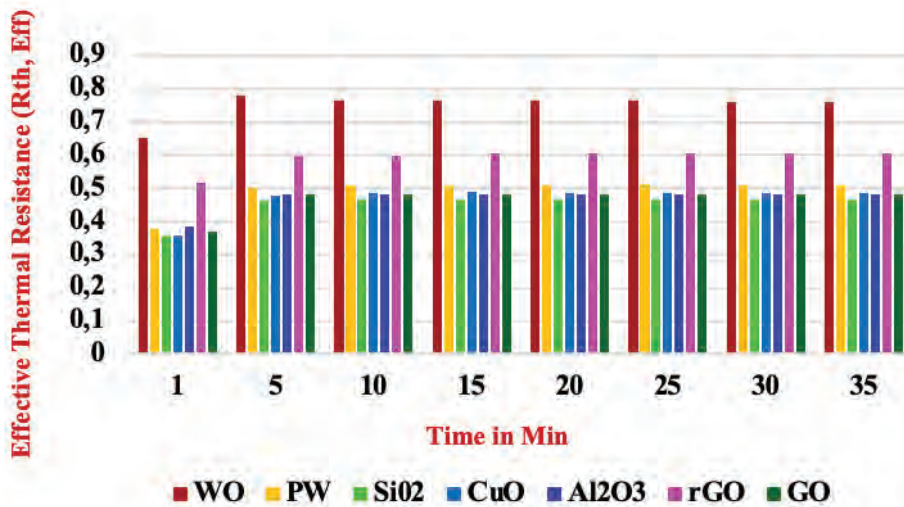


Figure 13. Comparison of different TIM at 75W input power at full pressure.

task to dissipate the heat from the electronic device as the power is raised. It is observed that $R_{th, eff}$ values are significantly low after application of paraffin wax base fluid when compared to no base fluid condition. It is clearly evident that without base fluid the surface unevenness between the heater and sink is high and air fills the gap which has a very poor thermal conductivity of 0.025 W/m.K which resists the heat transfer. In this study, liquid paraffin wax was used which has a thermal conductivity of 0.3 W/m.K. It is hence validated that the base fluid fills that gap providing better heat transfer. Paraffin wax depicts 80% enhanced thermal conductivity compared to air. Further it is observed that addition of (SiO_2 , Al_2O_3 , CuO, GO and rGO) nanoparticles to the base fluid paraffin wax indeed decreases the effective thermal resistance to a large extent. In order to obtain homogenous mixture of nanoparticles and base fluid, uniformly 6wt% of all (SiO_2 , Al_2O_3 , CuO, GO and rGO) nanoparticles was used for the present study. Noteworthy reduction in effective thermal resistance is observed as the incorporated nanoparticles help in better heat conduction due to its higher surface area which enhances the thermal dissipation properly and further dipping in of effective thermal resistance ($R_{th, eff}$) is observed. Firstly we measured the without any base fluid and interface material readings at different power inputs of (25, 50 and 75W) at reduced pressure and full pressure. In the same manner readings were taken for the base fluid of 5ml paraffin wax and (SiO_2 , Al_2O_3 , CuO, GO and rGO) NPs sonicated and developed as a uniform paste-based TIM. Full pressure amounts to greater stress exerted at the sandwiched surface creating a closer interface between heat die and sink which helps in creating a conducive heat path. Hence, enhanced results are observed at full pressure.

In similar kind of research scientist have prepared nanofluids with base fluid DI water and filler particles Al_2O_3 , SiO_2 and ZrO_2 . Thermal conductivity of nanofluids

enhanced with increment of volume concentration of filler particles and these three nanofluids exhibited enhanced thermal conductivity than base fluid DI water. Thermal conductivity of the DI water at room temperature of 30°C and found to be 0.6 W/mK. It was clearly visible that the thermal conductivity enhancement of 10.13% for Al_2O_3/H_2O , 6.5% for SiO_2/H_2O and 8.5% for ZrO_2/H_2O happened at 1% volume concentration [18]. Another highlighting observation is made that at reduced pressure ($R_{th, eff}$) escalates owing to poor gap filling and lack of fine bondline thickness. In depth evaluation of the presented graphical data gives an insight into the individual nanoparticle performance towards achieving superior TIM. rGO unexpectedly fails in conducting the heat favorably as its effective thermal resistance were almost comparable to that of base fluid paraffin wax. The answer may lie in agglomeration or poor dispersive of rGO in paraffin wax which might have hindered the proper heat flow. SiO_2 , CuO and Al_2O_3 give almost similar results. All the three filler materials are very good conductors of heat with proven high thermal conductivity factor. Maximum uncertainty measurement for results at 25 W was found around 3%.

Figure 14 and 15 depicts the average effective thermal contact resistance of different TIM at different power inputs and RP followed by FP. It is clearly evident that GO gives the best results as humane reduction of effective thermal resistance is observed for all the parameters. A Recent research showcases the fabrication of multilayer graphene-paraffin composite PCMs with different mass fractions. The thermal conductivities of composite PCMs enhanced with an increasing weight percentage of multilayer graphene. 10 wt % graphene in the PCM composite exhibited a 150% increase in thermal conductivity [19]. Our finding corroborates with the above stated research perfectly. Silica, CuO and Al_2O_3 give similar results.

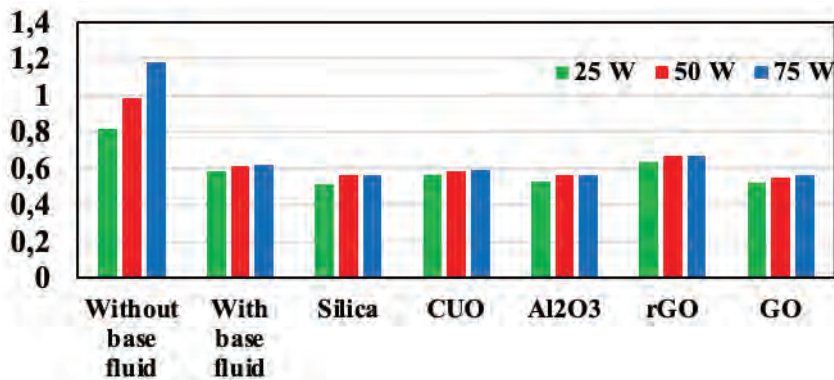


Figure 14. Comparison of average (R_{th} , E_{eff}) of different TIM at RP.

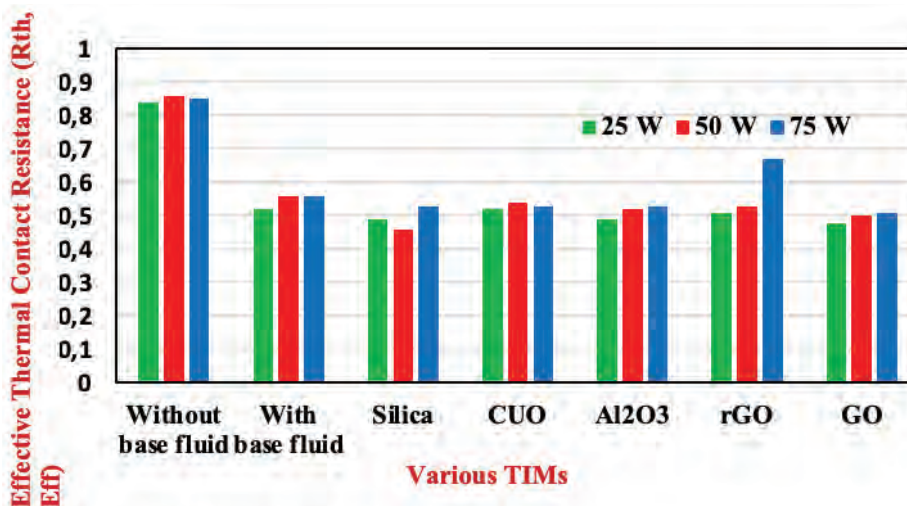


Figure 15. Comparison of average (R_{th} , E_{eff}) of different TIM at FP.

CONCLUSION

The present experimental study reports the relative study of effective thermal contact resistance ($R_{th, eff}$) measured for different thermal interface materials prepared as fine composite paste using paraffin wax as base fluid and SiO_2 , Al_2O_3 , CuO , GO and rGO as filler particles at different pressures, power inputs (25W, 50W and 75W) and time variations. In house built test rig using aluminium fin-based sink and copper heater was utilized to carry out heat dissipation experiments. It is clearly evident from the data that without any interface material the thermal contact resistance is observed to be highly escalated. The value at 75 W and FP is $0.85^\circ C/W$, whereas when base fluid is added it drastically reduces to $0.6^\circ C/W$. These results fortifies the need and importance of filling the uneven gap between heater and sink for proper dissipation of heat. As an overall observation, addition of nanoparticles (Al_2O_3 , CuO , GO , rGO) in paraffin wax base fluid significantly improved the proper evacuation of heat and hence lower

values of ($R_{th, eff}$) were observed in nanoparticles based TIM. Interestingly silica based TIM gives the best performance of heat transference to ambience at 50 W power input ($R_{th, eff} = 0.46^\circ C/W$) and slightly becomes inferior to GO at higher power input. Again instigating the researches to seriously ponder on the agglomerative behaviour of nanoparticles at higher temperature. This behaviour is observed at raised values of ($R_{th, eff}$) around $0.67^\circ C/W$ for rGO based TIM, measured at 75 W and FP. GO based TIM shows the best results at higher power of 75 W, its $R_{th, eff}$ being $0.51^\circ C/W$ reinforcing the behaviour of GO nanoparticle stability at high temperatures. Exceptions of results apart, at 75 W and FP conditions, all nanoparticle based TIMs have shown similar results for ($R_{th, eff}$) i.e $0.53^\circ C/W$ for silica, CuO and Al_2O_3 based TIMs. Highlighting the importance of pressure on TIM, it was inferred from the findings that at full pressure ($R_{th, eff}$) lessens significantly for all samples.

NOMENCLATURE AND ABBREVIATION:

TIM	Thermal Interface Material
$R_{th, eff}$	Effective thermal contact resistance
RP	Reduced Pressure
FP	Full Pressure
WO	Represents the Without base fluid
PW	Represents the Paraffin wax base fluid
SiO ₂	Represents the SiO ₂ based TIM
CuO	Represents the CuO based TIM
Al ₂ O ₃	Represents the Al ₂ O ₃ based TIM
GO	Represents the GO based TIM
rGOAg	Represents the rGOAg based TIM
CHB	Copper Heater Block
HC	Heater Cartridge
AF	Aluminum Fins
TS	Thermal Sink
AC	Alternating Current
AS	Adjusting Screws
BL	Blower
AG	Adjusting guide for fan
SP	Spring for loading

HIGHLIGHTS

1. Nanoparticle based thermal interface materials prepared for electronic cooling application
2. TIM prepared with GO and aluminum nanoparticles gives the best results.
3. ($R_{th, eff}$) visibly decreases for all samples at full pressure which significantly tells the importance of bondline thickness.

ACKNOWLEDGMENTS

We are grateful to Karunya University for providing excellent facilities to carry out the entire research work. Also, I acknowledge the contribution of Mother Teresa Women's University, Kodaikanal in successfully accomplishing the research work.

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

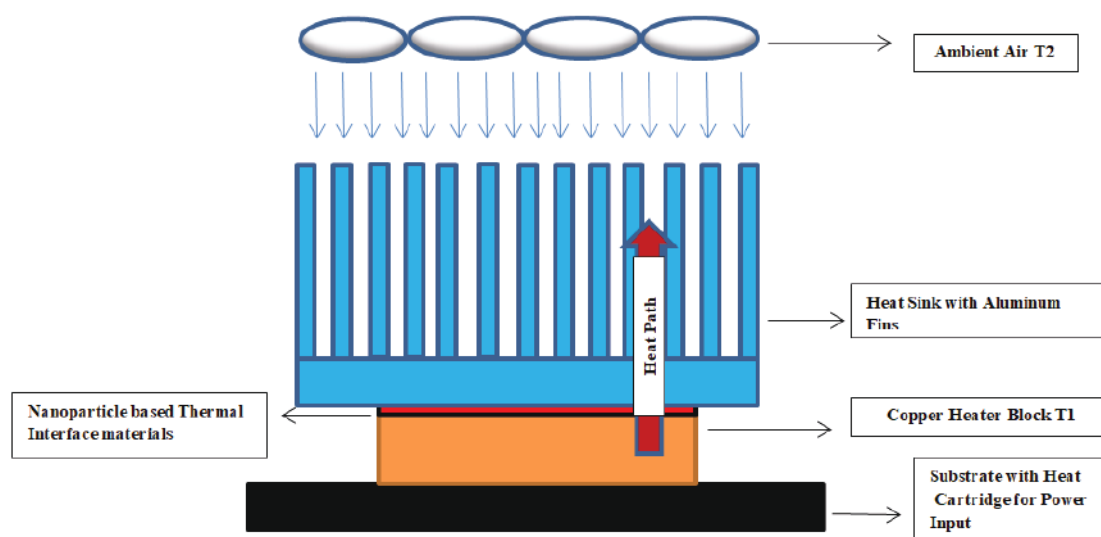
Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Micheal HB, de sorgo M, John AM. Permalink Machine translation. 2005;US6054198
- [2] Micheal HB, de sorgo M, John AM. Conformal thermal interface material for electronic components. 2000; US6054198.
- [3] Prasher R. Thermal interface materials: historical perspective, status, and future directions. Proc IEEE. 2006; 94:1571-1586. [\[Crossref\]](#)
- [4] Razeeb KM, Dalton E, Cross GLW, Robinson AJ. Present and future thermal interface materials for electronic devices. Int Mater Rev. 2018; 63:1-21. [\[Crossref\]](#)
- [5] Mao D, Xie J, Sheng G, Ye H, Yuen MM, Fu XZ, Sun R, Wong CP. Aluminum coated spherical particles filled paraffin wax as a phase-change thermal interface materials. In: Proceedings of the 18th International Conference on Electronic Packaging Technology ;2017. pp 828-830. [\[Crossref\]](#)
- [6] Bhanushali S, Ghosh PC, Simon GP, Cheng W. Copper nanowire-filled soft elastomer composites for applications as thermal interface materials. Adv Mater Interfaces 2017; 4:1700387. [\[Crossref\]](#)
- [7] Lv L, Dai W, Yu J, Jiang N, Lin CT. A mini review: application of graphene paper in thermal interface materials. New Carbon Mater. 2021; 36:930-8. [\[Crossref\]](#)
- [8] Shahil KMF, Balandin AA. Graphene-multilayer graphene nanocomposites as highly efficient thermal interface materials. Nano Lett. 2012; 2:861-7. [\[Crossref\]](#)
- [9] Park W, Guo Y, Li X, Hu J, Liu L, Ruan X, et al. High-performance thermal interface material based on few-layer graphene composite. J Phys Chem C 2015;119:26753-26759. [\[Crossref\]](#)
- [10] Sun Y, He Y, Tang B, Tao C, Ban J, Jiang L. Influence from the types of surface functional groups of RGO on the performances of thermal interface materials. RSC Adv 2017;7:55790. [\[Crossref\]](#)
- [11] Schacht R, May D, Wunderle B, Wittler O, Gollhardt A, Michel B, et al. Characterization of thermal interface materials to support thermal simulation. 2007. arXiv:0709.1849v1 [\[Crossref\]](#)
- [12] Tong T, Zhao Y, Delzeit L, Kashani A, Meyyappan M, Majumdar A. Dense vertically aligned multiwalled carbon nanotube arrays as thermal interface materials. IEEE Trans Compon Packag Manuf 2007;30:1. [\[Crossref\]](#)

- [13] Narumanchi S, Mihalic M, Kelly K, Eesley G. Thermal interface materials for power electronics applications. 11th Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems IEEE 2008;395-404. [\[Crossref\]](#)
- [14] Yada S, Oyake T, Sakata M, Shiomi J. Filler-depletion layer adjacent to interface impacts performance of thermal interface material. AIP Adv 2016;6:015117. [\[Crossref\]](#)
- [15] Guo Y, Yang X, Ruan K, Kong J, Dong M, Zhang J, et al. Reduced Graphene Oxide Heterostructured Silver Nanoparticles Significantly Enhanced Thermal Conductivities in Hot-Pressed Electrospun Polyimide Nanocomposites. ACS Appl Mater Interfaces 2019;11:25465-73. [\[Crossref\]](#)
- [16] Chung DDL. Thermal Interface Materials. J Mater Eng Perform 2001;10:56-59. [\[Crossref\]](#)
- [17] Maheshwari SU, Pillai BC, Govindan K, Raja M, Raja A, Pravin MBS, et al. Development of Low Resistance Thermal Interface Material (TIM) Using Nanomaterials. Int J of Eng Sci Inv 2018;2319-6734:39-46.
- [18] Iqbal SM, Raj CS, Michael JJ, Irfan AM. A Comparative Investigation of $\text{Al}_2\text{O}_3/\text{H}_2\text{O}$, $\text{SiO}_2/\text{H}_2\text{O}$ and $\text{ZrO}_2/\text{H}_2\text{O}$ Nanofluid for Heat Transfer Applications. Digest J Nanomater Biostruct 2017;12:255-263.
- [19] Balan AE, Sharea AA, Lavasani EJ, Tanasa E, Voinea S, Dobrica B et al. Paraffin-Multilayer Graphene Composite for Thermal Management in Electronics. Materials 2023;16:2-11. [\[Crossref\]](#)

Graphical abstract



Principle of Effective Heat Transfer Using TIM



Research Article

Performance of Mylar and Teflon as compound parabolic concentrators in solar desalination system

Shridhar KEDAR^{1,*}, Deepak WATVISAVE¹, Prerna MISHRA², Ganesh Vijay MORE³,
Adarsh Abi MATHEW⁴, Neeta MANDHARE⁵, Sumedh INGLE⁶

¹Department of Mechanical Engineering, MKSS's Cummins College of Engineering for Women, Karvenagar Pune, 411052, India

²School of Mechanical and Manufacturing Sciences, JSPM University Pune, Pune 412207, India

³Department of Mechanical Engineering, PVG's College of Engineering, Technology and Management, Pune 411009, India

⁴Renewable Energy Techniques Conestoga College Kitchener, Ontario, 1530, Canada

⁵Department of Mechanical Engineering, Pimpri Chinchwad College of Engineering, Pune Maharashtra, 411044, India

⁶Department of Mechanical Engineering, Sanjivani College of Engineering Kopergaon, Maharashtra, 423603, India

ARTICLE INFO

Article history

Received: 15 July 2024

Accepted: 18 December 2024

Keywords:

Compound Parabolic
Concentrator; Evacuated Tube
Collector; Reflectivity; Solar
Energy

ABSTRACT

The water-energy nexus is an important and difficult issue and must be resolved for present and also for the future. The process of producing freshwater requires a lot of energy, therefore a workable solution to this issue is crucial. In the current situation, solar energy is one of the best options for desalination as it is inexpensive, environment friendly, and widely accessible. In general, flat plate collectors and evacuated tube collectors have been used as solar collector for desalination. In this study, a single-stage hybrid groundwater solar desalination system has been used for experimental investigation. Compound parabolic concentrator is positioned to gather solar radiations and transfer heat to evacuated tubes, for improving the performance under various weather conditions in Pune, India. The ideal distance between evacuated tube collector and compound parabolic concentrator was 20 mm. The current study primarily focuses on the performance of two polymeric materials, Teflon and Mylar, as reflector, on the rate of soft water production. It was observed that when mylar was used reflecting material the rate of soft water production amounts to be 3.5 litres per day whereas teflon was used as reflecting material the rate of soft water production amounts to be 3.0 litres respectively. As per the validation of results for hybrid solar desalination system gives better results than solar-wind hybrid energy system. Thus Mylar shows more promising result than Teflon for production of soft water. It shows 20 % increase soft water production using mylar material on compound parabolic concentrator as compared to the teflon.

Cite this article as: Kedar S, Watvisave D, Mishra P, More GV, Mathew AA, Mandhare N, Ingle S. Performance of Mylar and Teflon as compound parabolic concentrators in solar desalination system. J Ther Eng 2025;11(5):1327–1338.

*Corresponding author.

*E-mail address: sangola_shri@yahoo.co.in, shridhar.kedar@cumminscollege.in

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkılıç



INTRODUCTION

The worldwide need of solar energy, to produce power and perform other beneficial tasks, is constantly increasing. The largest global problems are the growing population, the need for fresh drinking water, and the demand for power to meet the needs studied by Ghorbani et al. [1]. Since, there is less freshwater and electricity available in rural areas, life of human beings is very tough. Solar energy is an attractive alternative, particularly in rural regions, for producing drinking water at low cost. In globe scenario, the major source of water (roughly 97%) is ocean, and about 2% water is from variety of sources, including lakes, rivers, and groundwater studied by Baci et al. [2]. The remaining 1% or less of the freshwater will be used for people and animals, some of which is in the form of ice. Freshwater delivery to urban and rural communities is one of the biggest global problems studied by Ahmadi et al. [3]. The development of cutting-edge technologies like desalination and recycling, as well as investments in water infrastructure, are some of the strategies being used to overcome the shortage of water. Ensuring that everyone has access to clean and sufficient water is an important concern that requires international cooperation and coordination argued by Kalogirou [4].

Desalination, the process of purifying salt and other contaminants from saltwater to make it fit for farming and human use, is receiving more attention as a long-term response to the world's water shortage. Use of solar energy in desalination is a potentially beneficial approach. Through solar stills or solar-assisted reverse osmosis systems, solar desalination system uses the sun's abundant power to generate sufficient power for the process of purification. Seawater is heated by sunlight in solar stills, which causes evaporation of water and leaves behind impurities and salt, after that water vapour condenses into freshwater. On the other hand, solar-powered pumps in solar-assisted reverse osmosis systems pressurise seawater across membranes to separate contaminants and salt. Using solar energy for desalination offers a viable way to meet the growing demand for fresh water in an environmentally responsible way while simultaneously reducing dependency on fossil fuels and contributing to the global movement towards sustainable and renewable energy sources.

Devices called solar collectors are made to collect sunlight and transform it into useful heat or power. They are essential for the process of using solar energy for a variety of purposes. Solar thermal collectors and solar photovoltaic (PV) collectors are the two primary categories of solar collectors. The solar thermal collectors are further classified into flat plate collector and concentrating collectors. Among the two, concentrating collectors increase the strength/intensity of the sun rays by concentrating them onto a smaller area using mirrors or lenses. Heat is subsequently produced due to the concentrated light. Concentrating collectors are available in variety of forms, such as solar power towers, parabolic dishes, parabolic trough concentrator

(PTC) and compound parabolic concentrators (CPC). Among all these, CPC has highest efficiency because of its geometrical shape. It limits the wastage of solar energy from ETC tube by concentrating it back to the evacuated tube. Desalination systems including reverse osmosis (RO), electro-dialysis (ED), and multistage flash (MSF) have been created by engineers and scientists, however, the high cost of the produced energy is one of the major limitations. Use of fossil fuels to a large extent will have an adverse effect on the environment studied by Ali et.al [5]. Kedar et.al [6] had mainly studied effect of reflecting material on CPC to improve the performance of hybrid groundwater solar desalination system. Immersed fins were the subject of an experimental study by Moungar et al. [7] that will affect the generation of double slope stills. Their findings demonstrate that the authors organized the entire study for south Algeria for a variety of meteorological and environmental circumstances in order to attain superior soft water production utilizing a double slope solar still. The traditional solar still was created by Sathyamurthy et al. [8] using a sand heat energy storage method. While Morad et al. [9] worked on improving thermal performance in the solar still, their experimental results demonstrate high heat storage capacity in the solar still. Bhambare et al. [10] primarily investigated solar thermal desalination as a viable and environmentally friendly option for the Sultanate of Oman. The research primarily demonstrates the viability of solar thermal desalination for Oman. Mohd A. Al-Nimr et.al [11] primarily investigated a solar wind water desalination system. The findings primarily indicate that the hybrid desalination system produces 0.5 litres of soft water per day. Franz Trieb [12] primarily focused on researching concentrated solar power for the purpose of desalinating seawater in the Middle East and North Africa. Experimental research was conducted on the use of concentrated solar electricity for desalinating saltwater. Enas Taha Sayed et al. [13] primarily investigated desalination methods using renewable energy in the Middle East and North Africa (MENA) region. An investigation was conducted on desalination facilities in the Middle East and North Africa (MENA) region that are powered by renewable energy sources (RES). Renewable energy technologies are rapidly advancing and becoming more affordable, making them a promising alternative to conventional fuel-powered facilities. An exergy-based optimization of solar thermal collectors was done by Grosu et al. [14]. For forecasting the effectiveness of flat-plate solar collectors, Sadeghzadeh et al. [15] used smart models based on machine learning techniques. The use of nanofluids has been considered as a potential method of enhancing the efficiency of solar parabolic collectors by Olia et al. [16]. ANN model to predict the performance parabolic dish collector entirely studied by Loni et al.[17] whereas solar heat flux distribution in hybrid combination through heat pipe and biomass gasifier analysed by Jilte et al.[18]. A parabolic trough collector's operating parameters were tracked, and a unique sensitivity analysis was carried out by Rafiei et al.

[19] under specific conditions. Sampathkumar et al. [20] thorough study of active solar distillation provides insight into the direction of future research in the area of composite materials. Thermal performance of the desalination system was studied by Siddiqui et al. [21] utilizing a solar air heater. Their findings demonstrate that the observed productivity improved to 72% and the gained output ratio (GOR) increased up to 57%.

Kabeel et al. [22] had mainly studied hybrid solar desalination system of air humidification, dehumidification and water flashing evaporation. He argued that air humidification and water flashing evaporation get good agreement in solar desalination system in urban areas whereas Sapre et al. [23] mainly studied the design and manufacturing of absorber for solar desalination system. Zheng et al. [24] mainly experimentally investigated multi-effect tubular solar desalination devices whereas Khalil et al. [25] mainly analysed solar water desalination using an air bubble column humidifier. Ahmed et al. [26] mainly studied the characteristics of multistage evacuated solar distillation whereas Wu et al. [27] had mainly studies on the heat and mass transfer in air-bubbling enhanced vacuum membrane distillation. Jahangiri Mamouri et al. [28] experimentally investigated the effect of using thermosyphon heat pipes and vacuum glass on the performance of solar still whereas Dwivedi et al. [29] had mainly studied internal heat transfer coefficients in passive solar stills by different thermal models. Rahbar et al. [30] had mainly focus on experimental study of a novel portable solar still by utilizing the heat pipe and thermoelectric module whereas Kargar Sharif Abad et al. [31] had developed a novel integrated solar desalination system with a pulsating heat pipe.

In order to create a desalination machine for residential usage, Praveen Hunashikatti et al. [32] combined solar stills and evacuated tubes. Using renewable energy sources, Soteris A. Kalogirou [33] primarily investigated saltwater desalination. This report also reviews various desalination methods that employ renewable energy sources. The monthly averages of a sunlight hour and the monthly variance of total solar radiation are primarily covered by Ashok Kumar Rajput et al. [34]. Low-temperature, multi-effect desalination system was used by X.H Liu [35]. The article basically focussed on the energy deficit in Pune's district round the year. Thermal and economic analysis was conducted and it was found that the multi-effect desalination system proves to have better thermal performance. Hanane Ait Lahoussine Ouali et al. [36] had mainly studied integrated desalination unit. Morocco has experienced several years of drought, conventional water resources were insufficient to meet the needs of the population, which prompted the kingdom to seek other resources to ensure the supply of drinking water to this population. The most suitable solution was the desalination of seawater. In this context, the design and investigation of concentrated solar power (CSP) plant integrated with a desalination unit in Morocco are presented in this paper. The results mainly shows that the

lowest freshwater production cost is 0.84\$/m³ in Dakhla, followed by 0.98\$/m³ in Saidia, with the highest production cost at 1.20 \$/m³ in Agadir. Brenda Kalista [37] et al. had mainly studied current development and future prospect review of freeze desalination. In this paper few post treatment techniques have also been developed for the productivity of fresh water. This research trends in freeze desalination as an comprehensive review along with the future prospectus utility for freeze desalination process.

Qichao Sun [38] et al. had mainly investigated research process on the integration and optimal design of desalination process. In this paper various challenges in the integration and optimal design of desalination process have been presented. The paper mainly focus on valuable insights for the enhancing the efficiency of the desalination plants. The control strategies of RES desalination for reducing environments had been discussed in the paper. Argyris Panagopoulos et al. [39] had mainly studied comparative study on minimum and actual energy consumption for the treatment of desalination of brine. Developed model for calculating the minimum energy consumption. The results showed that an increase of 5 °C in feed temperature results in an increase of 1.6 % in the minimum energy consumption whereas the actual energy consumption is at least two times higher than the minimum energy consumption for the desalination brine treatment. Dongpeng Zhao et al. [40] had mainly developed a new energy analysis model of seawater desalination based on thermodynamics. The results showed that minimum energy consumption for the ideal desalination process decreases with increases salinity of sea water and operation temperature. Author had efforts should be made to develop the new desalination techniques which can operate under the low temperature. Kedar et al. [41-48] had mainly studied hybrid solar desaliantion system using evacuated tube collector and compound parabolic concentrator. He had developed mathematical model of hybrid system and their exprimetal results mainly shows that 27 – 28 L of soft water geneeated through enlarge 50 evacuated tube collector system. Tony et al. [49] had mainly studied recents developments for solar water desalination technology whereas Elfaqih et al. [50] had mainly reviewed solar photovoltaic power for desalination system. Their study mainly shows that solar photovoltaic system is the better option for solar desalination ststem for better future.

Most of the investigations had been focussed on single basin solar still as well as hybrid system solar still. The current research focuses on ETC and CPC hybrid desalination system with comparison of different reflecting materials. On the basis of reflectivity of the material used the overall production of soft water changes. Now-a-days soft water is demand of the society with ever increasing population. Due to this it is very essential to consider the production of soft water with economic and inexhaustible resources. Many investigators have studied solar desalination system experimentally, and numerically by numerous methods but it involved consumption of fossil fuel and electric power. Among the various

classes of materials, polymers were found to exhibit better results. Hence, this investigation was focussed on the production of soft water using polymeric materials as reflectors, in depth. The favourable factors of polymers in comparison of metals, in certain aspects of solar desalination systems, are weight, portability, corrosion resistance, flexibility, formability, cost effectiveness, thermal insulation, ease of manufacturing and dielectric properties. The current work presents comparison of the performance of the two polymeric materials, Mylar and Teflon, as reflector. It was observed that production of soft water with Mylar as reflecting material is higher in comparison with Teflon for ETC tubes. The reason behind this may be attributed to higher reflectivity of Mylar, which maximizes heat transmission and increases the output of soft water.

MATERIALS AND METHODS

Components Used During Experimentation

Single ended ETC is used as a main component of solar desalination system. Rows of parallel tubes are made of borosilicate glass for the ETC. The inner tube, also known as absorber, is coated with selective black coating. This coating absorbs solar energy efficiently and prevents radioactive heat losses. Air in the space of ETC tube was removed to create vacuum to check conductive and convective heat losses. ETC tubes are primarily suitable for cold ambient temperatures and are widely used for various industrial applications. The basic use of ETC tube is to generate steam from sea water. There is an insulated tank of water kept at 10 ft height and sea water from the tank flows to the evacuated tube solar collector by natural thermosyphon (natural flow). ETC is built to have a cold-water inlet, and when solar radiations fall on it, the water gets heated. The heated water moves up due to decrease in density. The ETC used in experiment gets heated up and condenses into the condenser. For single tube ETC experimentation surface condenser is not used because for the single tube ETC, naturally steam flows through the tubes and on cooling small droplets of water fall in the jar. The ETC used is circular in shape having length of 1500 mm and inner and outer diameter of 47 mm and 65 mm respectively. A copper tube is there in the Evacuated Tube Collector, through which water flows and gets heated and heated steam comes out from the other side. Use of copper tube inside ETC increases the rate

of heat transfer. Aluminum, silver, and steel can be used for this purpose. The various instruments used during experiments are explained in Table 1 along with their model name and range. For their entire experimentation four instruments were used such as pyranometer, infrared thermometer, water flow meter and anemometer. For all instruments measured parameter and range of the instrument mentioned in Table 1.

Experimental Uncertainty Analysis

The experimental measuring system, which included instruments such as a solar power meter, anemometer, water flow meter, thermocouple, and infrared thermometer, was not without uncertainties. Uncertainty measurements were carried out. The uncertainty in estimating daily performance efficiency and litres of soft water was roughly $\pm 2\%$ and $\pm 4\%$, respectively.

Various parameters such as ambient water temperature, wind speed, solar radiation intensity, evacuated tube temperature, and water vapor were measured at half-time intervals. The inflow mass flow rate of water was maintained constant. Figure 1 mainly shows that schematic of compound parabolic concentrator and evacuated tube collector used for solar desalination system. Both were mounted on stand with the proper basement to the horizontal. For the

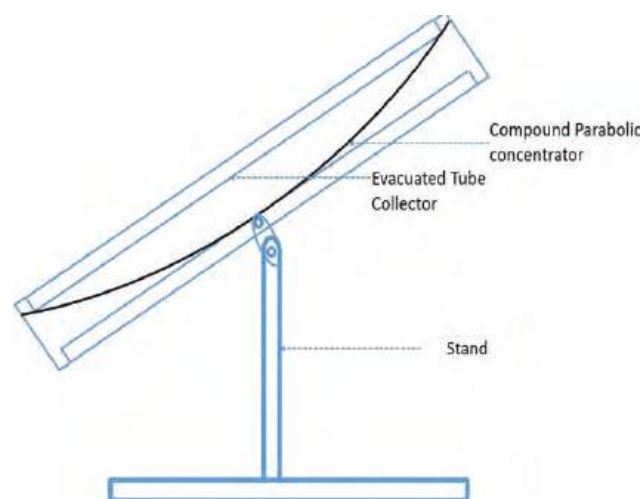


Figure 1. Schematic of CPC and ETC solar desalination system.

Table 1. Instruments used during experimentation with their specified range.

S. No.	Instrument used	Type/Model	Parameter	Range	Accuracy
1	Intensity of solar radiation	Pyranometer (Solar Power Meter)	KM-SPM-530	0 – 2000 W/m ²	+/- 0.2 W/m ²
2	Temperature	Infrared Thermometer	IRL-380	-50 – 380 °C	< 1000 °C +/- 2 °C
3	Flow rate of water	Water Flow Meter	STD	0.01 – 100 lpm	+/- 1% lpm
4	Wind Velocity	Anemometer	KM-910	0.0 - 45 m/s	+/- 3 m/s

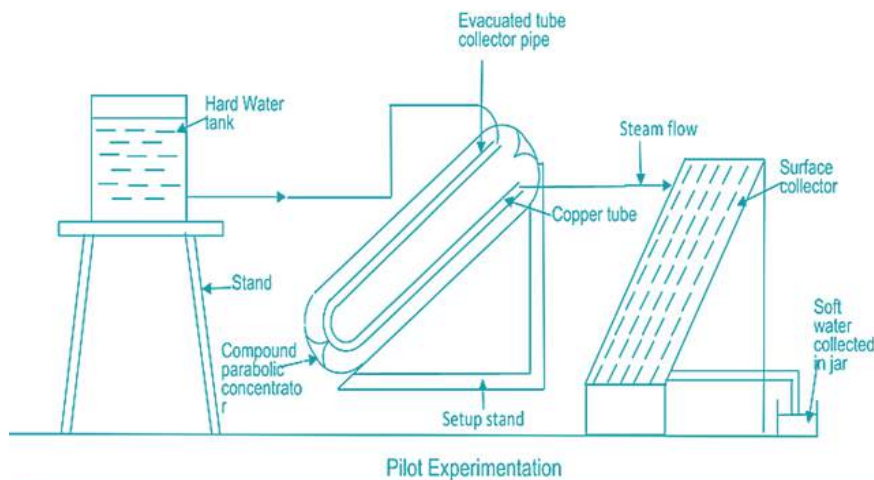


Figure 2. Complete schematic solar desalination system



Figure 3. Photographs of CPC and ETC solar desalination system.

stand there is an arrangement for angle variation depending on location. The entire setup mounted CPC - ETC combination was designed and implemented at terrace of Cummins College of Engineering for Women in Pune, India. Figure 2 mainly shows that complete schematic of solar desalination system whereas Figure 3 shows photographs of CPC and ETC solar desalination system. Figure 4 mainly shows that a view of compound parabolic concentrator during the actual experimentation whereas Figure 5 mainly shows that compound parabolic concentrator with mylar material. For the compound parabolic concentrator mylar material wrapped in such way that solar rays reflected back to ETC and within less time more amount of hot water generated.

Experimental Procedure

Several experiments were conducted in March 2023 on the campus of MKSS's Cummins College of Engineering for Women in Pune, Maharashtra, India. The detailed

experimental procedure is as follows.- Fill the hard water tank to the operational level. After that, change the test case parameters (temperature and flow rate). A compound parabolic concentrator is situated below a set of evacuated tube collectors. All measuring instruments will be ready to measure a variety of factors, including solar radiation intensity, wind velocity, thermometer, flow rate, and so on. Open the non-return valve so that hard water runs totally through copper tubes in the evacuated tube collector. Set and change valve timing based on water flow rate. In practical operation, water is heated inside copper tubes to produce steam. The valve is moved to another location in a copper tube that transports steam through the condenser. The steam stop valve is set to ensure a smooth flow of steam to the surface condenser. After steam condensation, saline/soft water is collected in a jar. The same technique should be performed for varied flow rates and distances between ETC and CPC, resulting in a variety of tests.

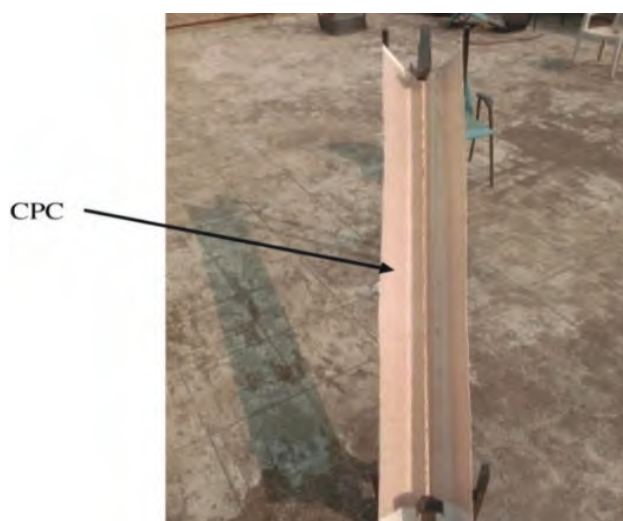


Figure 4. View of Compound Parabolic Concentrator during actual experiment.



Figure 5. Compound Parabolic Concentrator with Mylar.

RESULTS AND DISCUSSION

The results and discussion section mainly include results of the various experiments for the solar desalination system and accordingly discussions are presented in this section. In this investigation only the polymeric materials Mylar and Teflon have been used as reflecting material.

There are various advantages of using polymeric materials, as elaborated above.

During the experiment, the hourly change of the measured parameters such as ambient temperature, outside evacuated tube temperature, outlet water temperature, and CPC temperature, as well as the intensity of solar radiation, wind velocity, and the rate of freshwater generation, were

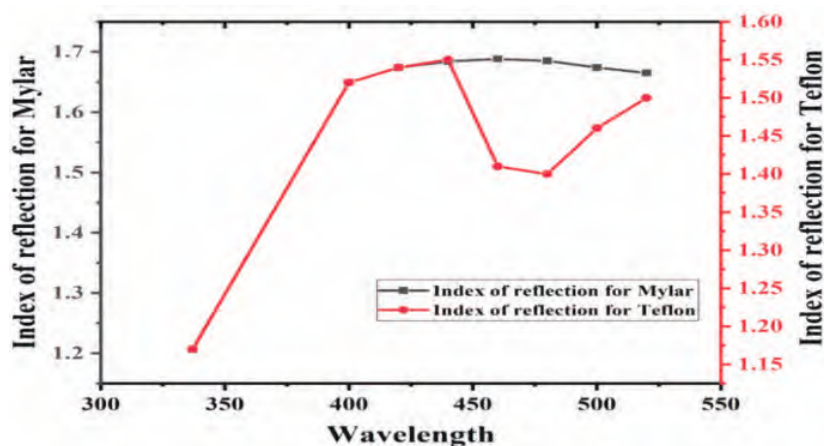


Figure 6. Variation of reflection from Mylar and Teflon with wavelength.

continuously monitored over a one-month period. In this investigation, the materials used for CPC are Mylar and Teflon of 2 mm thickness. Fig. 6 depicts index of reflection for Mylar and Teflon with wavelength. It was observed that with increase in the wavelength, the index of reflection in Mylar is in the range of 1.6 -1.7, whereas in the case of Teflon, with increase in the wavelength there is decrease in the index of reflection. This shows that the index of reflection of Mylar is higher than that of Teflon. It was observed that from Mylar the reflections are more which get transferred to ETC tubes and finally to copper tubes which enhances the formation of steam and thereby leads to higher amount of soft water generation from sea water. Index of refraction in case of Mylar is greater than Teflon for the entire wave length 450 – 525 μm which shows more possibility for generation of drinkable water.

Fig. 7 shows index of absorption for Mylar and Teflon with wavelength. In the case of Mylar it was observed that

with increase in the wavelength, in the range of 0.004 to 0.010, there is decrease in the index of absorption whereas it decreases above the wave length of nearly 450 μm in the case of Teflon. During experimentation it was observed that in Teflon the index of absorption is more than that of Mylar, hence less amount of heat is transferred to ETC tubes and then to copper tubes, from the Teflon reflector. This results in relatively less amount of steam formation for the generation of soft water from sea water. There was deviation observed in Teflon as compared to Mylar shows continuously decreasing nature for the index of absorption.

The intensity of solar radiation and production of soft water in litres with Mylar with time is depicted in Fig.8. It was observed that intensity of solar radiation gradually increased from 400 - 930 W/m^2 during the time interval of 12:30 PM to 1:30 PM. In the afternoon time soft water generation rate is very high. At the end of the day 3.5 litres of soft water was collected which clearly explains that higher

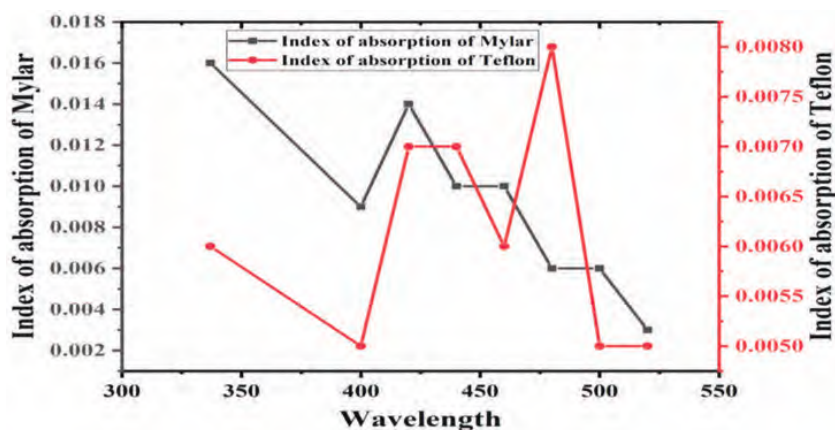


Figure 7. Variation in index of absorption of Mylar and Teflon with wavelength.

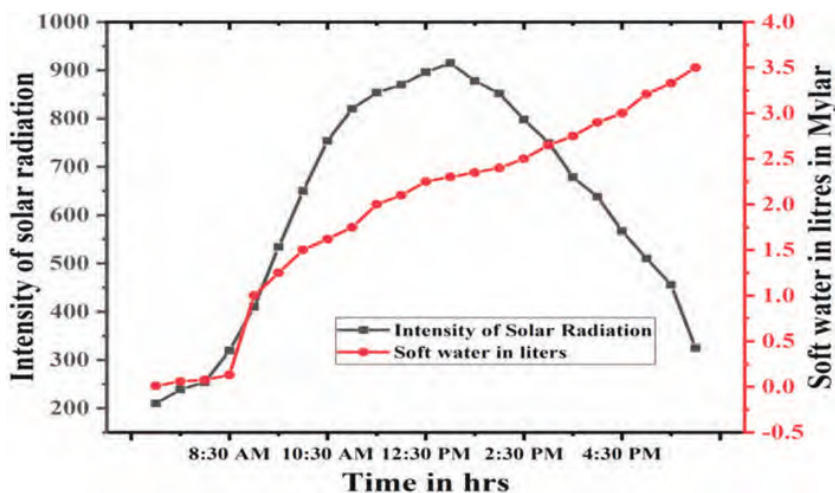


Figure 8. Variation in intensity of solar radiation and output of soft water in litres with time the time of exposure, in case of Mylar reflector.

index of reflection and smaller index of absorption of Mylar lead to higher amount of soft water generated.

The Fig. 9 shows intensity of solar radiation and soft water in litres for Teflon versus time. It is clearly observed that intensity of solar radiation gradually increases from 400 - 930 W/m² at the time interval of 12:30 PM to 1:30 PM. In the afternoon generation rate of soft water is very high. At the end of the day 3.2 litres of soft water was collected. The reason behind this is less index of reflection of Teflon and more index of absorption. Thus, on comparing the CPC materials it was noted that there is higher amount of soft water generation in case of Mylar in comparison with Teflon. Initially the formation of soft water rate is very slow in morning session, thereafter in afternoon session the formation rate of soft water is very high due to intensity of solar radiation increases continuously.

Fig. 10 shows soft water production in litres with Mylar and Teflon reflectors with the time of exposure. It

may clearly be observed that soft water produced by Mylar reflectors higher in comparison with that from Teflon reflector Throughout the entire day using Mylar as coating material, production of soft water is 3.5 litres per day whereas using Teflon as a coating material there is production of 3.2 litres of soft water. This result clearly indicates that with Mylar the rate of soft water production is more in comparison with Teflon. The reason behind this enhancement from Mylar is its higher index of reflection in comparison with Teflon. Hence, Mylar gives better result in comparison with Teflon.

A table of comparison has been shown for the production of soft water along with reflectivity values for Aluminium foil, Mylar and Teflon in Table 2. It clearly shows that Mylar as a candidate reflecting material with respect to the other two, and it also shows better results for polymeric material as compared to metallic material.

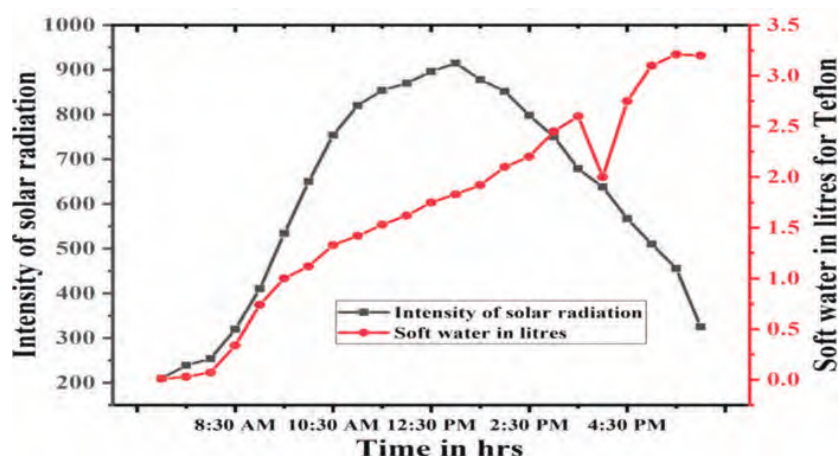


Figure 9. Intensity of solar radiation and soft water in litres for Teflon versus time.

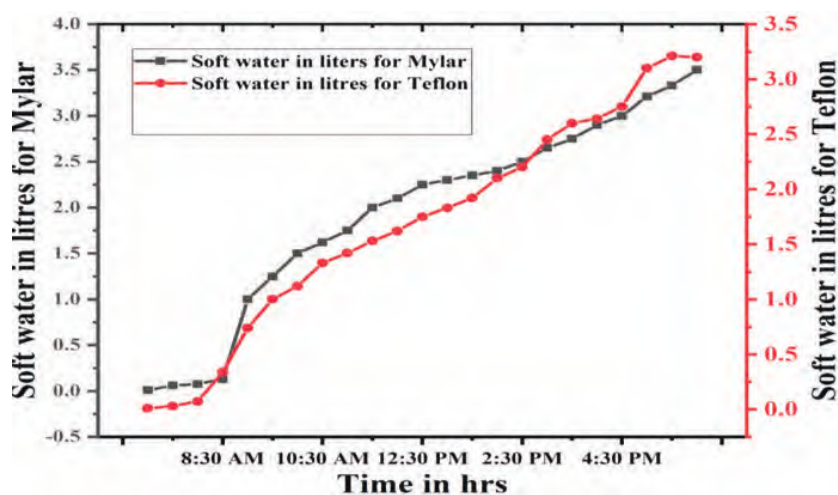


Figure 10. Soft water output in litres with Mylar and Teflon reflectors with time of exposure.

Table 2. Comparison of reflectivity and amount of soft water production for Mylar, Teflon and Aluminium foil as reflecting material used in hybrid ETC and CPC solar desalination system

Sr. No.	Reflector material	Reflectivity	Soft water (in litres)
1	Mylar	98	3.50
2	Teflon	94	3.00
3	Aluminum foil	90	2.15

Table 3. Hybrid Solar system comparison of reflectivity and amount of soft water production

Sr. No.	Hybrid solar system	Soft water (in litres)
1	Solar wind hybrid energy system	0.50
2	Solar still coupled with ETC system	3.28

Based on the comparison between Table 2 and Table 3, it is evident that the materials Mylar, Teflon, and Aluminium foil yield fresh water amounts of 3.5 L, 3.00 L, and 2.15 L per day, respectively. On the other hand, the solar wind hybrid energy system and the solar still coupled with ETC system produce fresh water amounts of 0.50 L and 3.28 L per day, respectively. This comparison demonstrates that the hybrid solar groundwater desalination system is better suited for the production of fresh and drinking water. [11, 32]. Table 3 shows that as per the comparison point of view solar wind hybrid energy system gives 0.50 l soft water per day whereas solar still coupled with ETC system gives 3.28 l soft water per day. This results shows mylar and teflon gives better performance results in hybrid solar desalination system.

Limitation for the entire study

a) Only two materials, Mylar and Teflon, were used during the whole study of the solar desalination system. b) The system is non-functional for the entire heavy rainy season due to insufficient sun radiation on rainy days. c) The entire study exclusively utilizes only groundwater for testing purposes [6].

Environmental Benefit of the Study

The environmental impacts (EIs) of the desalination process vary greatly depending on the characteristics of the feedwater and the specific desalination method being

used. This experiment aims to reduce carbon emissions in the desalination process by using Mylar and Teflon as reflector materials. By doing so, the experiment will enable a specific organisation to get advantages under the Clean Development Mechanism (CDM). A society that prioritises a greener environment would benefit from reduced pollution and the implementation of eco-friendly systems [37].

Environmental Impact

It is known that technologies made by humans have significant impact on the environment. Most of the times, these effects lead to unfavorable consequences. Although desalination provides a consistent water supply regardless of the natural freshwater ecosystems and delivers relief from water stress and scarcity, many concerns are still expressed due to the potential negative repercussions. This is despite the fact that desalination supports human health and socio-economic welfares. Desalination has had a variety of effects on the environment, each of which is distinct and occurs on a separate level. It is possible to describe desalination that interacts with the world around it through the use of inputs and outputs through as a process straightforward explanation. The terms “geosphere,” “hydrosphere,” “biosphere,” and “atmosphere” are typically used to refer to these interactions, which are inputs from and outputs to nature at various environmental subsystems, such as land, water, living things, and air [38].

Table 4. Specification of Mylar and Teflon

Sr. No.	Property	Mylar	Teflon	Unit
1	Tensile strength	28,000	3,900	PSI
2	Tensile modulus	7,40,000	80,000	PSI
3	Surface roughness	38	30	nm
4	Density	1.39	2.16	g/cm ³

Table 5. Economic analysis of hybrid Solar desalination system (ETC – CPC)

Sr. No.	Quantity of the system	Cost
1	One Single ETC tube	5.83
2	One single CPC for ETC	17.50
3	Stand for the entire setup	7.29
4	Cu tubes for Insides ETC tube	14.58
5	Hard/Soft water tank	11.67
6	Condenser for desalination system	20.42
7	Entire plumbing/ pipe line work	14.58
8	Valves and labor cost	21.88
	Total Cost (USD)	113.75

Specifications of Mylar and Teflon

Economic Analysis of Single Hybrid Solar Desalination System (ETC+CPC)

The economic evaluation of the present work is undertaken for life cost analysis. The entire cost of the current study is primarily made up of fixed costs, production costs, and running costs as is listed in Table 4. The basis of these costs is the real purchase price of the apparatus. The cost basis that we have adopted is for the year 2023 in US dollars.

For the economic analysis shows that the total cost associated with the hybrid energy system is USD 113.75

CONCLUSION

In the current investigation compound parabolic concentrator was employed for desalination purpose and various thermal properties with respect to Mylar and Teflon as coating material have been analysed. During this investigation mylar and teflon material used over CPC and above ETC, single effect boiling approach is suggested for producing drinkable water. The major conclusions from the comparative study are as follows:

- Mylar and Teflon belong to same class of materials (polymers). The comparison between the two is related to their thermal properties. It shows that Mylar has higher index of reflection and less index of absorption as compared to Teflon
- For maximum production of soft water the coating thickness for Teflon should be of 3 mm as observed from the experiment.
- In case of Mylar the coating thickness should be of 2 mm for the maximum production of soft water.
- Since the index of reflection is higher for Mylar in comparison with Teflon, the production of soft water is higher for Mylar.
- The overall experimental procedure is cheap, inexhaustible, safe and without power consumption, and is useful from the society point of view.

NOMENCLATURE

I_g	Intensity of solar radiation, W/m^2
t	Time, hrs
T	Temperature, $^{\circ}C$
V	Wind velocity, m/sec
n	Refractive index
u	Velocity of the medium, m/sec.
ETC	Evacuated Tube Collector
CPC	Compound Parabolic Concentrator

ACKNOWLEDGEMENTS

The author would like to thank Head of the Department, Dr.Ajit Bhosale, and Domain Expert Dr.Anand Bewoor, Dr.G.Murali, Dr.K.Arul Raj, Dr.Vakil Singh for their valuable guidance and comments. Also, the authors would like to acknowledge as a facilitator for data collection Cummins College of Engineering for Women, Karvenagar Pune - 411052.

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Ghorbani B, Mehrpooya M, Sadeghzadeh M. Developing a tri-generation system of power, heating, and freshwater (for an industrial town) by using solar flat plate collectors, multi-stage desalination unit, and Kalina power generation cycle. *Energy Convers Manag* 2018;165:113-126. [CrossRef]

- [2] Baci AB, Salmi M, Menni Y, Ghafourian S, Sadeghzadeh M, Ghalandari M. A new configuration of vertically connecting solar cells: solar tree. *Int J Photoenergy* 2020;2020:8817440. [\[CrossRef\]](#)
- [3] Ahmadi MH, Sayyaadi H, Mohammadi AH, Barranco-Jimenez MA. Thermo-economic multi-objective optimization of solar dish-Stirling engine by implementing evolutionary algorithm. *Energy Convers Manag* 2013;73:370-380. [\[CrossRef\]](#)
- [4] Kalogirou SA. Seawater desalination using renewable energy sources. *Prog Energy Combust Sci* 2005;31:242-281. [\[CrossRef\]](#)
- [5] Ali MT, Fath HES, Armstrong PR. A comprehensive techno-economical review of indirect solar desalination. *Renew Sustain Energy Rev* 2011;15:4187-4199. [\[CrossRef\]](#)
- [6] Kedar SA, Bewoor AK, Murali G, Kumar R, Sadeghzadeh M, Issakhov A. Effect of reflecting material on CPC to improve the performance of hybrid groundwater solar desalination system. *Int J Photoenergy* 2021;2021:6675236. [\[CrossRef\]](#)
- [7] Moungar H, Ahmed A, Youcef S, Aabdelkrim H. Immersed fins influence on the double slope solar still production in south Algeria climatic condition. *Int J Heat Technol* 2017;35:1065-1071. [\[CrossRef\]](#)
- [8] Sathyamurthy R, Nagarajan P, Edwin M, et al. Experimental investigations on conventional solar still with sand heat energy storage. *Int J Heat Technol* 2016;34:597-603. [\[CrossRef\]](#)
- [9] Morad MM, El-Maghawry HAM, Wasfy KI. Improving the double slope solar still performance by using flat-plate solar collector and cooling glass cover. *Desalination* 2015;373:1-9. [\[CrossRef\]](#)
- [10] Bhambare PS, Majumder MC, Sudhir CV. Solar thermal desalination: a sustainable alternative for Sultanate of Oman. *Int J Renew Energy Resour* 2018;8:733-751.
- [11] Al-Nimr MA, Kiwan SM, Talafha S. Hybrid solar wind water desalination system. *Desalination* 2016;395:33-40. [\[CrossRef\]](#)
- [12] Trieb F, Müller-Steinhagen H. Concentrating solar power for seawater desalination in the Middle East and North Africa. *Desalination* 2008;220:165-183. [\[CrossRef\]](#)
- [13] Sayed ET, Olabi AG, Elsaid K, Al Radi M, Alqadi R, Abdelkareem MA. Recent progress in renewable energy based-desalination in the Middle East and North Africa (MENA) region. *J Adv Res* 2023;48:125-156. [\[CrossRef\]](#)
- [14] Grosu L, Mathieu A, Rochelle P, Feidt M, Ahmadi MH, Sadeghzadeh M. Steady state operation exergy-based optimization for solar thermal collectors. *Environ Prog Sustain Energy* 2020;39. [\[CrossRef\]](#)
- [15] Sadeghzadeh M, Ahmadi MH, Kahani M, Sakhaeinia H, Chaji H, Chen L. Smart modeling by using artificial intelligent techniques on thermal performance of flat-plate solar collector using nanofluid. *Energy Sci Eng* 2019;7:1649-1658. [\[CrossRef\]](#)
- [16] Olia H, Torabi M, Bahiraei M, Ahmadi MH, Goodarzi M, Safaei MR. Application of nanofluids in thermal performance enhancement of parabolic trough solar collector: state-of-the-art. *Appl Sci* 2019;9:463. [\[CrossRef\]](#)
- [17] Loni R, Kasaeian A, Shahverdi K, Askari Asli-Ardeh E, Ghobadian B, Ahmadi MH. ANN model to predict the performance of parabolic dish collector with tubular cavity receiver. *Mech Ind* 2017;18:408. [\[CrossRef\]](#)
- [18] Jilte R, Ahmadi MH, Kalamkar V, Kumar R. Solar flux distribution study in heat pipe cavity receiver integrated with biomass gasifier. *Int J Energy Res* 2020;44:7698-7712. [\[CrossRef\]](#)
- [19] Rafiei R, Loni R, Ahmadi MH, Najafi G, Bellos E, Rajaei F, et al. Sensitivity analysis of a parabolic trough concentrator with linear V-shape cavity. *Energy Sci Eng* 2020;8:3544-3560. [\[CrossRef\]](#)
- [20] Sampathkumar K, Arjunan TV, Pitchandi P, Senthilkumar P. Active solar distillation - a detailed review. *Renew Sustain Energy Rev* 2010;14:1503-1526. [\[CrossRef\]](#)
- [21] Siddiqui FR, Elminshawy NAS, Addas MF. Design and performance improvement of a solar desalination system by using solar air heater: experimental and theoretical approach. *Desalination* 2016;399:78-87. [\[CrossRef\]](#)
- [22] Kabeel E, El-Said EMS. A hybrid solar desalination system of air humidification, dehumidification and water flashing evaporation: part II. Experimental investigation. *Desalination* 2014;341:50-60. [\[CrossRef\]](#)
- [23] Sapre M, Auti A, Singh T. Design and manufacturing of absorber for solar desalination system. *Appl Mech Mater* 2013;446-447:716-720. [\[CrossRef\]](#)
- [24] Zheng H, Chang Z, Chen Z, Xie G, Wang H. Experimental investigation and performance analysis on a group of multi-effect tubular solar desalination devices. *Desalination* 2013;311:62-68. [\[CrossRef\]](#)
- [25] Khalil S, El-Agouz SA, El-Samadony YAF, Abdo A. Solar water desalination using an air bubble column humidifier. *Desalination* 2015;372:7-16. [\[CrossRef\]](#)
- [26] Ahmed MI, Hrairi M, Ismail AF. On the characteristics of multistage evacuated solar distillation. *Renew Energy* 2009;34:1471-1478. [\[CrossRef\]](#)
- [27] Wu C, Li Z, Zhang J, Jia Y, Gao Q, Lu X. Study on the heat and mass transfer in air-bubbling enhanced vacuum membrane distillation. *Desalination* 2015;373:16-26. [\[CrossRef\]](#)
- [28] Jahangiri Mamouri S, Gholami Derami H, Ghiasi M, Shafii MB, Shiee Z. Experimental investigation of the effect of using thermosyphon heat pipes and vacuum glass on the performance of solar still. *Energy* 2014;75:501-507. [\[CrossRef\]](#)
- [29] Dwivedi VK, Tiwari GN. Comparison of internal heat transfer coefficients in passive solar stills by different thermal models: an experimental validation. *Desalination* 2009;246:304-318. [\[CrossRef\]](#)

- [30] Rahbar N, Esfahani J. Experimental study of a novel portable solar still by utilizing the heat pipe and thermoelectric module. *Desalination* 2012;284:55-61. [\[CrossRef\]](#)
- [31] Kargar Sharif Abad H, Ghiasi M, Jahangiri Mamouri S, Shafii MB. A novel integrated solar desalination system with a pulsating heat pipe. *Desalination* 2013;311:206-210. [\[CrossRef\]](#)
- [32] Hunashikatti PT, Suresh KR, Prathima B. Development of desalination unit using solar still coupled with evacuated tubes for domestic use in rural areas. *Curr Sci* 2014;107:1683-1693.
- [33] Kalogirou SA. Seawater desalination using renewable energy sources. *Prog Energy Combust Sci* 2005;31:242-281. [\[CrossRef\]](#)
- [34] Rajput AK. Utility-based estimated solar radiation at destination Pune, Maharashtra, India. *Int J Pure Appl Sci Technol* 2012;13:19-26.
- [35] Liu XH. Thermal and economic analyses of solar desalination system with evacuate tube collector. *Sol Energy* 2013;93:144-150. [\[CrossRef\]](#)
- [36] Ouali HAL, Soomro MI, Touili S, Merrouni AA. Assessment of direct contact membrane distillation system driven by parabolic trough collector plant technology for electricity and freshwater production: feasibility and benchmark under different Moroccan climates. *Sustain Mater Technol* 2024;39:e00818. [\[CrossRef\]](#)
- [37] Kalista B, Shin H, Cho J, Jang A. Current development and future prospect review of freeze desalination. *Desalination* 2018;447:167-181. [\[CrossRef\]](#)
- [38] Sun Q, Mao Y, Wu L. Research progress on the integration and optimal design of desalination process. *Sep Purif Technol* 2024;337:126423. [\[CrossRef\]](#)
- [39] Panagopoulos A. A comparative study on minimum and actual energy consumption for the treatment of desalination brine. *Energy* 2020;212:118733. [\[CrossRef\]](#)
- [40] Zhao D, Deng S, Shao Y, Zhao L, Lu P, Su W. A new energy analysis model of seawater desalination based on thermodynamics. *Energy Procedia* 2019;158:5472-5478. [\[CrossRef\]](#)
- [41] Kedar S, Kumaravel AR, Bewoor AK. Experimental investigation of solar desalination system using evacuated tube collector. *Int J Heat Technol* 2019;37:527-532. [\[CrossRef\]](#)
- [42] Kedar SA, Bewoor AK, Raj K. Design and analysis of solar desalination system using compound parabolic concentrator. *IOP Conf Ser Mater Sci Eng* 2018;455:012063. [\[CrossRef\]](#)
- [43] Kedar SA, Bewoor AK, Madhusudan S. Solar desalination system using evacuated tube collector and compound parabolic concentrator - theoretical approach. In: *Natl Conf Adv Electr Eng Energy Sci*; 2016. p. 60-62.
- [44] Kedar SA, Murali G, Bewoor AK. Mathematical modelling and analysis of hybrid solar desalination system using evacuated tube collector (ETC) and compound parabolic concentrator (CPC). *Math Model Eng Probl* 2021;8:45-51. [\[CrossRef\]](#)
- [45] Kedar SA, Raj KA, Bewoor AK. Performance analysis of hybrid solar desalination system using ETC and CPC. *SN Appl Sci* 2019;1:1-17. [\[CrossRef\]](#)
- [46] Kedar SA, Murali G, Bewoor AK. Effective hybrid solar groundwater desalination in rural areas. *Int Trans J Eng Manag Appl Sci Technol* 2021;12:1-10. [\[CrossRef\]](#)
- [47] Kedar SA, Raj KA, Bewoor AK. Thermal analysis of solar desalination system using evacuated tube collector. *AIP Conf Proc* 2018;2039:020061. [\[CrossRef\]](#)
- [48] Kedar S, Bewoor A, Murali G, More GV, Roy A. Thermal analysis of sea water hybrid solar desalination system - an experimental approach. *Int J Heat Technol* 2024;42:1349-1358. [\[CrossRef\]](#)
- [49] Tony MA, Nabwey HA. Recent advances in solar still technology for solar water desalination. *Appl Water Sci* 2024;14:147. [\[CrossRef\]](#)
- [50] Elfaqih AK, Elbaz A, Akash YM. A review of solar photovoltaic-powered water desalination technologies. *Sustain Water Resour Manag* 2024;10:123. [\[CrossRef\]](#)



Research Article

Thermal design and CFD simulation of an internal heat exchanger with R1234yf as an alternate refrigerant to R134a

B. K. PATEL^{1,2,*}, A D. PAREKH²

¹Department of Mechanical Engineering, Government Engineering College, Valsad, Gujarat, 396001, India

²Department of Mechanical Engineering, Sardar Vallabhbhai Patel National Institute of Technology, Surat, Gujarat, 395007, India

ARTICLE INFO

Article history

Received: 18 October 2024

Accepted: 06 November 2024

Keywords:

Automobile Air Conditioning;

COP; CFD; R1234yf; R134a;

Internal Heat Exchanger

ABSTRACT

The most widely utilized refrigerant in car air conditioners today is still R134a, particularly in developing nations, but its global warming potential is very high (1430). The use of high global warming potential working substances is progressively prohibited and restricted by the F-gas act and the Kigali Amendment in developing and developed nations. R1234yf has been the subject of thermodynamic research as a potential replacement for R134a in vehicle air conditioning systems. Compared to R134a, R1234yf performs somewhat inferior, that can be improved by incorporating an internal heat exchanger into the existing system. Refereeing to the literature available, computational fluid dynamics analysis of the internal heat exchanger for an automobile air conditioning system with refrigerant R1234yf is rarely observed. Hence the novel concept of computational fluid dynamics analysis of thermally designed internal heat exchanger is focused here. This study analyzes the thermal design of an internal heat exchanger and its impact on the coefficient of performance for R1234yf and R134a. And computational fluid dynamics analysis of the thermal designed internal heat exchanger is performed to finalize the dimension of the internal heat exchanger. For similar cooling capacity, the system with refrigerant R134a performance is not much affected by the application of an internal heat exchanger. Its COP increases from 3.636 to 3.676, i.e., only 1.09%. While the performance gap of the system with refrigerant R1234yf as compared to the system with R134a without an internal heat exchanger is 5.17%, while the gap is decreased up to 3.16% with an internal heat exchanger. Identical results are obtained in computational fluid dynamics analysis, with an increment in the internal heat exchanger length, heat transfer increases as well, and the outlet temperature meets the necessary level within a tolerable pressure drop.

Cite this article as: Patel BK, Parekh AD. Thermal design and CFD simulation of an internal heat exchanger with R1234yf as an alternate refrigerant to R134a. J Ther Eng 2025;11(5):1339–1354.

*Corresponding author.

*E-mail address: bkpgcev@gmail.com, adp@med.svnit.ac.in

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkilic



INTRODUCTION

Scientific study now prioritizes understanding how refrigerant affects the environment. Global warming potential (GWP) reports that the environmental emissions of several gases from industries are comparable to carbon dioxide. The amount of heat on Earth is greatly increased by high greenhouse gas concentrations. Third-generation HFC refrigerants, which have a high GWP and zero ODP, are now used in the automotive industry. Finding an R134a substitute in accordance with the Kyoto and Montreal protocols is therefore required to solve the issue of excessive GWP [1]. Environmentally acceptable alternative refrigerants are gradually replacing hazardous refrigerants as part of regulatory efforts to reduce their use. In addition, it preserves the ozone layer and lowers greenhouse gas emissions. Additionally, the substitute refrigerant for the current car air conditioner needs to be affordable and safe. Some nations concentrate on cutting emissions through the use of appropriate refrigerants and good equipment design. The Kigali Amendment [2], which aims to gradually phase out HFCs by reducing their production and usage, was accepted at the 28th conference of parties to the Montreal Protocol. By 2050, the Kigali Amendment calls for an over 85% reduction in the world's HFC usage. According to research, India's population demand prediction for passenger cars could rise by up to 9% between 2017 and 2038 [3]. According to the study, a car's lifespan is estimated to be 15 years, and there should only be 200 passenger automobiles for every 1,000 people. However, taking into account the advancements in cooling system technology, based on different system leak rates, the refrigerant demand scenario is evaluated. The refrigerant consumption by 2038 is estimated by the India Cooling Action Plan to be between 1900 and 24000 MT. A significant contributor to greenhouse gas emissions is the use of mobile air conditioning systems. According to Yang et al. [4], up to 20% of the fuel energy used to run the vehicle can be used by the air conditioning system. Shaikh et al. [5] state that the refrigerant R1234yf is a good substitute for R134a because both refrigerants have similar energy consumption, volumetric cooling capacity, COP and refrigerating effect. Satapathy [6] et al. performed comparative analysis using an ozone friendly mixture of refrigerant R23 and R507A for the auto cascade system

and the cascade system to achieve better performance with environmental aspects. Table 1 summarizes the comparison of the thermophysical characteristics of refrigerants R134a and R1234yf [7-9].

Direk et al. [11] investigated the energy and exergy of refrigerant R1234yf in vehicle air conditioner. It was shown that the internal heat exchanger reduced energy destruction per cooling capacity by 13%–16% and raised COP by 4%–6%, respectively. An experimental investigation was conducted. And a two-pipe heat exchanger was chosen. Cooling capacity, COP and overall energy destruction with respect to cooling capacity were calculated and presented as functions of constant temperatures for condensation or evaporation based on experimental data. The results showed that the R1234yf with internal heat exchanger had a mean increase in cooling capacity of 11% to 12%, an improvement in COP value of 4% to 6%, an increase in compressor power of 5% to 8%, an increase in volumetric efficiency value of 4% to 5%, and an increase in isentropic efficiency value of 3% to 4%. The average reduction in total energy destruction per cooling capacity value was between 13% and 16%. And come to the conclusion that the system with R1234yf can benefit from an internal heat exchanger in terms of cooling and COP.

Hmood et al. [12] suggested that R1234yf is a suitable drop-in substitute for R134a in refrigeration systems. With few small adjustments to the AACs system, R1234yf can also be viewed as a viable long-term environmentally beneficial solution because it is non-flammable. Sharif et al. [13] concluded that R1234yf has somewhat lower heat transfer coefficients than R134a, but it makes up for it with smaller pressure drops. Furthermore, it is possible to replace R1234yf directly in current systems with the least amount of adjustment. There is potential for improving the efficiency of vapour compression systems with R1234yf refrigerant by integrating IHX and LSHX. Moreover, it has been discovered that employing IHX reduces energy usage and raises the COP of R1234yf systems.

Strong effects on heat transport were noted by Karademir et al. [14] using R1234yf refrigerant, which has minimal volatility and low GWP. It was proven that as the quality of the vapor increased, so did the heat transfer coefficient. Agrawal et al. [15] concluded that R1234yf's performance parameter values are lower than those of HFC-R134a. It can

Table 1. Comparison of property between R134a and R1234yf [Alkan et al [10], with permission from Yıldız Technical University Press]

Refrigerant	Chemical composition	Boiling point [°C]	Critical Temperature [°C]	Critical pressure [MPa]	ODP	GWP	Safety class (ASHRAE34)	Atmospheric lifetime (year)
R134a	CH ₂ FCF ₃	-26.07	101.1	4.059	0	1430	A ₁	14
R1234yf	CF ₃ CF=CH ₂	-29.45	95	3.382	0	4	A ₂ L	<0.05 ys) 11 days

replace R134a quite effectively, though, as the differences are negligible and it has eco-friendly qualities.

Daviran et al. [16] performed simulated analysis on R134a and R1234yf. The lower discharge temperature and pressure ratio for R1234yf were observed. R1234yf experiences a smaller pressure drop during the condensation and evaporation process than R134a. So, a system of R1234yf with fewer thickness pipe, reduces price and improves performance. For R1234yf, COP is 1.3–5% less than that of R134a for similar cooling capacity, while it is 18% higher for constant mass flow rate. Lin et al. [17] recommend R1234yf and a blend of R1234yf and R125 as a very good substitute for R134a because of their comparable cooling capacities and COPs, which are only 5.5% lower than those of R134a. Similarities in the discharged temperature, capacity, and coefficient of performance of refrigerants R134a, R1234yf, and the combination of R134a and R1234yf were noted by Lee et al. [18] on a heat pump bench testing under varying weather conditions. Compared to R134a, the initial charge needed for R1234yf and combination is 11% less. When the percentage of R134a in the mixture reaches 10%, it becomes non-flammable.

Patel et al. [19] estimated the cooling load of an automobile air conditioning system for local climate conditions and performed energy and exergy analysis for R134a and R1234yf, concluded that as the evaporating temperature rises, COP and EDR rise as well. The percentage difference in EDR for the R134a and R1234yf ranges from 6.79% to 2.87 percent when the evaporation temperature is changed from -10°C to 10°C , and the difference for COP ranges from 2.44 percent to 4.78 percent. The exergy efficiency falls as the temperature of evaporation rises. At evaporating temperatures of -10°C , the R1234yf exhibits a 4.78 % lower value of exergy efficiency than the R134a; the discrepancy narrows to 2.43 % once evaporating temperatures have reached 10°C . At a fixed evaporating temperature of 0°C , COP drops as the condensation temperature rises from 30 to 60°C . At 30°C , the COP of the system with the R1234yf system is 1.96 % less than the R134a system, and it increases to 9.2% at 60°C .

Wantha [20] investigated the relationship between COP and the coefficient of heat transfer as influenced by annular space, the heat exchanger's length and efficiency, and pressure drop at 6.4°C and -6.4°C for evaporation and 46°C for condensation. At the same operating temperatures of 46°C for condensation and 0°C for evaporation, while utilizing an internal heat exchanger, the COP of R134a increases by 2.11% and that of R1234yf increases by 3.78%. The tube in tube internal heat exchanger for R1234yf has a coefficient of heat transfer is 11–17% lower than that of R134a for an identical annular diameter ratio. Kwon et al. [21] made an experimental investigation of the internal heat exchanger's heat transfer properties in a CO_2 heat pump system. The longer the internal heat exchanger, the greater the effectiveness, capacity, and pressure decrease. Approximately twice as much capacity is available in micro-channel IHX as in

coaxial IHX. Compared to the hot side, the cold side has a greater pressure reduction.

Alpaslan et al. [22] discovered that R1234yf produced a 9.3–22.3% higher refrigerant mass flow rate and a 1.1 – 3.5°C higher temperature of conditioned air. However, in comparison to R134a, it also resulted in a 4.7 – 16.1°C lower compressor outlet temperature, a 5.5–11.6% worse COP, and a 0.4–10.9% lower refrigeration capacity. Additionally, the R1234yf system's components often destroyed more energy, and overall, the R1234yf system's exergy destruction rate per unit refrigeration capacity was 4.1–35.3% higher than that of the R134a system. Alpaslan et al. [23] examined the impact of compressor speed and the temperature range of the air stream at the evaporator and condenser inlet on the exergy and energy evaluation of the car air conditioning system. The condition for outside temperature is produced by 1.78 kW and 5.4 kW heaters in the condenser and evaporator ducts, respectively. Additionally, it is discovered that because R1234yf has a lower liquid density than R134a, its initial charge is 10% lower.

Direk et al. [24] compared R1234yf and R134a's performances by varied air stream temperatures and compressor speeds. According to the findings, the system incorporated with R1234yf had a lower cooling ability and COP of 17.1% and 12.4%, respectively, than the one with R134a. But with the IHX together with R1234yf system, the system's cooling capacity value and COP increased dramatically by 7.9% and 4.1%, respectively. The cooling capacity and coefficient of performance of the system using R1234yf were ontained to be 13.9% to 20.4% and 7.5% to 16.5% lower, respectively, than those of the system using R134a. The experimental techniques were replicated using a double pipe heat exchanger. According to the results, at 27°C and 35°C , respectively, the cooling capacity of R1234yf was enhanced by 6.3% to 8.6% at 27°C and 6.4% to 9.9% at 35°C . Additionally, for air stream temperatures of 27°C and 35°C , respectively, the system's COP was raised by 2.8% to 7.4% and 2.4% to 4.8%. Direk et al. [25] examined the performance of R134a and R1234yf based on the internal heat exchanger efficiency is between -20°C and 0°C for evaporation and 40°C and 50°C for condensation. and discovered that when efficacy increases, so does the compressor discharge temperature. Ultimately, the critical effectiveness value to achieve the same COP with R1234yf was discovered to be 50%.

According to an experimental investigation conducted by Cho et al. [26], refrigerant R1234yf has a low COP and cooling capacity of 4.5% and 7%, respectively. When an IHX is used, performance decreases by 2.9% to 1.8%. It is possible to achieve satisfactory cooling capacity by operating under varied operating conditions and increasing the COP to 4.6%. Babiloni et al. [27] calculated that in the absence of IHX, R1234yf's volumetric efficiency is 4% lower, its cooling capacity is 9% lower, and its coefficient of performance is roughly 7% worse. They recommend that the R1234yf system's COP disparities be decreased by using an internal heat exchanger. In their numerical analysis of three fluid

heat exchangers, in both parallel flow and counter flow systems, Mohapatra et al. [28] found that the total heat transfer coefficient and the efficiency of heat transfer via the helical tube side fluid to the outer side annulus fluid increase in tandem with the helix tube side fluid input temperature. It is further distinguished by the fact that TFHE is significantly more effective in the counter flow arrangement than in the parallel flow design with respect to fluctuation in the fluid inlet temperature on the helical tube side.

Navarro et al.[29] performed numerical analysis on automobile air conditioning systems with open piston compressors under different conditions, comparing R290, R134a, and HFO1234yf. Compared to R134a, the experimental investigation found that the volumetric efficiency of the R290 compressor improved significantly, while the efficiency of the R1234yf compressor improved as the pressure ratio increased. The system performance is even better with IHX: however, R1234yf's volumetric efficiency is 5% lower and its cooling capacity is 9% lower than R134a's. Additionally, the value deteriorates at high condenser temperatures. Kang et al. concluded that R1234yf is a recently synthesized artificial working medium: therefore, the production process is still in its infancy and its capacity is constrained. When choosing an automotive air conditioning refrigerant, one must take into account its high cost and patent limits.

This study aims to replace high GWP refrigerant with a less greenhouse gas-emitting alternative. An internal heat exchanger can be added to the current system to increase the performance of R1234yf, which is slightly lower than R134a. According to the literature, internal heat exchanger CFD analysis for car air conditioning systems using R1234yf refrigerant is not frequently seen. This work is therefore unique in that regard. This study's objective is to analyze the

internal heat exchanger's (IHX) thermal design for R1234yf and how it affects COP. The flow parameters and temperature of the internal heat exchanger have been investigated numerically using CFD.

MATERIALS AND METHODS

Thermal Design of Internal Heat Exchanger

Internal heat exchangers, such as double-tube heat exchangers or tube-in-tube, can be used in air conditioning systems. A pair of concentric tubes with distinct inner diameters served as the models for this experiment Figure 1.

The tube-in-tube, counterflow internal heat exchanger was modelled using the energy balance and log means temperature difference (LMTD) methodology. For the thermal design of tube-in-tube type heat exchangers, the subsequent assumptions were made:

1. Kinetic energy and potential energy changes are neglected
2. No heat exchange with surroundings
3. Isenthalpic process considered for the expansion device
4. Steady-state operating conditions

The thermal design of the internal heat exchanger is studied for a 3.5 kW cooling capacity automobile air conditioning system that operates at 44 °C condensation temperature and 0 °C evaporator temperature. The thermodynamic cycle representation of an air conditioning system with R134a, R1234yf and R1234yf incorporated with an internal heat exchanger is represented in Figure 2-4 respectively. The property plot and overlay are generated in EES [30].

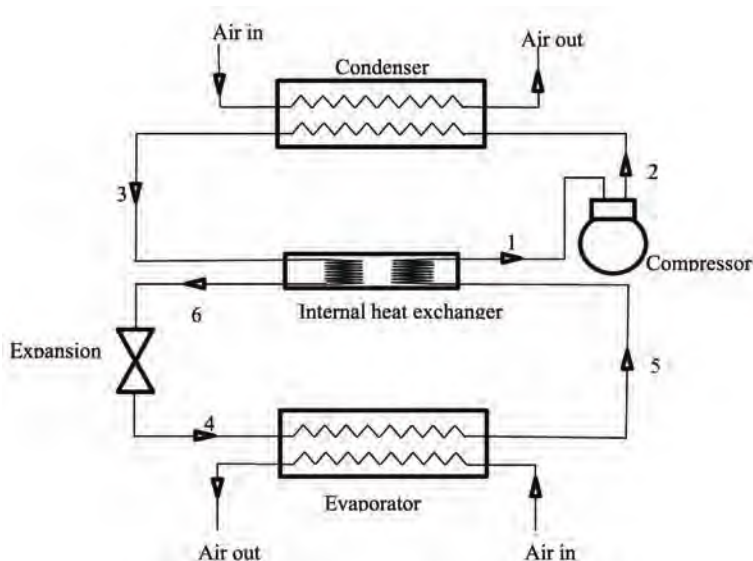


Figure 1. Internal heat exchanger (Tube in tube type).

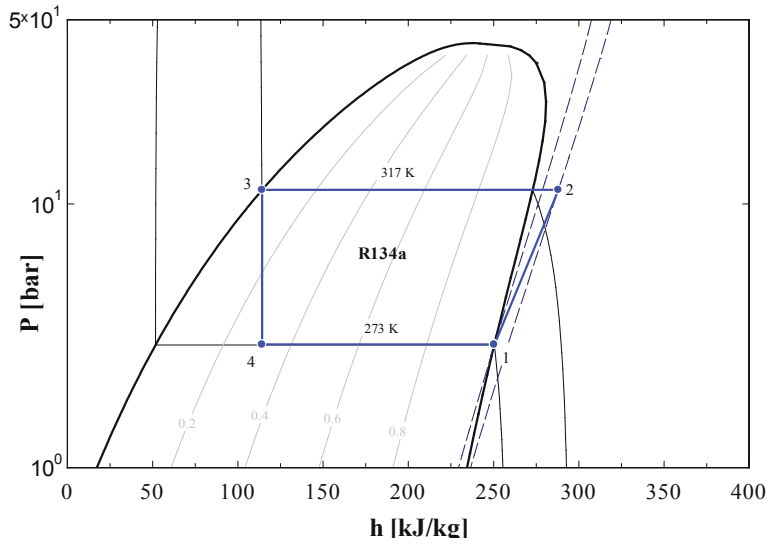


Figure 2. P-H diagram of R134a cycle.

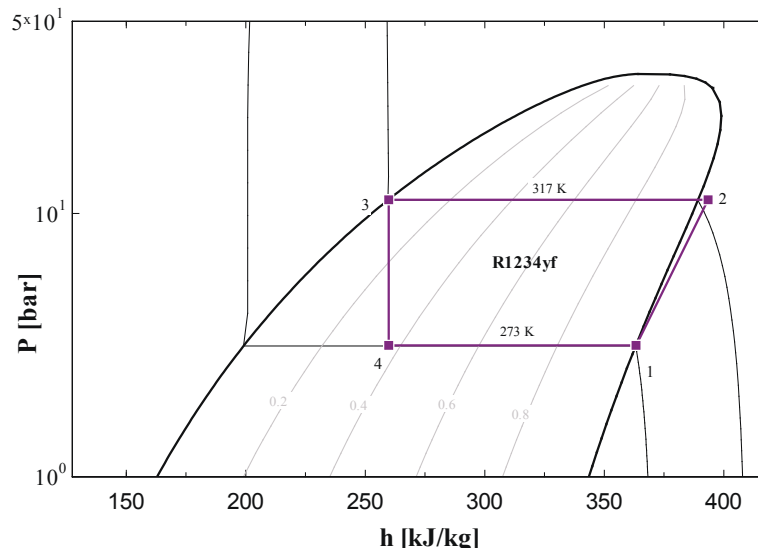


Figure 3. P-H diagram of R1234yf cycle.

The energy balance of the counterflow the tube in tube type heat exchanger can be established using the following equation from the fundamentals of the heat transfer.

The maximum amount of heat transfer rate that can occur is represented as,

$$Q_{\max} = c_{pc} \times (T_{hi} - T_{ci}) \quad (1)$$

The average specific heat of saturated vapor at evaporation temperature is c_{pc} , while the average specific heat of liquid at condenser pressure and condensation temperature is c_{ph} . The cold side of the internal heat exchanger's output temperature is 10 K, which is the degree of superheat taken into account in the original design.

$$Q_{CIHX} = c_{pc} \times (T_{co} - T_{ci}) \quad (2)$$

The effectiveness of the internal heat exchanger, ϵ , can be given by below mentioned equations. And the temperature outlets are calculated using the heat exchanger's effectiveness relation:

$$\epsilon = \frac{Q_{CIHX}}{Q_{\max}} \quad (3)$$

$$Q_{hiHX} = \epsilon \times Q_{\max} \quad (4)$$

where the coefficients of heat transmission within and outside the tube are h_i and h_o , respectively. In this study,

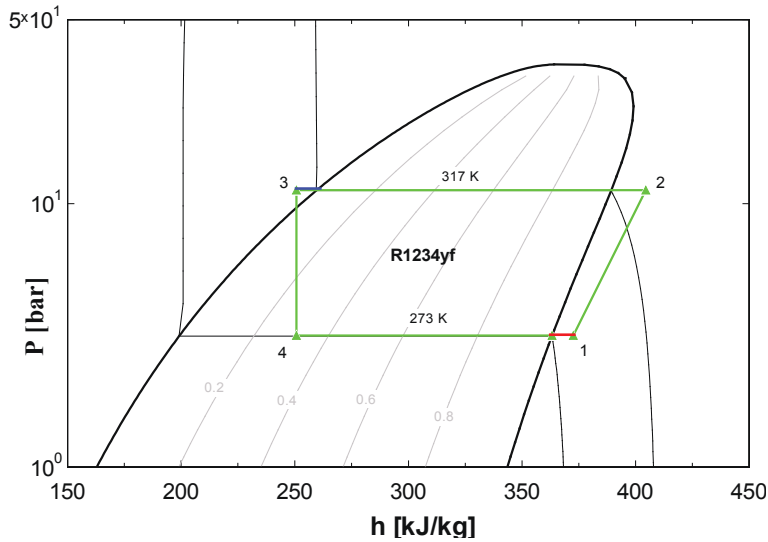


Figure 4. P-H diagram of R1234yf with IHX cycle.

the refrigerant vapour passes within the space between the heat exchanger's inner and outer tubes, while the refrigerant liquid passes via the inner tube. Consequently, for single-phase heat transfer the Dittus-Boelter correlation may be applied to calculate the heat transfer coefficient of the inner tube in turbulent fluid flows, which is given as [16].

$$N_u = 0.0238 \times Re^{0.8} \times P_r^n \quad (5)$$

When the vapor refrigerant is superheated, the Prandtl number exponent n is 0.4. The hydraulic diameter for the inner tube is $D_h = D_i$, annular space hydraulic diameter is determined by the formula $= D_o - D_i$. K and Re stand for the refrigerant's thermal conductivity and Reynolds number, respectively. If the Reynolds number in an annular space is less than 4000, the liquid flow is classified as laminar flow. For the fully developed flow, the Nusselt number expression is mentioned below.

$$h_h = 0.0238 \times \left(\frac{K_6}{D_h} \right) \times Re^{0.8} \times P_{rh}^{0.4} \quad (6)$$

$$h_c = 0.0238 \times \left(\frac{K_5}{D_h} \right) \times Re^{0.8} \times P_{rh}^{0.4} \quad (7)$$

For a counterflow type tube-in-tube type heat exchanger, T_m , R_{th} and L are the LMTD, heat resistance and heat exchanger length respectively, calculated by the following equations [31].

$$T_m = \frac{(Thi - Tco) - (Tho - Tci)}{\ln \frac{(Thi - Tco)}{(Tho - Tci)}} \quad (8)$$

$$R_{th} = \frac{T_m}{Q_{max}} \quad (9)$$

The heat transfer area, A is computed as follows, using the given relation:

$$A = \frac{\left(\frac{1}{h_h} \right) + \left(\frac{t}{K} \right) + \left(\frac{1}{h_c} \right)}{R_{th}} \quad (10)$$

$$A = L \times D_h \times \Pi \quad (11)$$

The pressure drop occurs in the internal heat exchanger on hot and cold sides can be calculated by considering the fluid properties at that state in the below equations.

$$\Delta P_{Pr} = \frac{2 \times f \times L \times \mu^2 \times Re^2}{D_h^3 \times \rho} \quad (12)$$

Where,

$$f = 0.0079 \times Re^{-0.22} \quad (13)$$

Without the internal heat exchanger cycle, the cycle's COP may be computed using

$$COP = \frac{RE}{W_c} \quad (14)$$

The COP of the air conditioning cycle with application of the internal heat exchanger can be calculated from

$$COP' = \frac{RE'}{W_c'} \quad (15)$$

An internal heat exchanger's efficiency and total heat transfer coefficient have a significant impact on how well refrigeration systems operate. The design of heat exchangers is determined after the heat transfer area of the heat exchanger. The software was utilized to extract the thermodynamic parameters of the refrigerant and write the mathematical model in Engineering Equation Solver (EES).

CFD Analysis of Internal Heat Exchanger

The current study looks into the flow physics inside the IHX that uses refrigerant R1234yf and is integrated into car air conditioning. A model of tube-in-tube heat exchanger with the internal diameter of 8.12 mm and thickness of 7 mm, whereas the outer tube has a diameter 17.45 mm and thickness of 8 mm is studied. The material of the heat exchanger

was selected as copper due to its high thermal conductivity. The cold liquid refrigerant temperature at the inlet is 273.2 K, the hot gaseous refrigerant temperature at the inlet is 317.2 K, and other boundary conditions are obtained from thermal design. The internal heat exchanger's temperature distribution and flow mechanics are predicted using the standard k- ϵ turbulence model. The geometry development Figure 5 and mesh generation Figure 6 are in Ansys 2021. For smoother transaction with better result, I had used program controlled linear mesh with element size of 0.5 mm. Here the best result outcome from three different length of internal heat exchanger is compared with theoretical thermal analysis results. The simulation study performed using the CFX solver in Ansys 2021 to explore the flow characteristics and flow parameters such as pressure, temperature

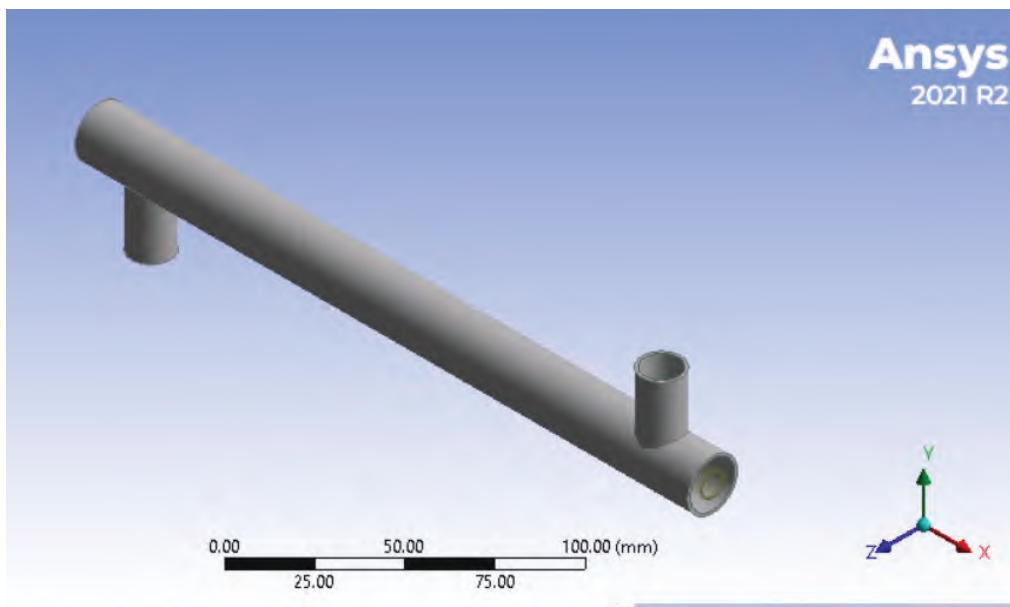


Figure 5. 3D Geometry of internal heat exchanger.

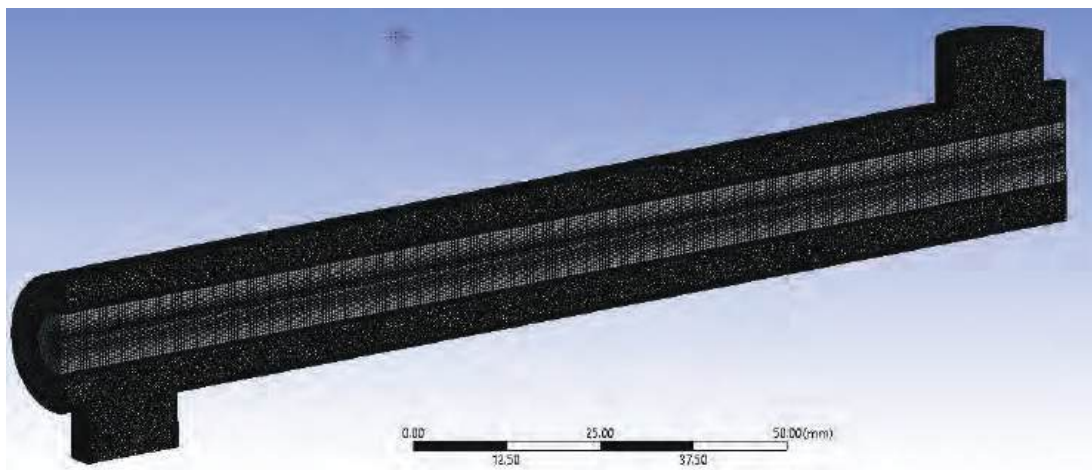


Figure 6. Meshing view of the internal heat exchanger.

and velocity is evaluated inside the tubes to obtain the data on the flow pattern.

To increase the accuracy and consistency of the numerical findings, the grid has been gradually improved by changing the element size and refinement level. Throughout the test, the number of elements was progressively raised from 827359 to 5934371. The temperature increase from intake to exit ($\Delta T_n = T_{n2} - T_{n1}$) of the cold fluid on the outer annulus side, or R1234yf, is the parameter taken into account for the grid independence test. This specific parameter was selected over others since the outer annulus side was most affected by the manual increase in cell size and refinement levels (because of the bigger volume). Therefore, comparing the grid independence to a metric that is most impacted by the intervention becomes relevant. The mesh elements of 4646235 and 5934371 exhibit a 0.16 K shift in ΔT_n as the grid is refined from 827359 elements to 5934371 elements with varying mesh sizes. Thus, for additional analysis and parametric examination, a grid of 4646235 components

has been selected. Figure 7 illustrates the results of the grid independence test.

The finite volume approach has been used to iteratively solve a three-dimensional computational domain of the IHX using the CFX (ANSYS 21 R2). Before the governing equations are solved, suitable boundary conditions have been established. No-slip boundary conditions have been established at the inner and outer tube walls. The velocity inlet boundary condition has been applied to both hot and cold fluids at the intake. As a thermal boundary condition indicated in table 2, the temperature at the entrance for both hot and cold fluids have been provided. The mass flow outlet boundary condition has been established at the exit of both fluids. The outer tube's outside walls have been made adiabatic to lessen the shell's thermal contact with the environment; this is comparable to the insulation on the heat exchanger's outer tube's outer side. Better thermal interaction between the hot and cold fluids is achieved by specifying an interface boundary condition at the proper walls.

RESULTS AND DISCUSSION

The table No. 3 shows the heat exchanger dimensions and performance parameters. And table No. 4 compares the performance parameters of the air conditioning system with R134a with the system with R1234yf and R1234yf+IHX. For the below mentioned initial data, copper (ASTM B68 C 12200) is chosen for the internal heat exchanger material and the design methodology performed.

For similar cooling capacity, the performance of system with refrigerant R134a is not much affected by the application of an internal heat exchanger. Its COP increases from 3.636 to 3.676, only 1.09%, while the performance gap of the system with refrigerant R1234yf as compared to the system with R134a without an internal heat exchanger is 5.17%, while the gap is minimized by 3.16% with an internal heat exchanger. Figure 8 represents the percentage increase of refrigerant effect and compressor work with an increase

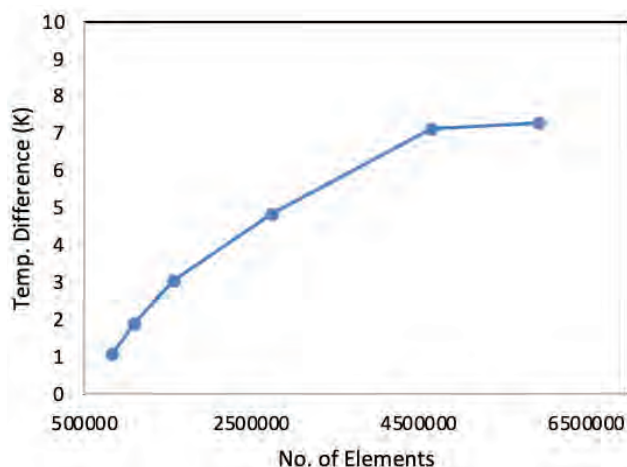


Figure 7. Grid independence test.

Table 2. Results of thermal design of internal heat exchanger

Cold Fluid (R1234yf)		
Boundary	Parameter	Value
Cold Fluid Inlet	Total Pressure	3.158 Bar
	Total Temperature	273.2 K
Cold Fluid Outlet	Mass Flow Rate	0.02873 kg/s
Hot Fluid (R1234yf)		
Boundary	Parameter	Value
Hot Fluid Inlet	Total Pressure	11.26 Bar
	Total Temperature	317.2 K
Hot Fluid Outlet	Mass Flow Rate	0.02873 kg/s

Table 3. Results of thermal design of internal heat exchanger

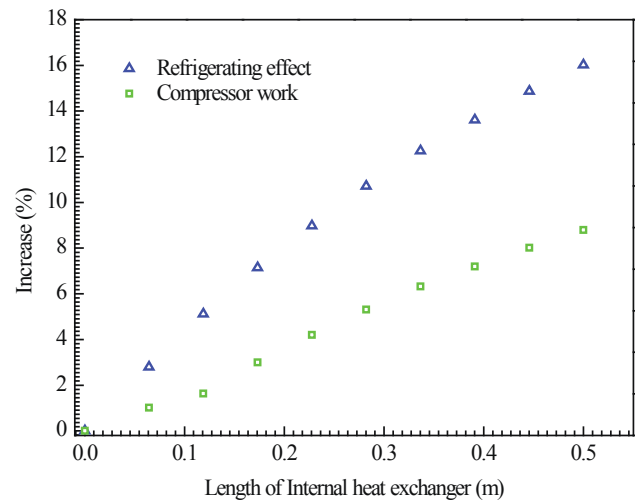
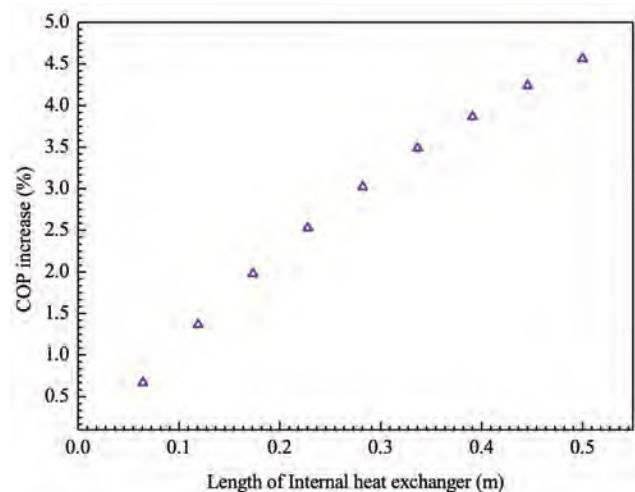
Inner tube diameter	8.12	mm
Inner tube thickness	0.7	mm
Outer tube diameter	17.45	mm
Outer tube thickness	0.8	mm
Length of internal heat exchanger	191.45	m
Heat transfer area	0.00561	m ²
Heat transfer coefficient on cold side	75.88	W/m ² K
Heat transfer coefficient on hot side	273.7	W/m ² K
Overall heat transfer coefficient	41.31	W/m ² K

in length of the internal heat exchanger. Whereas Figure 9 shows an increase in COP with variation in length of the internal heat exchanger. Figure 10 represents the change in degree of superheating and degree of subcooling with increase in length of the internal heat exchanger. Whereas Figure 11 shows pressure drops in hot side and cold side of the refrigerant in the internal heat exchanger with variation in length.

The flow pattern inside an internal heat exchanger expressed via velocity contours in Figures 12 and 13. The flow pattern inside an internal heat exchanger is expressed via temperature contours in Figures 14 and 15. The Figure 16 and Figure 17 represent the refrigerant temperature at the cold outlet and hot outlet junctions, respectively. Figure 18 and Figure 19 represent the cold gaseous refrigerant pressure contours and hot liquid pressure contours in tube in the tube internal heat exchanger, respectively.

The Table 5 represents the results of EES simulation and CFD analysis for different lengths of the internal heat exchanger as 191.4 mm, 220 mm and 240 mm and their comparison. Both the liquid and the vapor's temperature as they enter the internal heat exchanger determine how much heat is transferred there. The Figure 20 express the outlet temperature through the tube-in -tube internal heat exchanger. Additionally, it was observed that the heat exchange between the liquid and vapor in the heat exchanger caused a change in the temperature at which the liquid and vapor exit the tube-in-tube internal heat exchanger. The cold suction vapor refrigerant passing through the internal heat exchanger receives heat from the warm liquid refrigerant, which causes the vapor to become

superheated and increase in temperature while the liquid subcools and decreases in temperature. Figure 21 shows the degree of superheating and degree of subcooling through the tube-in-tube internal heat exchanger. Figure 22 presents the pressure drop occurs through the tube-in-tube internal

**Figure 8.** Effect of internal heat exchanger length on compressor work and refrigerating effect.**Figure 9.** Effect of internal heat exchanger length on % COP increase.**Table 4.** Result of thermal analysis

Parameter	R134a	R1234yf	R1234yf with IHX
m (kg/s)	0.0257	0.0338	0.03134
RE (kJ/kg)	136.2	103.6	111.7
Wc (kJ/kg)	39.89	30.07	31.75
COP	3.636	3.443	3.518

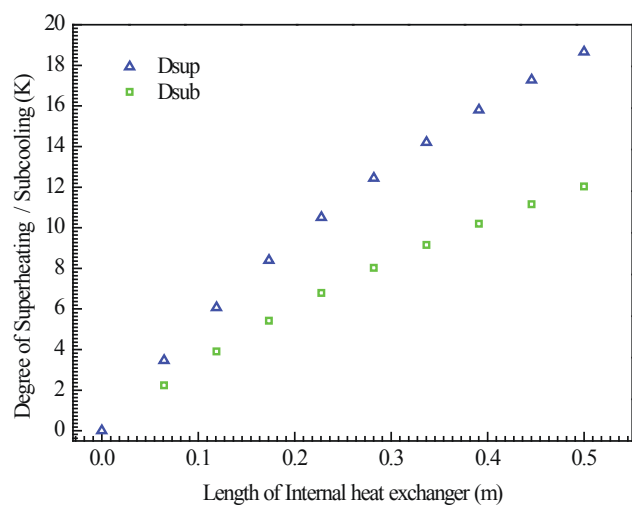


Figure 10. Effect of internal heat exchanger length on Degree of Superheating/Subcooling.

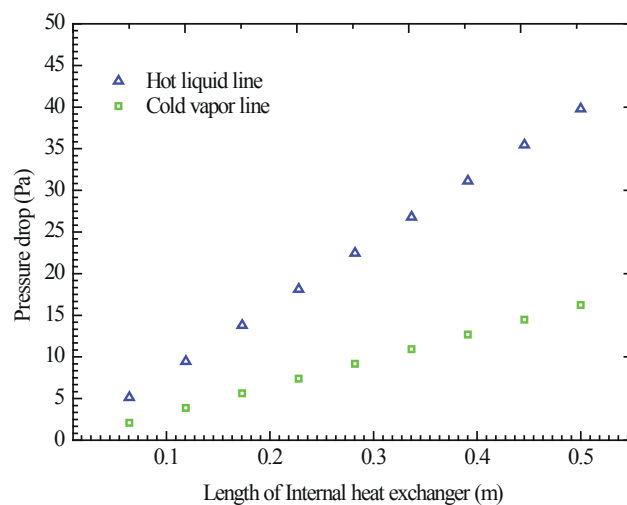


Figure 11. Effect of internal heat exchanger length on pressure drop in internal heat exchanger.

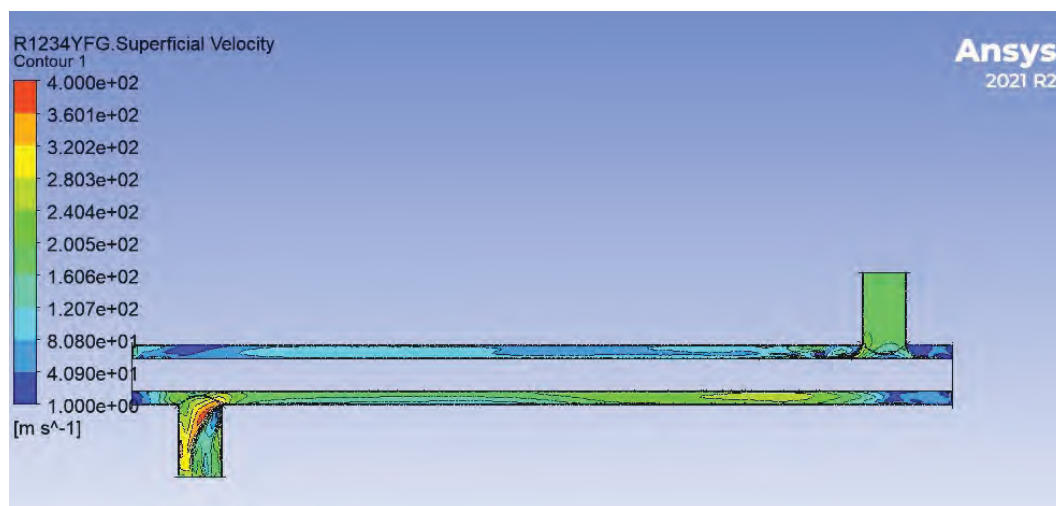


Figure 12. Velocity profile of gaseous R1234yf in outer tube of internal heat exchanger.

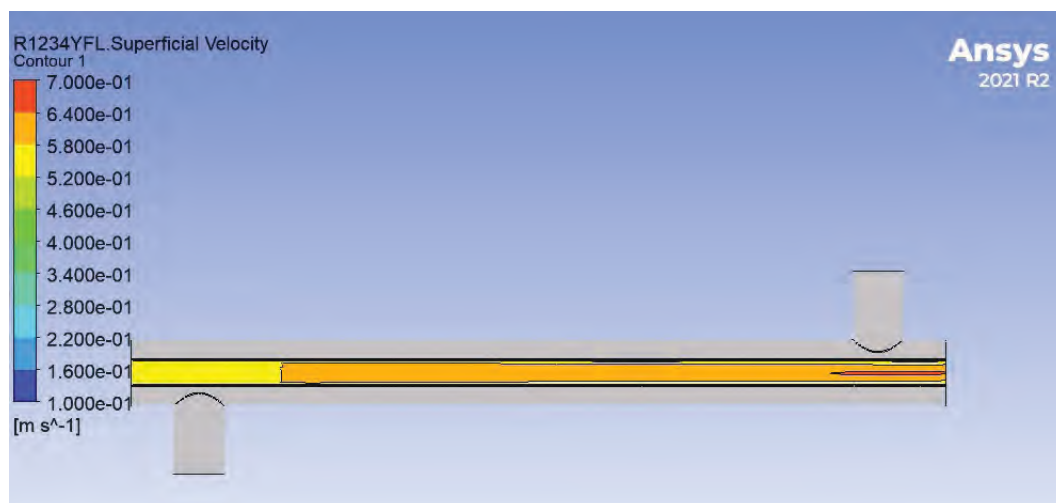


Figure 13. Velocity profile of liquid R1234yf in inner tube of internal heat exchanger.

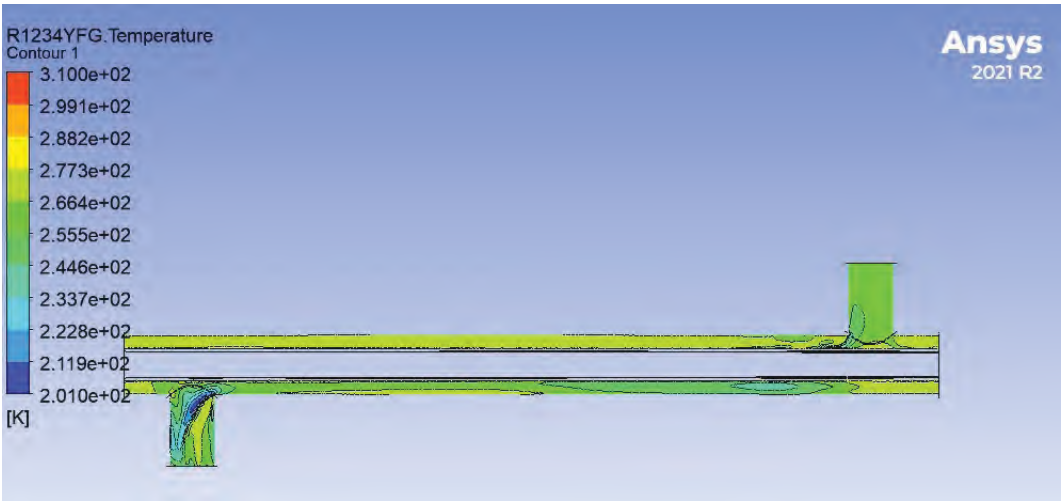


Figure 14. Temperature distribution vapour R1234yf in internal heat exchanger.

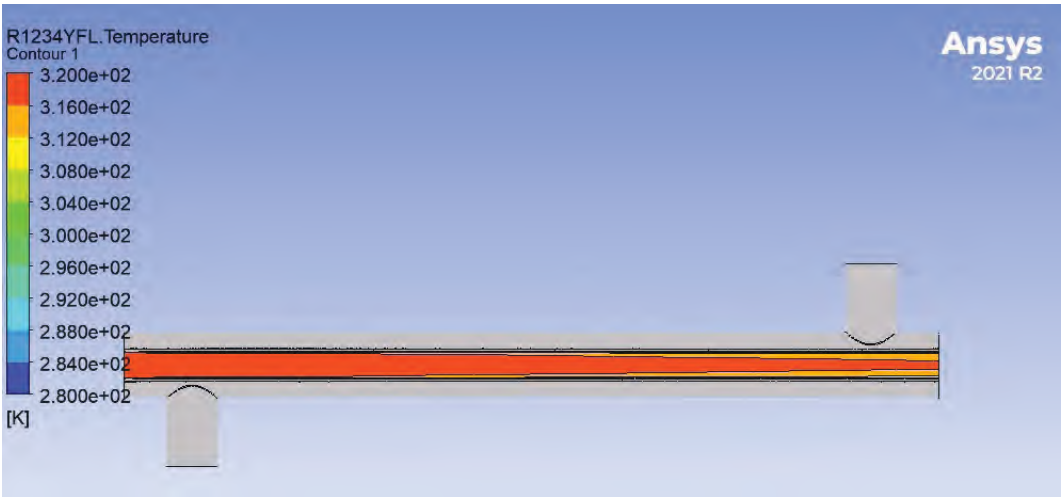


Figure 15. Temperature distribution liquid R1234yf in internal heat exchanger.

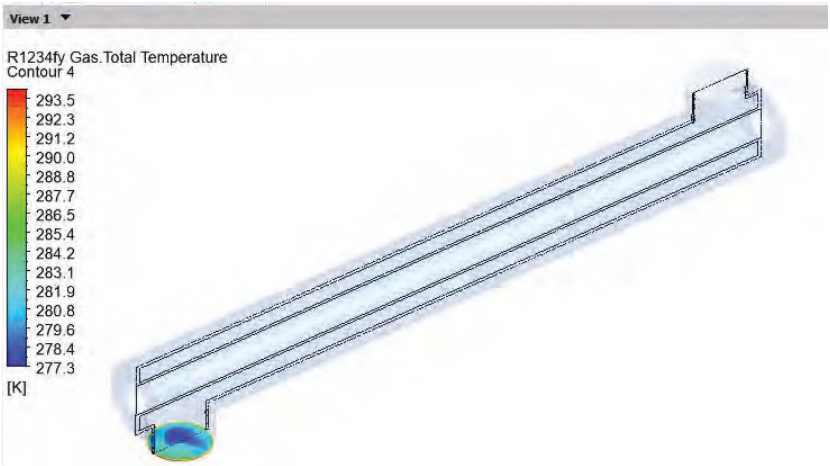


Figure 16. Cold outlet temperature of R1234yf.

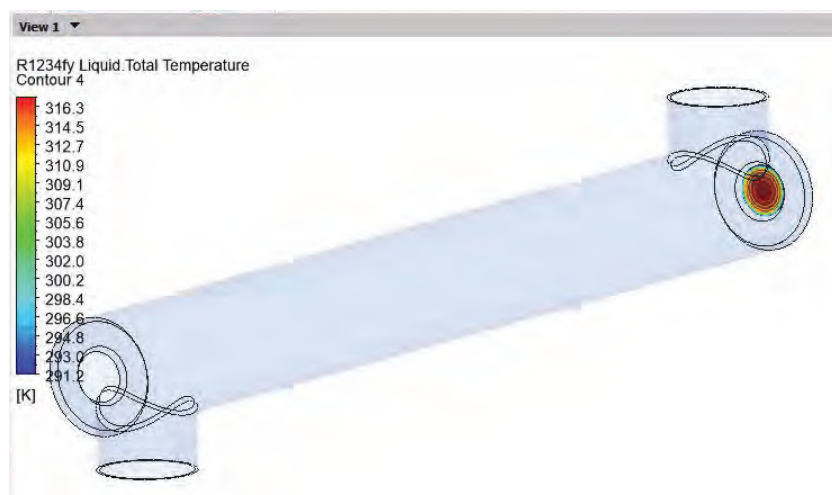


Figure 17. Hot outlet temperature of R1234yf.

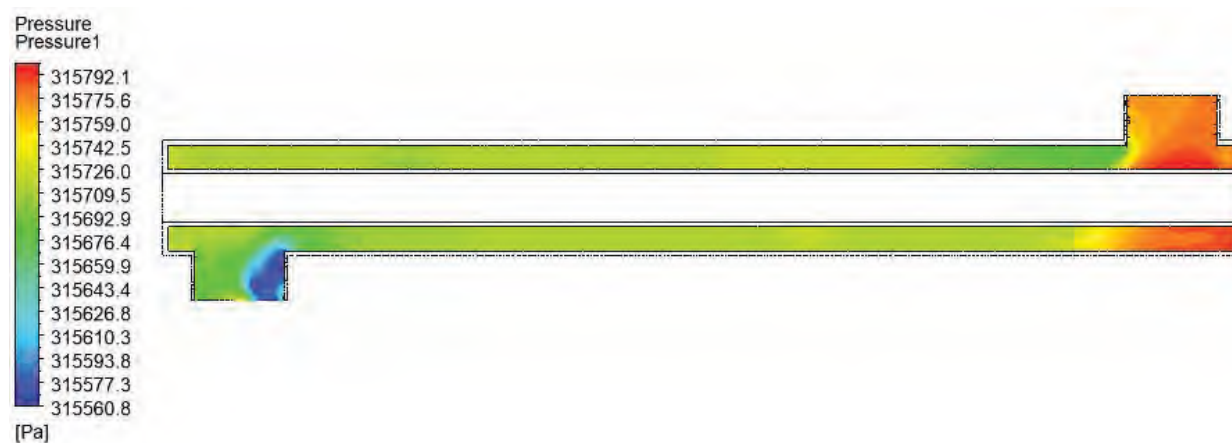


Figure 18. Pressure profile of gaseous R1234yf in outer tube of internal heat exchanger.

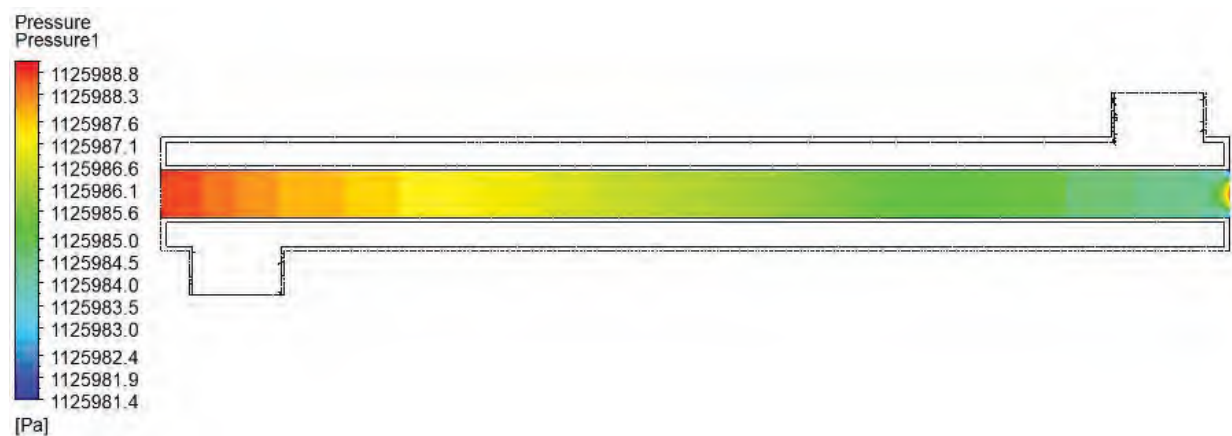


Figure 19. Pressure profile of liquid R1234yf in inner tube of internal heat exchanger.

Table 5. Result analysis table

Parameters	Theoretical	CFD Analysis		
Length of heat exchanger (mm)	191.45	191.45	220	240
Cold outlet temperature (K)	283.2	277.99	279.74	280.12
hot outlet temperature (K)	310.7	314.951	313.414	312.409
Degree of superheat (K)	10	4.99	6.74	7.12
Degree of subcooling (K)	6.5	2.249	3.786	4.791
Pressure drop hot side (Pa)	15.23	40	50	50
Pressure drop cold side (Pa)	6.207	14.86	24.43	25

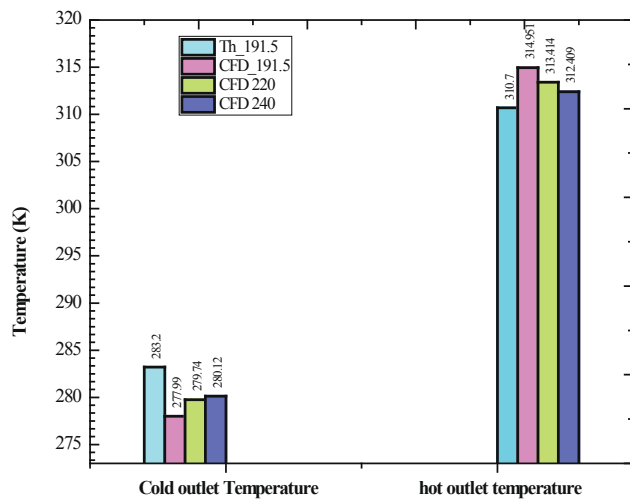


Figure 20. Internal heat exchanger outlet temperature.

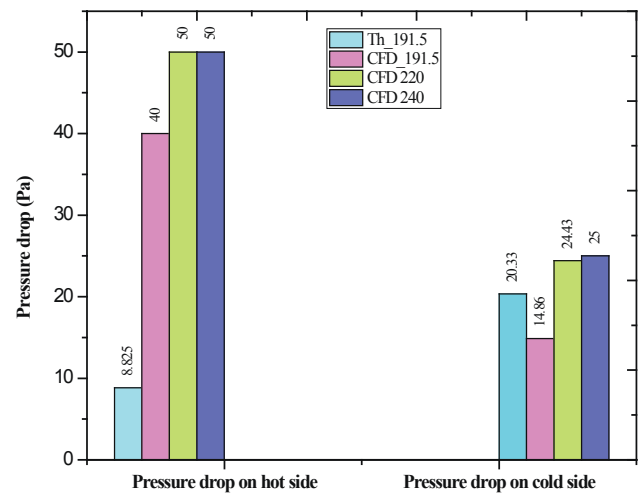


Figure 22. Pressure drop through in internal heat exchanger.

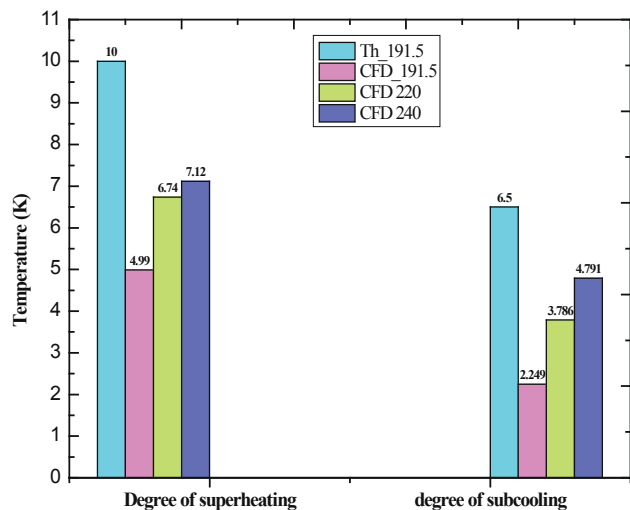


Figure 21. Degree of superheating and subcooling in internal heat exchanger.

heat exchanger on hot side and cold side of refrigerant. Two variables affect how much the liquid and vapor temperatures vary when they exit the internal heat exchanger. The

first is heat exchanger effectiveness, whereby suction vapor is progressively superheated as heat exchanger efficiency rises. The second is the fluid's specific heat, which is influenced by the working environment. However, the system's performance is hampered by a large amount of superheated vapor near the compressor entry. The suction vapor temperature in the R1234yf case increased to the superheated temperature through the internal heat exchanger, surpassing the superheated temperature of the cycle with R134a. Subcooling is less than superheating as a result of heat exchange between warm liquids and cold vapors in the internal heat exchanger. This is caused by the fluid's specific heat as well as the internal heat exchanger's heat transfer coefficient.

CONCLUSION

In the present study, the thermal design of an internal heat exchanger made for R1234yf and its effect on the coefficient of performance are analyzed with respect to R134a. Refereeing the literature available, CFD analysis of internal heat exchanger for automobile air conditioning system with

refrigerant R1234yf is rarely observed. Hence the novel study of the numerical investigation has been conducted utilizing computational fluid dynamics analysis to explore the flow characteristics and temperature in the internal heat exchanger. Through the addition of the internal heat exchanger to the refrigeration cycle, the performance parameters of R1234yf were assessed for a range of evaporation and condensation temperatures in this theoretical investigation. The outcomes were then compared with the R134a basic cycle under varied internal heat exchanger effectiveness. The greatest impact of subcooling and superheating effects on the cycle efficiency of the R1234yf system was observed. And recommended 8.5 K degree of superheating for refrigerant R1234yf [32]. In this study, performed to achieve 10 K degree of superheating. The similar trend of results, i.e. COP, heat transfer coefficient, effectiveness and pressure drop, are observed in research by Wantha [20].

- The numerical analysis shows that the automobile air conditioning system with R134a has a higher coefficient of performance than the system with R1234yf, but the application of an internal heat exchanger with R1234yf improved the system coefficient of performance higher than R134a.
- As an effect of an internal heat exchanger, the suction gas is heated, thus decreasing the suction density: hence, the compressor discharge temperature rises. The system capacity and efficiency can be increased with an internal heat exchanger, but the compressor discharge temperature may rise to an intolerable level.
- The increasing suction-side pressure drop results in an increase in compressor work at some level. The pressure drop is in tolerable limit that justifies the application of an internal heat exchanger in an air conditioning system with R1234yf.
- According to the numerical model, the internal heat exchanger increase in heat transfer will always lead to improved performance of the system with R1234yf. An increase in enthalpy changes across the evaporator was observed along with degree of subcooling. Furthermore, the rates of increase for degree of superheat are higher than those for degree of subcooling, and these values rise with increasing efficiency independent of the operating temperatures.
- The computational fluid dynamics analysis shows that increasing length of the internal heat exchanger increases the heat transfer. Increasing the suction-side pressure drop results in increase in compressor work at some level. The pressure drop is within a tolerable limit that suggests the application of an internal heat exchanger with an air conditioning system with R1234yf.
- For the system with refrigerant R134a, performance is not much affected by the application of an internal heat exchanger. Its coefficient of performance increases from 3.636 to 3.676, only 1.09%, while the performance gap of

the system with refrigerant R1234yf as compared to the system with R134a without an internal heat exchanger is 5.17%, while the gas is decreased up to 3.16% with an internal heat exchanger.

- The future research direction of the study is to conduct experimental analysis of automobile air conditioning system of R1234yf with IHX. And the ongoing attempt of manufacturer to increase the production of R1234yf, and decrease its cost that's the challenge.

ACKNOWLEDGEMENTS

The author is thankful to the Science and Engineering Research Board (SERB), Ministry of Education, Govt. of India, for providing funding for ongoing research on air conditioning systems for the automotive sector using new low GWP R1234yf refrigerant under IMPRINT-II (IMP/2018/001882) and also to the Department of Mechanical Engineering, SVNIT, Surat, for all support and encouragement.

ABBRIATION

GWP	Global Warming Potential
ODP	Ozone depleting potential
HFO	HFO Hydrofluoroolefin
MAC	MAC Mobile Air Conditioning
QL	Cooling capacity
IHX	Internal heat exchanger

NOMENCLATURE

$\dot{m}_r$	Mass flow rate (kg/sec)
Q_{cIHX}	Heat transfer in cold line of IHX
Q_{hIHX}	Heat transfer in hot line of IHX
Q_{max}	Maximum heat transfer through IHX
T_m	Logarithmic mean temperature difference
R_{th}	Thermal resistance
ϵ	Effectiveness of internal heat exchanger
Nu	Nusselt number
Re	Reynold's number
Pr	Prandtl number
h	Heat transfer coefficient
A	Heat transfer area
K	Thermal conductivity
RE	Refrigerating effect

Subscript

a	air
c	condenser
Comp	compressor
e	evaporator
ex	Expansion valve
i	inlet
r	refrigerant
sub	subcooling

sup superheating
h hot side refrigerant
c cot side refrigerant

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] UNEP Technical Options Committee. 2018 report of the Refrigeration, Air Conditioning and Heat Pumps Technical Options Committee. Nairobi: United Nations Environment Programme; 2019.
- [2] United Nations. Kigali Amendment 2016;2:1-16. [\[Crossref\]](#)
- [3] India Cooling Action Plan. Ozone Cell, Ministry of Environment, Forest & Climate Change, Government of India; 2019.
- [4] Yang CY, Nalbandian H. Condensation heat transfer and pressure drop of refrigerants HFO-1234yf and HFC-134a in small circular tube. *Int J Heat Mass Transf* 2018;127:218-227. [\[Crossref\]](#)
- [5] Hasheer SM, Srinivas K, Bala PK. Energy analysis of HFC-152a, HFO-1234yf and HFC/HFO mixtures as a direct substitute to HFC-134a in a domestic refrigerator. *Stroj Cas* 2021;71:107-120. [\[Crossref\]](#)
- [6] Satapathy PP, Satapathy PK, Sahoo SS. Comparative performance study of aut cascade and cascade refrigeration systems using working fluid pair R23/R507A. *Recent Adv Mech Eng* 2024;411-426. [\[Crossref\]](#)
- [7] Reasor P, Aute V, Radermacher R. Refrigerant R1234yf performance comparison investigation. *Int Refrig Air Cond Conf Purdue* 2010:1-7.
- [8] Devotta S, Waghmare AV, Sawant NN, Domkundwar BM. Alternatives to HCFC-22 for air conditioners. *Appl Therm Eng* 2001;21:703-715. [\[Crossref\]](#)
- [9] Ghodbane M. An investigation of R152a and hydrocarbon refrigerants in mobile air conditioning. *SAE Trans Sect J Passeng Cars* 1999;108:1658-1673. [\[Crossref\]](#)
- [10] Alkan A, Kolip A, Hosoz M. Energetic and exergetic performance comparison of an experimental automotive air conditioning system using refrigerants R1234yf and R134a. *J Therm Eng* 2021;7:1163-1173. [\[Crossref\]](#)
- [11] Direk M, Kelesoglu A. Automotive air conditioning system with an internal heat exchanger using R1234yf and different evaporation and condensation temperatures. *Therm Sci* 2019;23:1115-1125. [\[Crossref\]](#)
- [12] Hmood KS, Apostol V, Pop H, Badescu V, Pop E. Drop-in and retrofit refrigerants as replacement possibilities of R134a in domestic/commercial refrigeration and automobile air conditioner applications. *J Therm Eng* 2021;7:1815-1835. [\[Crossref\]](#)
- [13] Sharif MZ, Azmi WH, Ghazali MF, Samykano M, Ali HM. Performance improvement strategies of R1234yf in vapor compression refrigeration system as a R134a replacement: a review. *J Taiwan Inst Chem Eng* 2023;148:105032. [\[Crossref\]](#)
- [14] Karademir H, Wongwises S. Comprehensive review on the flow characteristics of two-phase flows in inclined tubes. *J Therm Eng* 2021;7:483-549. [\[Crossref\]](#)
- [15] Agarwal SA, Akhilesh AA, Arora BB. Energy and exergy investigations of R1234yf and R1234ze as R134a replacements in mechanically subcooled vapour compression refrigeration cycle. *J Therm Eng* 2020;7:109-132. [\[Crossref\]](#)
- [16] Daviran S, Kasaeian A, Golzari S, Mahian O, Nasirivatan S, Wongwises S, et al. A comparative study on the performance of HFO-1234yf and HFC-134a as an alternative in automotive air conditioning systems. *Appl Therm Eng* 2017;110:1091-1100. [\[Crossref\]](#)
- [17] Lin Y, Meng Z, Huo Z, Ding C, Wang S, Wang L. Theoretical analysis of R1234yf and R1234yf/R125 mixture as replacement of R134a in vapor compression system. *Int J Low-Carbon Technol* 2024;19:490-496. [\[Crossref\]](#)
- [18] Lee Y, Kang DG, Jung D. Performance of virtually non-flammable azeotropic HFO1234yf/HFC134a mixture for HFC134a applications. *Int J Refrig* 2013;36:1203-1207. [\[Crossref\]](#)
- [19] Patel B, Parekh A. Energy, exergy and entropy analysis with R1234yf as an alternate refrigerant to R134a of automobile air conditioning system. *J Therm Eng* 2024;10:101-114. [\[Crossref\]](#)
- [20] Wantha C. Analysis of heat transfer characteristics of tube-in-tube internal heat exchangers for HFO-1234yf and HFC-134a refrigeration systems. *Appl Therm Eng* 2019;157:113747. [\[Crossref\]](#)

- [21] Kwon YC, Kim DH, Lee JH, Choi JY, Lee SJ. Experimental study on heat transfer characteristics of internal heat exchangers for CO₂ system under cooling condition. *J Mech Sci Technol* 2009;23:698-706. [\[Crossref\]](#)
- [22] Alkan A, Kolip A, Hosoz M, Yashawantha KM, Vinod AV, Khatoon S, et al. Energetic and exergetic performance comparison of an experimental automotive air conditioning system using refrigerants R1234yf and R134a. *J Therm Eng* 2021;7:490-504. [\[Crossref\]](#)
- [23] Alkan A, Kolip A. Thermodynamic analysis of an automotive air conditioning system using R1234yf and R134a. In: *Sixth Int Exergy, Energy Environ Symp (IEEEES-6)*, Recep Tayyip Erdoğan Univ, Rize, Turkey; 2013.
- [24] Direk M, Kelesoglu A, Akin A. Drop-in performance analysis and effect of IHX for an automotive air conditioning system with R1234yf as a replacement of R134a. *Stroj Vestn J Mech Eng* 2017;63:314-319. [\[Crossref\]](#)
- [25] Direk M, Kelesoglu A. Theoretical performance analysis of an R1234yf refrigeration cycle based on the effectiveness of internal heat exchanger. *Hittite J Sci Eng* 2017;4:23-30. [\[Crossref\]](#)
- [26] Cho H, Lee H, Park C. Performance characteristics of an automobile air conditioning system with internal heat exchanger using refrigerant R1234yf. *Appl Therm Eng* 2013;61:563-569. [\[Crossref\]](#)
- [27] Mota-Babiloni A, Navarro-Esbrí J, Barragán Á, Molés F, Peris B. Drop-in energy performance evaluation of R1234yf and R1234ze(E) in a vapor compression system as R134a replacements. *Appl Therm Eng* 2014;71:259-265. [\[Crossref\]](#)
- [28] Mohapatra T, Ray S, Sahoo SS, Padhi BN. Numerical study on heat transfer and pressure drop characteristics of fluid flow in an inserted coiled tube type three fluid heat exchanger. *Heat Transf Asian Res* 2019;48:1440-1465. [\[Crossref\]](#)
- [29] Navarro E, Martinez-Galvan IO, Nohales J, González-Macia J. Comparative experimental study of an open piston compressor working with R1234yf, R134a and R290. *Int J Refrig* 2013;36:768-775. [\[Crossref\]](#)
- [30] Klein SA, Alvarado F. *Engineering Equation Solver*. Middleton, WI: F-Chart Software; 2005.
- [31] Shah RK, Sekulic DP. *Fundamentals of heat exchanger design*. 1st ed. Hoboken, NJ: John Wiley & Sons; 2003.
- [32] Perez Gracia V, Belman-Flores JM, Rodriguez-Munoz JL, Rangel-Hernandez VH, Gallegos-Munoz A. Second law analysis of a mobile air conditioning system with internal heat exchanger using low GWP refrigerants. *Entropy* 2017;19:40175. [\[Crossref\]](#)



Research Article

A novel approach to augment stepped solar still productivity using the heat storage characteristics of water

J.S D'COTHA^{1,2}, P. R. SURESH³, P. SAJEESH^{3,*}

¹APJ Abdul Kalam Technological University, Trivandrum, 695016, India

²Department of Mechanical Engineering, SCMS School of Engineering and Technology, Ernakulam, Kerala, 683576, India

³Department of Mechanical Engineering, NSS College of Engineering, Palakkad, Kerala, 678008, India

ARTICLE INFO

Article history

Received: 31 August 2024

Revised: 17 January 2025

Accepted: 19 January 2025

Keywords:

Feed Water; Heat Storage;

Nocturnal Yield; Stepped Still

ABSTRACT

Improvement in freshwater productivity by minimizing the carbon footprint is the main goal paving way to the development and modification of solar stills. The primary objective of this study is to enhance the productivity of the solar still while optimizing construction and maintenance costs. In contrast to other phase change materials and heat storage materials, this study utilizes water as the sensible heat storage medium. This investigation deviates from established protocols by implementing a feed water addition strategy that exceeds evaporated water quantities. The still's performance was enhanced by using a dynamic water depth strategy, where lower depths during morning hours boosted daytime yield, and higher depths later in the day improved night-time output. This process optimizes the use of incident energy, converting it into specific heat energy while minimizing the impact on temperature increase. Experiments in Kochi, India, revealed that the solar still's overall yield, without feed water addition, decreased with increasing initial water depth, yielding 5290 ml/m², 5140 ml/m², and 4800 ml/m² at depths of 0.5 cm, 1.0 cm, and 1.5 cm, respectively. Adding 100 ml of feed water to each tray at the three initial water depths resulted in productivity increases of 2.3%, 2.5%, and 2.9%, respectively, compared to no feed water additions. Furthermore, adding 200 ml of feed water led to more significant improvements in productivity, with increases of 5.9%, 8.2%, and 7.5%, respectively. A solar still with an initial water depth of 0.5 cm and a feed water inlet of 400 ml per tray achieved the highest distillate output of 6330 ml/m²/day, corresponding to a maximum daily efficiency of 66.45%. Experimental values were validated against theoretical energy balance equations, with a deviation of less than 6%. The cost of distillate produced was \$0.03, which is approximately one-tenth the price of water available in the regional market. Solar stills offer a viable solution for producing potable water, addressing the drinking water needs of communities and urban areas facing water scarcity. The low cost, simplicity, and eco-friendliness of this desalination process make it ideal for household use. Future research should focus on developing intelligent feed water systems that adapt to solar radiation and ambient temperature fluctuations, optimizing feed water quantity and maximizing daily still output.

Cite this article as: D'Cotha JS, Suresh PR, Sajeesh P. A novel approach to augment stepped solar still productivity using the heat storage characteristics of water. J Ther Eng 2025;11(5):1355–1373.

*Corresponding author.

*E-mail address: sajeeshperil@gmail.com

This paper was recommended for publication in revised form by

Editor-in-Chief Ahmet Selim Dalkilic



INTRODUCTION

Potable drinking water is the essential commodity ensuring the sustainability of mankind on earth. Population growth combined with technological advancements has increased the per-capita consumption of fresh water. Technological growth has improved the comforts and living standards, but at the cost of a climatic imbalance in the environment. Nine out of ten people lack access to clean drinking water, and by 2025, half of the world's population is predicted to live in water scarce regions [1]. Growing population, urbanization and change in climatic patterns has increased the number of water stressed regions and thus increasing the demand for potable water. Large scale desalination techniques like reverse osmosis, electrodialysis and thermal methods are classified as energy-intensive processes [2] and pose a serious threat to the environment in terms of carbon footprint [3] and salt disposals.

Thermal and membrane-based desalination processes are classified as high specific energy consumption technologies contributing to higher water production costs limiting their utilization [4]. The materials required for the fabrication of the still are produced through energy intensive manufacturing processes. However, the emissions from the manufacturing phase can be considered minimal, taking into account the benefits gained by solar stills. Thus the solar still during its operational period can be considered as a green method compared to other desalination technologies. The need for solar desalination systems gained importance due to environmental risks, posed by burning of fossil fuels [5] and its impact on the carbon footprint. Though solar energy is available intermittently, it is available in abundance at zero cost. Furthermore, even the most isolated, remote and arid regions of the planet, have accessibility to solar energy, which is crucial in harnessing the solar power to meet the energy requirements for water purification. With minimal investment cost and operational expenses for solar stills, the production of potable water comes at an exceptionally low unit cost in comparison to alternative desalination technologies. In addition, simple operation and less maintenance makes this technology more acceptable by the common man. Potable water production using solar stills is most viable for small and arid communities, in spite of its low productivity.

Energy storage elements store the solar energy received and release this thermal energy when the incident solar radiation is low. Attaining a steady pattern of hourly yield during cloudy and rainy days is desirable and can be accomplished with the inclusion of heat storage elements. Sandy beds were used as the heat storage element in stepped trays, and it was observed that, as the depth of the bed increased the productivity decreased [6]. Jathar and Ganeshan [7] have used different masses of sand, broken bricks and broken concrete to evaluate the productivity of concave stepped trays. It can be observed that, the productivity improved with mass and specific heat capacity of the stones.

Black poly propylene plastic tubes with 6 mm diameter and 2.25 cm height has been used to form honeycomb matrix in the flat basin to reduce the reflection losses to augment the distillate output [8]. Usage of conch biomaterial shells as energy storage in the still achieves a yield 10.8% more than that of the conventional still [9]. Ramalingam et al. [10] used copper scraps coupled with coconut coir disk in stepped still to produce 44.24% increased yield compared to the one without them. Researchers also used aluminum fillings [11], pin fins [12], copper scraps [10] and fins [13] as heat sensible materials.

Some researchers have also utilized water as a sensible heat storage medium due to its larger heat capacity. Due to the presence of high energy hydrogen bonding (10–30 kJ/mol) in water, it shall be considered the best possible thermal fluid at an operating range of 0°C to 100°C [14]. The quantum of sensible heat energy stored in water is augmented, when there is an increase in the temperature of water or by increasing the mass of water exposed to solar radiation [15]. In the absence of solar energy, the stored sensible energy is further utilized to assist the evaporation of water to achieve a steady productivity [16]. According to Kabeel et al. [17], the productivity of the still is dependent on the mass of the sensible heat storage material and depth of water.

The addition of feed water can be classified into static and dynamic mode [18]. In static mode, the feed water is introduced into the still once at the start of the day, whereas in dynamic mode the feed water is fed periodically. Few researchers abstained from feed water addition [19–21], while most researchers added makeup water on an hourly basis [13] or in half hour intervals [7,22,23] to compensate the evaporated water. Abujazar et al. [24] used water level sensor to maintain a constant depth of water in the triangular trays. In the case of inclined and weir type stills, the feed water is allowed to overflow the absorber plate at a constant flowrate [10].

Siddula et al. [16] compared the productivity of a triangular shaped and conventional still with different depths of water. It was recorded that, as the depth of water was increased from 1.5 cm to 3.0 cm in the triangular and conventional still, the output declined by 19.09% and 35.92% respectively. At very low depths of water in the basin, chances of dry spots been developed can reduce the productivity [19]. Hence it is advised to always keep the tray water levels at least minimal. Kabeel et al. [25] tested extended type trays with water depths 5 mm, 10 mm and 20 mm and observed that the productivity decreased with the increase in depth of water. According to Omara et al. [26], the extended type trays have more surface area and for a given depth of water, they can contain more mass of water compared to a conventional still. As the water depth increases and due to the high specific heat capacity of water, a greater amount of heat is stored in the water, which promotes the formation of vapors during the nocturnal hours [27]. Due to the small quantity of water in the trays of the

stepped solar still, the evaporation rate is more due to the less time consumed to convert water particles to vapor [28]. The hourly efficiency of the still gradually increases from the start of the day to reach its maximum and then gradually declines thereafter [29].

Abdullah et al. [30] in their recent review, found that increasing the water depth in stills leads to a decrease in distillate production, indicating an inverse relationship. A study by Alwan et al. [31] investigated the relationships between climatic, operational, and design parameters and still productivity, concluding that increased water depth in the still leads to decreased productivity. Nougriaya et al. [32] found that stills with 1-2 cm water depth produce more distillate efficiently. Research by Siddula et al. [16] on single slope solar stills with varying water depths (1.5-3 cm) revealed that shallower depths produce higher hourly yields in the morning, whereas deeper depths yield more in the afternoon.

It can be concluded from the extensive literature that, distillate output of a solar still can be augmented by utilizing more radiant energy through heat storage methods. The liquid in solar stills can also serve as a heat storage medium, and the dynamic impact of this sensible heat storage approach has received limited attention in the existing literature. Diverging from the conventional approach of adding feed water to replenish evaporated water, this research explores the addition of a fixed volume of water at fixed intervals to enhance the water depth in the trays. It is worth observing that the amount of water added exceeds the quantity of water evaporated. This feeding mechanism increases the water depth in the trays compared to the initial water depth. This gradual increase in water depth facilitates more efficient utilization of solar energy from the sun, primarily through sensible heat storage.

Theoretical Model of the Solar Still

A theoretical model of the stepped solar still is developed in this section, based on energy balance at the basin, water, and glass cover [33]. Glass accumulates the solar energy from the sun in the specific heat capacity which raises its temperature. A part of the energy is lost to the ambient through convection and radiation from the condensate surface. The solar energy transmitted by glass is received by the water body in the trays. The energy absorbed by water is accumulated in the water contributing the increase in temperature. The temperature difference between the water and the glass cover causes the vapors to advance towards the lower temperature condensate cover. The vapors condense at the glass surface and this energy is transferred to the condensate cover. Similarly convective and radiation energy is transferred between the water body and the glass cover. A portion of the energy transmitted through water is absorbed by the trays. The trays heat at a faster rate achieving the highest temperatures due to its high absorptivity character. Convective heat energy exchange occurs from the tray surface to the water in the trays and similarly from

the tray to the ambient. The model is evaluated for water depths ranging from 0.5 cm to 2.0 cm. The thermal network of the solar still is depicted in Figure 1.

The energy balance equation at node 1 (condensate cover) comprises of the energy received from the sun, condensation of vapors, convective and radiation heat transfer from water, energy accumulated in the form of specific heat and the energy lost to the ambient. Similarly at node 3 (trays) the energy is received from the sun and accumulated in the form of specific heat, while convective energy is lost both to water and ambient. The energy balance at node 2 (water) constitutes the energy received from the sun and the trays, accumulating in the form of specific heat. It constitutes the evaporative energy producing the vapors and the convective and radiation heat loss to the glass cover. The energy balance equations for the glass (node 1), water (node 2), and trays (node 3) are expressed in matrix form, as presented in Equation 1. The temperature of tray, water, glass and the distillate output for unit area of the tray from the stepped solar still can be predicted by solving this matrix.

$$\begin{bmatrix} 0 & -h_{twg} & (h_{tga} + h_{twg}) \\ -(h_{tbw}) & (h_{twg} + h_{tbw}) & -h_{twg} \\ 0 & -(h_{tbw}) & (U_b + h_{tbw}) \end{bmatrix} \times \begin{bmatrix} T_g \\ T_w \\ T_b \end{bmatrix} = \begin{bmatrix} I_g(t) - m_g C_{pg} \frac{dT_g}{dt} + h_{tga} T_a \\ I_w(t) - m_s C_{pw} \frac{dT_w}{dt} \\ I_b(t) - m_b C_{pb} \frac{dT_b}{dt} + U_b T_a \end{bmatrix} \quad (1)$$

Where total heat transfer coefficient between water and glass (h_{twg}), between glass and ambient (h_{tga}) and basin and water (h_{tbw}), and can be found out from the following equations [33],

$$h_{twg} = 0.884 \left[(T_w - T_g) + \frac{(P_w - P_g)(T_w + 273)}{268.9 \times 10^3 - P_w} \right]^{1/3} + \frac{(16.28 \times 10^{-3}) h_{cnp} (P_w - P_g)}{(T_w - T_g)} + \frac{\epsilon_{eff} \sigma [(T_w + 273)^4 - (T_g + 273)^4]}{(T_w - T_g)} \quad (2)$$

$$h_{tga} = \frac{[\epsilon_g \sigma (T_g + 273)^4 - (T_{sky} + 273)^4]}{(T_g - T_{sky})} + (2.8 + 3.0 V_w) \quad (3)$$

$$h_{tbw} = f(h_{cbw}, h_{cwbw}) \quad (4)$$

Experimental Setup

Various configurations of the stepped solar still and the characteristics that enhances the distillate output is discussed in our earlier work [27]. Studies of solar still using flat, convex and convex stepped trays reveal that, the productivity of the still has improved compared to single basin due to the increase in the surface area of contact between the water and basin surface [34]. It can be observed that the still with tray width of 120 mm produced 26.9%, 11.7% and 5.7% respectively more distillate output compared to that of tray widths 100, 110 and 130 mm [25].

The stepped tray was designed to maintain a constant air cavity between the central axis of each tray and the condensate cover. Compared to the conventional still, the cavity space between the trays and the condensate cover

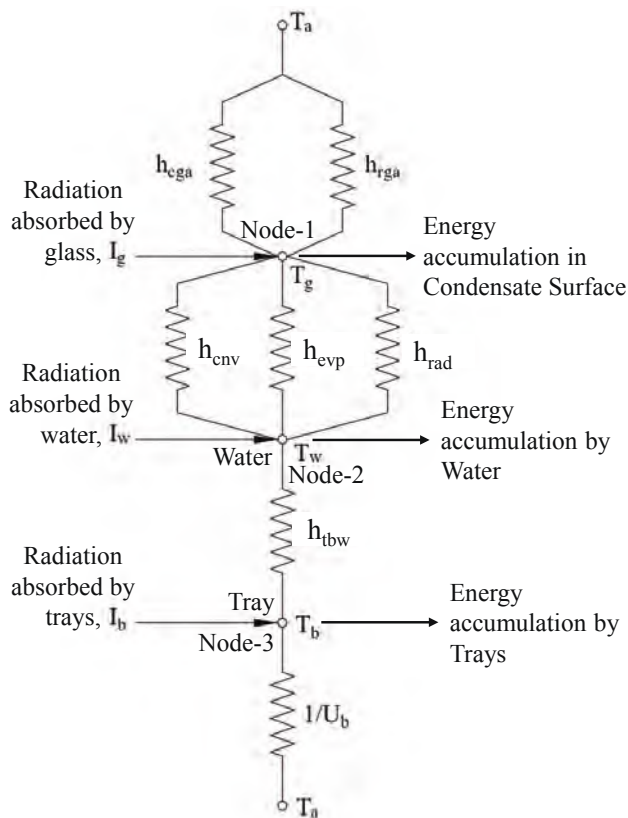


Figure 1. Thermal network and temperature nodes in the solar still.

in stepped still is comparatively less. Thus a stepped solar still with extended type tray made of galvanized steel is selected for the proposed experiments. For obtaining a total projected area of 0.5 m², a set of five trays with dimensions 1000 x 20 x 120 mm was used to build the stepped still. Each tray could hold a maximum capacity of 2400 ml of water. Trays were painted black to increase its absorptivity to 0.9.

The difference in temperature between the water in the trays and the condensate cover is one of the governing parameters in improving the cumulative output from the still [35]. The specific height of the still equals the distance between the center of the absorber surface and the condensate surface. For achieving higher distillate output, evaporating and condensing surfaces are required to be parallel [36]. Reducing the distance between the water surface and condensate cover increases the convective heat transfer between the duo and improves the daily productivity [37]. This is evident from the Dunkle equation and it enhances the convective heat transfer coefficient [38]. Thus a specific height of 120 mm with a cavity ratio of 4.16 was selected for the still design.

A transparent glass cover of 4 mm thick was used as the condensate surface to ensure optimal sunlight reception during the bright sunny days in Kochi, India (9.93°N,

Table 1. Dimensions of stepped solar still

Design Element	Parameter	Value
Still Box	Length (m)	1.12
	Front wall Height (m)	0.21
	Rear Wall height (m)	0.54
	Projected Area (m ²)	0.77
Tray	Length (m)	1.0
	Width (m)	0.12
	Height	0.02
	Projected Area (m ²)	0.52
	Mass / Tray (Kg)	2.89
	No. of Steps/Trays	5
Glass	Thickness (mm)	4
	Area (m ²)	0.645
	Mass (Kg)	6.45
	Angle	25°
Insulation	Thickness (m)	0.04

76.26°E) as shown in Figure 2. Glass inclination was set at an angle of 25° to maintain an equal cavity space between the cover and the consecutive trays. The reflectivity and transmissivity of the selected glass is 0.05 and 0.9.

A graduated steel scale is fixed onto the middle tray to measure the level of water. Table 1 provides the dimensional specifications for the stepped solar still employed in this study. Saw dust was used as insulation with 4 cm thickness. Provisions are provided to ensure uniform supply of feed water at constant heads into each of the trays.

Experimental Procedure

Experiments were conducted in the sunny days of March to May 2024 at Kochi, India with coordinates 9.93°N, 76.26°E. Each day, the tests commenced at 08.00 am and extended beyond dusk till midnight or when the yield ceases. The orientation of the stepped solar still was located facing south, since the location of Kochi is above the equator. Experiments were conducted with three different initial depths of water (dw) 0.5 cm, 1.0 cm and 1.5 cm. Each case of dw was tested by different conditions of feed water addition including (1) no feed water added, (2) 100 ml of feed water added into each tray at 09.00 am, 10.00 am and 11.00 am and (3) 200 ml of feed water added into each tray at 09.00 am, 10.00 am and 11.00 am. In addition, 400 ml feed water was added to initial depths of 0.5 cm and 1.0 cm to study its sensible storage effects. Surface cleaning of the condensate cover was performed daily to prevent any hindrance of heat transfer through the glass due to the presence of foreign elements.

Thermocouples coupled with a 4-channel data logger was used to measure the basin, water and glass temperatures, while the ambient temperature was measured using

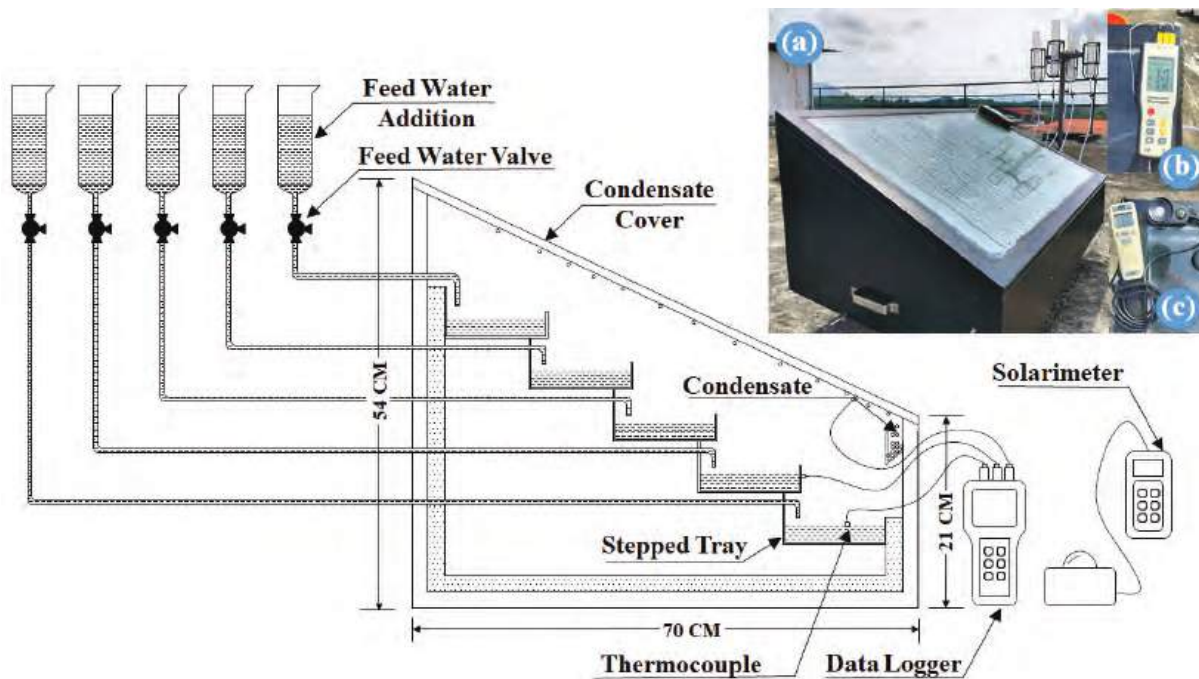


Figure 2. Schematic diagram of experimental setup for extended tray type solar still (Insert image of (a) stepped solar still with feed water system, (b) data logger, (c) solarimeter).

Table 2. Combinations of depth of water and feed water quantities

Depth combination	Initial Depth of Water (cm)	Quantity of water (ml) added at each hour given		
		09.00 AM	10.00 AM	11.00 AM
dw _{0.5}	0.5	0	0	0
dw _{0.5-100}	0.5	100	100	100
dw _{0.5-200}	0.5	200	200	200
dw _{1.0-0}	1.0	0	0	0
dw _{1.0-100}	1.0	100	100	100
dw _{1.0-200}	1.0	200	200	200
dw _{1.5-0}	1.5	0	0	0
dw _{1.5-100}	1.5	100	100	100
dw _{1.5-200}	1.5	200	200	200

Table 3. Maximum quantity of water in the stepped still for different dw and feed water input, assuming no evaporation from the tray

Initial Depth of Water (cm), dw	Feed water quantity added hourly per tray at 09.00 am, 10.00 am and 11.00 am			
	0 ml	100 ml	200 ml	400 ml
Final Quantity of water in the trays after feed water addition				
0.5	3000	4500	6000	9000
1.0	6000	7500	9000	12000
1.5	9000	10500	12000	-

Table 4. Measuring instrument specifications

Sl. No.	Measuring Instrument	Range	Accuracy	Standard Uncertainty (σ)
1	Solarimeter (MECO)	0-1000 W/m ²	±1 W/m ²	0.5773
2	Multi-channel digital temperature indicator	-50 °C to 110 °C	0.1 °C	0.0577
3	Calibrated Thermocouples	0 °C to 100 °C	±0.5 °C	0.2886
4	Measuring Flask-2	0 - 30 ml	5 ml	0.0028
5	Anemometer (MECO - 961P)	1- 25 m/sec	±0.2 m/sec	0.1154

a conventional thermometer. Solar radiation intensity was recorded using the solar meter (MECO, India), placed on the inclined surface of the glass. The air velocity was recorded using an anemometer (MECO, India) and the distillate output was measured using graded bottles.

The measurement uncertainties were determined based on the specifications provided by the instrument manufacturers. Error analysis was conducted for a solarimeter, a multi-channel data logger, thermocouples, and an anemometer. The errors could get accumulated due to the limited precision of the instruments, human errors and uncertainty of the measuring technique during the experimentation. The error values of each measurement were calculated from the least count of the instrument and the minimum output to be measured and are reported in Table 4. Experimental uncertainty is calculated by considering the standard uncertainties of each measuring instrument. The standard uncertainties provides the possible errors in the experimental values measured [39,40]. The mathematical expression for percentage of uncertainty is given by [41],

$$\% \text{ Uncertainty} = \frac{U_i}{B} \times 100 \quad (5)$$

$$U_i = \frac{\sqrt{\sigma_1^2 + \sigma_2^2 \dots + \sigma_n^2}}{n^2} \quad (6)$$

$$\sigma = \frac{\sqrt{\sum (X - \bar{X})^2}}{n_o} \quad (7)$$

The percentage of experimental uncertainty was estimated to be less than 1.5%. Water in the trays absorb a portion of the solar radiation received by the stepped still. A part of energy is used for evaporation of water molecules and other part of energy is stored in the form of sensible heat. The magnitude of solar energy stored depends on the mass of water in the trays and temperature rise. With less quantity of water in the trays, most of the absorbed energy is utilized for the phase change of water molecules. An increase in the initial depth of water in the trays shall also delay the evaporation process [19].

RESULTS AND DISCUSSION

As indicated in Table 2, a predetermined measured quantity of feed water is added at 09.00 am, 10.00 am and 11.00 am to three initial depth conditions including 0.5, 1.00 and 1.5 cm. The maximum quantity of water that the still can accommodate is limited to 12000 ml. The maximum quantity of water in the trays after the addition of feed water under the circumstance of extreme cloudy and rainy days when no evaporation occurs is depicted in Table 3.

Effect of Solar Radiation Intensity

The variation of solar radiation intensity with time is plotted in Figure 3 for nine different combinations of depth of water and feed water quantities (Table 2). It is noticeable that solar stills under different experimental combinations received identical solar energy inputs throughout the day of the experimentation. The still performance was dependent on the intensity of solar radiation. Thus peak hourly productivity was achieved just at after noon (between 12.00 pm and 01.00 pm), when the solar radiation intensity was maximum. The peak of radiation intensity was received during noon and the maximum and minimum cumulative radiation received during the days of study were 19.56 MJ/m² and 21.95 MJ/m². The average value of cumulative solar energy received during the first hour (08.00 am to 09.00 am) from the commencement of experiment was 1.715 MJ/m² and the maximum average energy received between 11.00 am and 12.00 pm were 3.09 MJ/m². This increasing trend in cumulative energy ensures the feasibility to increase the quantity of water in the tray to store the increased input energy radiated from the sun.

Effect of ambient conditions

The average ambient conditions of the still during the days of experimentation, ranges from 30.9°C to 32.2°C. The ambient temperature characteristics followed a similar trend that of the solar radiation as shown in Figure 4. The curve profiles of ambient temperature and solar radiation were similar, indicating the dependency of ambient temperature on the energy received from the sun. The increased ambient temperature during the noon session was favorable in reducing the heat loss between the condensate cover and the ambient. During these days the highest ambient temperature recorded was 37.9°C at 02.00 pm.

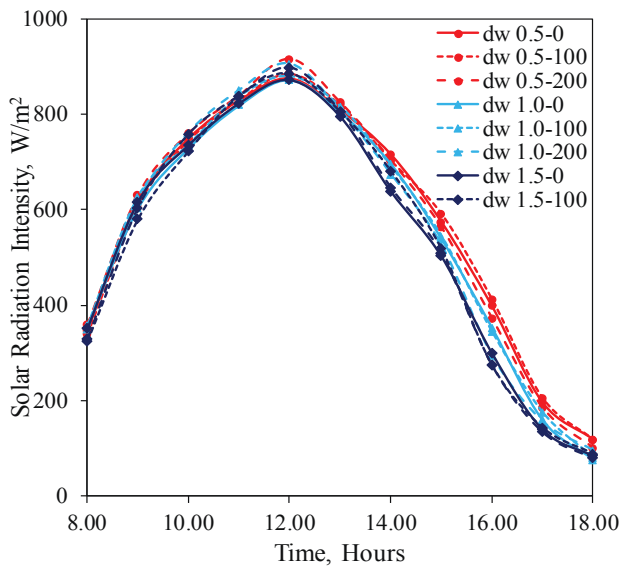


Figure 3. Variation of intensity of solar radiation for different initial depth conditions and feed water input.

The wind speed varied throughout the day and was measured hourly between 08.00 am and 12.00 midnight. The average velocity during the experimentation period varied between 4.88 m/sec and 9.66 m/sec. For any given day, the highest average wind velocity was during the period between 12.00 pm and 06.00 pm, followed by the nocturnal period between 06.00 pm and 12.00 am. Higher wind velocity during the noon period, improved the cooling of condensate cover, resulting in better condensation of vapors and augmented productivity. According to El-Sebaai [42], the productivity of conventional stills decreases as the wind velocity increases when the mass of water in the basin is below the critical mass of 45 kg. In our cases, an initial depth of water of 0.5 cm and 1.0 cm is the conditions attributed to the quantity of water in the still falls below the 45 kg. For every other case the level of water in the stills remained above 45 kg and hence the wind velocity does not have a negative effect on the yield.

Influence of depth of water on still productivity with no feed water addition

During these set of experiments, no feed water was added into the trays for compensating the quantity of water evaporated during each hour. Hence, the level of water in the tray descends as the day progresses. The rate of descend in the trays depend on the intensity of solar radiation received and the initial depth of water.

The trays have a projected area of 0.52 m² and with an initial water depth of 0.5 cm, the cumulative distillate output produced from 9.00 am to 1.00 pm was 14.2% and 31.6% higher than that of the initial water depth 1.0 cm and 1.5 cm respectively. By maintaining lower quantity of water in the tray, less energy is required to augment its temperature rise.

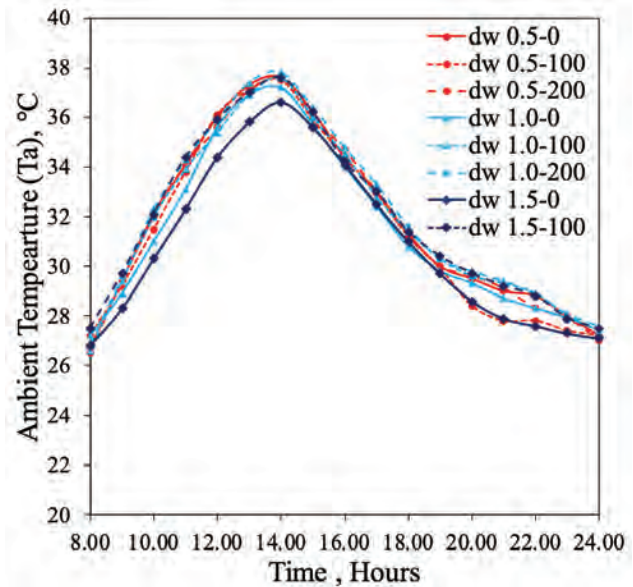


Figure 4. Variation of ambient temperature for different initial depth and feed water input.

As the depth of water increases more energy is required to raise the temperature of water. With low depth of water, the initial temperature rise is more, favoring the achievement of maximum temperature at noon, compared to higher depth of water in the trays. This resulted in higher productivity during the morning hours, as illustrated in Figure 5.

With a smaller volume of water in the trays, there is less thermal inertia, leading to a quicker increase in temperature at the beginning of the evaporation process. Lower depths of water had improved heat transfer coefficients resulting in increased productivity. Conversely, a low volume of water means reduced storage of specific heat energy, causing the water temperature to decrease at a faster rate as solar radiation intensity diminishes after 02.00 pm resulting in decreased productivity. From 02.00 pm to 06.00 pm the cumulative yields for $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ are 475 ml, 585 ml and 585 ml. The depths $dw_{1.0}$ and $dw_{1.5}$ produced 23% more output compared to the depth of $dw_{0.5}$.

By collecting the nocturnal yield produced by the still, there has been an improvement in their productivity. When more water is in the trays, it stores more energy as sensible heat, allowing it to retain a higher temperature for an extended period, even as solar input diminishes. The temperature difference between water and glass for $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ at 6.00 pm are 4.4°C, 6.3°C and 10°C respectively. This variation in temperature resulted in nocturnal yields of 15 ml, 40 ml, and 130 ml for trays with water depths $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ respectively, each covering a projected area of 0.5 m².

The total daytime yield is the sum of the hourly yield collected from the still each hour starting from 09.00 am to 06.00 pm. Since the experiment is initiated at 08.00 am no hourly yield is produced at this hour. The total daytime

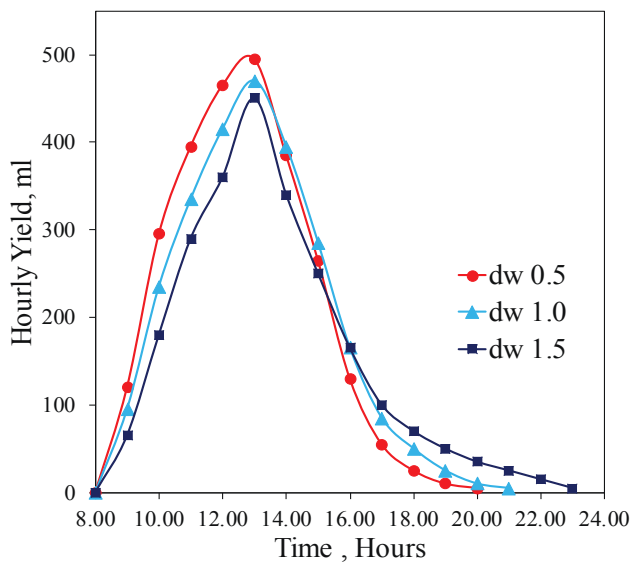


Figure 5. Hourly distillate yield from the solar still for different initial depths of water without any feed water input.

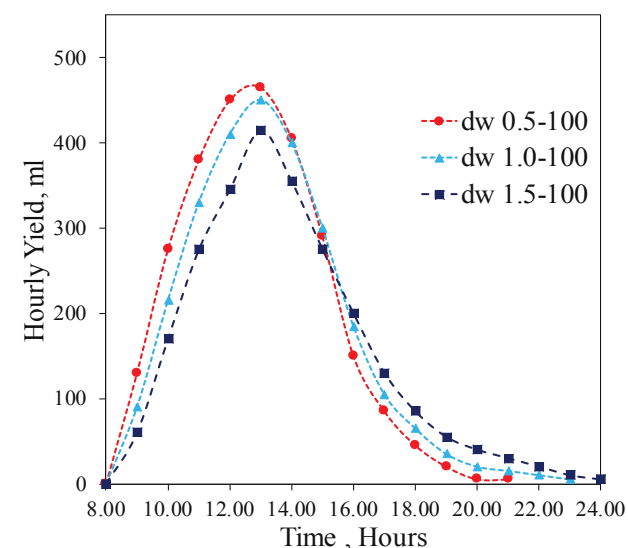
productivity for stills with $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ conditions was 5260 ml, 5060 ml and 4540 ml respectively, for a projected area of one square meter. This clearly indicates that, as the depth of water increases, the productivity of the stepped solar still decreases. Incorporating nocturnal yield into the daytime productivity of the stepped still increased its output by 0.5%, 1.5%, and 5.7% for $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ respectively, relative to the daytime yield. The overall yield increases for stills with low initial depths of water. For initial depth of water $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ the daily

distillate output is 5290 ml/m², 5140 ml/m² and 4800 ml/m² respectively.

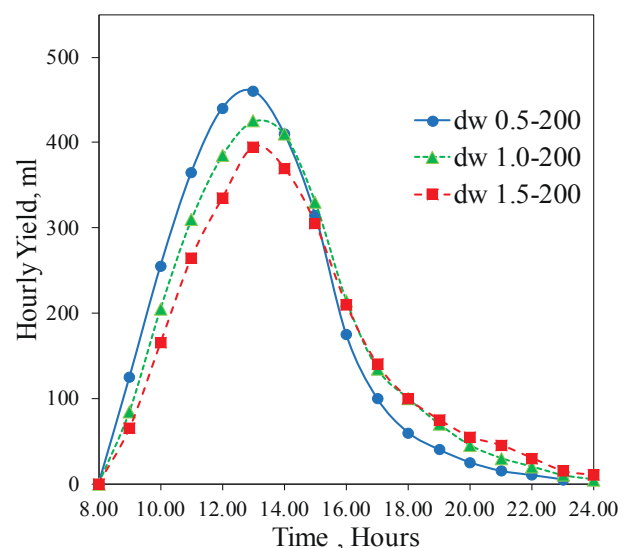
Influence of depth of water on still productivity with input feed water

During this experimental phase, 100 ml and 200 ml of water were introduced into each tray of the stepped still at 9:00 am, 10:00 am, and 11:00 am, respectively. The addition of feed water causes a slight drop in the temperature of the water in the trays, which is affected by the quantity and temperature of the feed water introduced. As solar radiation intensity decreases, the water temperature in the trays follows a corresponding decline. To mitigate the temperature drop caused by feed water addition, its introduction is restricted until 11:00 am. The strategy of adding water aims to harness excess heat received by the still in the form of sensible heat, leveraging the increasing solar radiation intensity during the morning hours. Experiments were conducted with 100 ml and 200 ml of feed water addition for the three different depth of water viz. 0.5 cm, 1.0 cm and 1.5 cm.

Experimental studies by [13] clearly indicate that, low initial depth of water in the trays reduces the thermal inertia of water mass resulting in an improvement in the yield. The tray with minimum initial depth of water produced the highest yield in the first hour of the experimental study. At 09:00 am, with the addition of 100 ml feed water, the yields produced by initial depths 0.5 cm, 1.0 cm and 1.5 cm were 130 ml, 90 ml and 60 ml respectively. From 09:00 am to 01:00 pm, the cumulative productivity for $dw_{0.5-100}$ and $dw_{0.5-200}$ slightly declined by 3.9% and 7.0% respectively compared with the scenario where no feed water was added



(a)



(b)

Figure 6. Hourly distillate yield, with (a) 100 ml of feed water added to each tray, (b) 200 ml of feed water added to each tray

as shown in Figure 6(a) and 6(b). This decline in productivity is due to the increased volume of water in the trays due to feed water addition.

Beyond 1.00 pm, with the declining solar radiation intensity the temperature drastically drops for trays with lesser volume of water. Addition of feed water during the forenoon session augments the quantity of water in the trays enhancing the specific heat storage capacity. The specific heat energy stored in the water is released to compensate the temperature decline due to reduced radiation intensity received by the still. Maintaining higher water temperature in the afternoon session helped to prolong the yield in the nocturnal hours. Hence in the afternoon session the trays with higher quantity of water produced more distillate output. Compared to the trays without feed water addition, by adding 100 ml feed water in each tray, the productivity improved by 2.3%, 2.5% and 2.9% for $dw_{0.5-100}$, $dw_{1.0-100}$ and $dw_{1.5-100}$ respectively as shown in Figure 6(a) and 6(b). Similarly, for 200 ml water addition, the productivity of the stepped still improved by 5.9%, 8.2% and 7.5% respectively. Hence as the initial depth of water increased, the productivity also reduced.

Effect of varying the quantity of feed water added

Conventional practice among researchers involves feeding water into solar stills at hourly intervals to sustain a constant water level in the trays or basin. When no compensating feed water is supplied, the level of water declines, and thus results in lower quantity of water in the trays. Consequently, a smaller amount of sensible heat can be stored by the tray water, rendering it incapable of generating nocturnal yield. Notably, this investigation deviates from established research protocols by incorporating a feed water addition strategy that exceeds the quantity of evaporated water.

From Figure 7, it can be noted that, level of water in the tray rises on an hourly basis till 11.00 pm as the feed water is added to the trays. After 11.00 pm, no feed water is added and the depth of water in the tray descends as evaporation continues. The augmented quantity of water facilitated the storage of more energy in the form of sensible heat. This ultimately helps to improve the yield during the afternoon hours and the nocturnal period. Whereas in the case of $dw_{0.5}$, as the evaporation continues the depth of water in the trays descends when no feed water is added.

The hourly productivity of the stepped still with 0.5 cm initial depth, with feed water quantity ranging from 0 ml to 400 ml is depicted in Figure 8. For low initial quantity of water in the trays, the evaporation commenced early and the temperature rise of water was at a faster pace. Upon introducing the feed water, the mixing of hot and cold fluids, along with the increased quantity in the tray, reduced the rate of temperature rise. Augmentation in the volume of water in the trays, led to a minor decrease in the overall distillate generated from 9:00 am to 1:00 pm. This decline is due to the decrease in temperature of tray water due to feed water addition. The accumulated distillate yield from 9.00 am to 1.00 pm for the stepped still with initial depth of 0.5 cm with hourly feed water quantity of 0, 100, 200 and 400 ml are 3540, 3400, 3290 and 3240 ml/m² respectively.

In contrast, there was an enhancement in the cumulative yield observed between 2:00 pm and 6:00 pm, as well as during the nocturnal hours. This improvement can be attributed to the release of energy stored as sensible heat energy from the water in the trays. The accumulated yield between 02.00 pm and 06.00 pm for $dw_{0.5-100}$, $dw_{0.5-200}$ and $dw_{0.5-400}$, were 975 ml, 1060 ml and 1220 ml respectively. Compared to no feed water addition, the improvement in

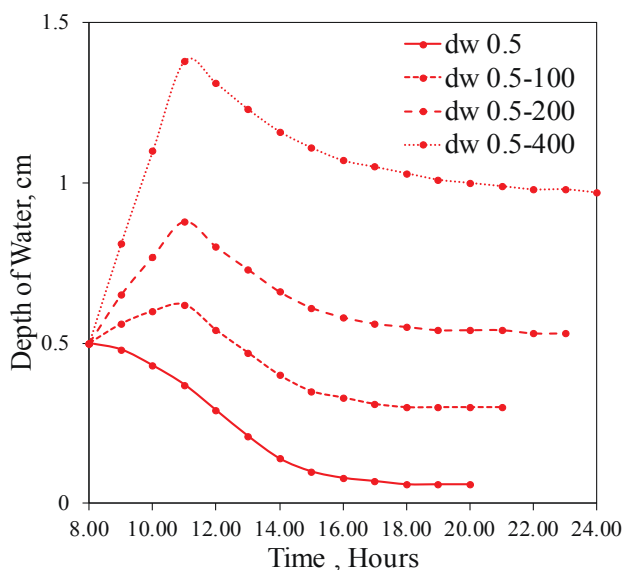


Figure 7. Hourly variation in water level for diverse feed water inflows at initial water depth 0.5 cm.

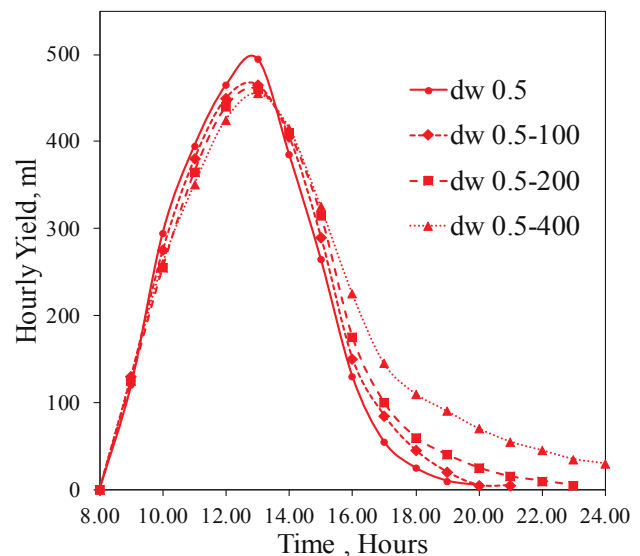


Figure 8. Hourly distillate yield for 0.5 cm initial depth and at different combinations of feed water input to each tray.

afternoon yield for $dw_{0.5-100}$, $dw_{0.5-200}$ and $dw_{0.5-400}$ were 13.3%, 23.2% and 41.8% respectively.

At 06.00 pm, the temperature of water for $dw_{0.5-100}$, $dw_{0.5-200}$ and $dw_{0.5-400}$ were 35.7°C, 38.2°C, 39.1°C and 41.4°C respectively. Correspondingly the nocturnal output obtained were 15 ml, 30 ml, 95 ml and 325 ml respectively. The distillate output for $dw_{0.5}$, $dw_{0.5-100}$, $dw_{0.5-200}$ and $dw_{0.5-400}$ are 5290 ml/m², 5410 ml/m², 5600 ml/m² and 6330 ml/m² respectively. The daily output improved by 19.7% as the quantity of water added was increased from 0 ml to 400 ml for an initial depth of 0.5 cm.

Temperature of water in the stepped still

The specific heat storage capacity is dependent on the water temperature and quantity of water in the trays. The sensible energy stored in the water is released when the radiation intensity declines. This released energy partly compensates for the reduction of energy received from the sun, hence contributing to the evaporative heat transfer coefficient.

The energy received by $dw_{0.5}$, $dw_{0.5-100}$, $dw_{0.5-200}$ and $dw_{0.5-400}$ are 21.6, 21.7, 21.7 and 22 MJ/m² respectively. The temperature rise from 08.00 am to 9.00 am was almost the same for all the combinations, since they all started with the same initial depths of water and with similar energy received. The temperature difference of water between 08.00 am and 01.00 pm for $dw_{0.5}$ and $dw_{0.5-400}$ were 42.2 °C and 37.6°C respectively. Thus the rate of increase of temperature was more with trays with lesser quantity of water. Again from Figure 9, it can be seen that as the quantity of feed water increased, the peak temperature also decreased. For $dw_{0.5}$ and $dw_{0.5-400}$, the peak temperatures obtained were 69.1°C and 65.3°C respectively. At 01:00 pm, the water volumes in

trays $dw_{0.5}$ and $dw_{0.5-400}$ were 1230 ml and 7380 ml, respectively. The stepped still $dw_{0.5-400}$ carries at least five times the quantity of water than that of $dw_{0.5}$. Consequently, it is evident that more energy is needed to elevate the temperature of water as the water mass increases, hence resulting in lower peak temperature for $dw_{0.5-400}$ compared to $dw_{0.5}$.

At 06.00 pm with minimal solar radiation intensity, the temperature of water for $dw_{0.5}$ and $dw_{0.5-400}$ is 35.7°C and 41.4°C. The volume of water in $dw_{0.5}$ and $dw_{0.5-400}$ at 06.00 pm are 370 ml and 6160 ml respectively. Ultimately, a higher water temperature is observed with a larger quantity of water, indicating the influence of sensible heat. Since $dw_{0.5-400}$ possesses higher mass of water than that of $dw_{0.5}$ at 06.00 pm, the temperature of water is 16% higher due to the energy storage in the form of specific heat. It augmented the hourly yields during the nocturnal hours due to the higher stored sensible heat energy and temperature of water at 06.00 pm.

Validation of the experimental results with the theoretical model

The heat balance matrix of the solar still, represented by Equation 1, was resolved to predict the temperatures of the basin, water, and glass. The predicted temperatures were compared with experimental results as shown in Figure 10, for an initial water depth of 0.5 cm and a feed water addition of 100 ml. Generally, the theoretical predictions were found to be slightly higher than the actual values. The maximum percentage deviations between predicted and actual temperatures for the basin, water, and glass were 3.9%, 3.8%, and 4.3%, respectively.

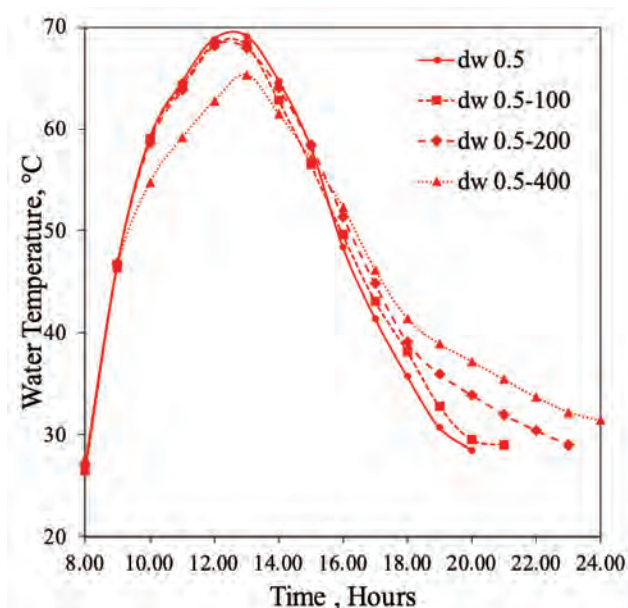


Figure 9. Temperature variation of water in the tray stepped solar still on an hourly basis.

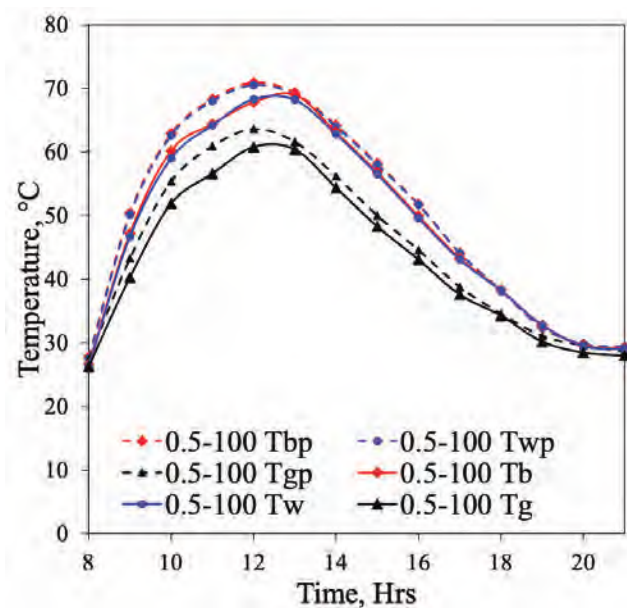


Figure 10. Comparison of predicted temperature of basin, water and glass with experimental results for $dw_{0.5-100}$.

Daily efficiency

The theoretical hourly distillate output from the stepped solar still is computed using the relation [33],

$$m_w = \frac{h_{evp} \times (T_w - T_g) \times \Delta t}{L_{ev}} \quad (8)$$

Where the evaporative (h_{evp}) and convective (h_{cnv}) heat transfer coefficients are directly dependent on the temperature difference between the water and the condensate cover, which are given as [33],

$$h_{evp} = 16.28 \times 10^{-3} \times h_{cnv} \times \frac{(P_w - P_g)}{(T_w - T_g)} \quad (9)$$

$$h_{cnv} = 0.884 \left[(T_w - T_g) + \frac{(P_w - P_g) \times (T_w + 273.16)}{((268.9 \times 10^3) - P_w)} \right]^{1/3} \quad (10)$$

The hourly efficiency of the still is determined by taking into account the average solar energy input during the hour, the latent heat of vaporization, and the hourly distillate output [40]. Similarly, the daily thermal efficiency of the still is a function of solar radiation intensity, distillate production, and the amount of water in the still, as formulated in [33].

$$\eta = \frac{\sum m_w \times L_{ev}}{A_t \times \sum I(t) \times \Delta t} \quad (11)$$

The theoretical thermal efficiency can be computed by substituting the theoretical value of distillate quantity calculated using Equation 8 in Equation 11. While the actual

daily efficiency can be calculated using the measured hourly distillate quantities.

The hourly efficiency can be calculated by substituting the quantity of distillate produced during an hour into Equation 11. As depicted in Figure 11, the maximum hourly efficiencies obtained without feed water replenishment were 76.4% at 0.5 cm, 72.8% at 1.0 cm, and 70.4% at 1.5 cm initial water depths. These peak efficiencies occurred at 01:00 pm, corresponding to the maximum hourly yields. The efficiency of the dw_{0.5-400} still arrangement, which received 400 ml of feed water, was found to be 13.5% higher than that of the dw_{0.5} arrangement without feed water addition. The enhancement in efficiency can be attributed to the improved heat storage capacity, primarily in the form of sensible heat.

In Figure 12, for each initial depth of water, the daily efficiency improved with the augmented feed water input. The daily efficiency includes both the daytime and nocturnal yield, while the daytime yield only includes the cumulative yield during the sunshine hours. The daily efficiency consistently surpassed the daytime efficiency as seen in Figure 13, due to the inclusion of the nocturnal yield. For initial water depths of 0.5 cm, 1.0 cm, and 1.5 cm, the improvement in daily yield compared to daytime yield continued to increase as the amount of added feed water increased.

Utilization of received energy and improvement of daily efficiency can be enhanced with the addition of feed water to a given initial depth of water. Considering nocturnal yield, the daily efficiency at an initial water depth of 0.5 cm showed improvements of 0.32%, 0.64%, and 2.02% for feed water additions of 0 ml, 100 ml, and 200 ml, respectively. Similarly as the initial depth of water was increased, there was an improvement in the nocturnal yield produced. For no feed water condition, the daily efficiency improvements

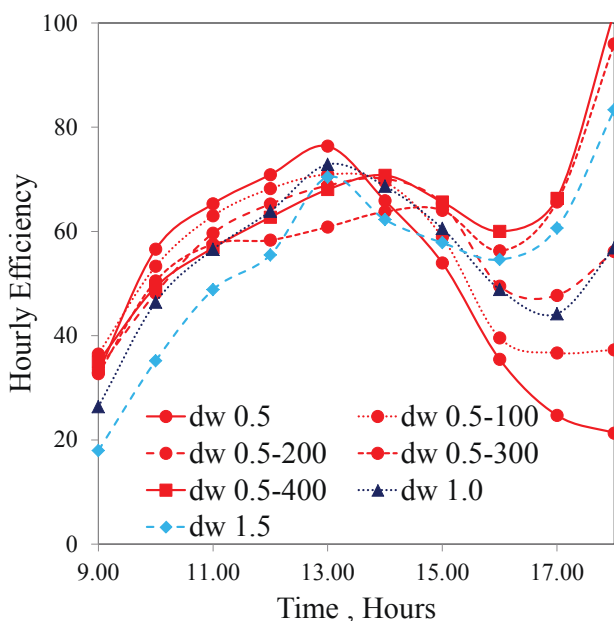


Figure 11. Hourly efficiency of stepped still for different water depth combinations.

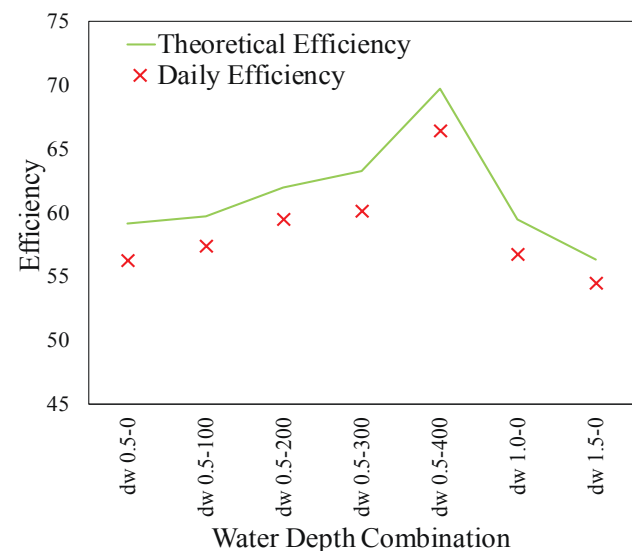


Figure 12. Stepped still efficiency –validation of experimental values.

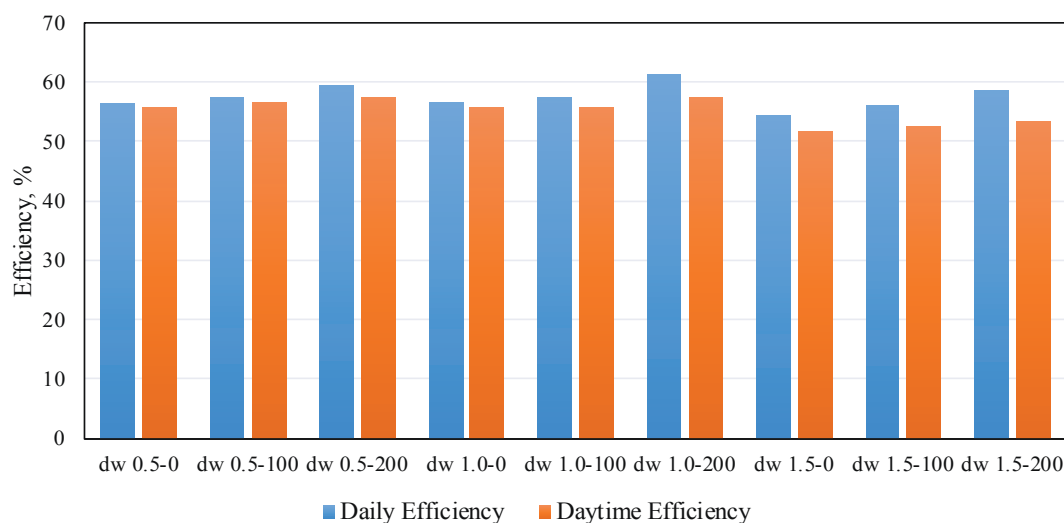


Figure 13. Daily efficiency of the stepped solar still for different quantities of feed water.

were 0.32%, 0.88%, and 2.95% for initial water depths of 0.5 cm, 1.0 cm, and 1.5 cm, respectively. With the addition of 200 ml of feed water, the improvements in daily efficiency over daytime efficiency were 2.02%, 3.97%, and 5.21% for $dw_{0.5-200}$, $dw_{1.0-200}$ and $dw_{1.5-200}$ respectively. While for the initial depth of water at 0.5 cm with 200 ml and 400 ml feed water addition, the improvement in daily efficiency over daytime efficiency was 2.02% and 6.82% respectively.

The highest daily efficiency of 66.45% was obtained for $dw_{0.5-400}$ (Figure 12). With $dw_{0.5-100}$ and $dw_{0.5-400}$, the efficiency of the stepped still improved by 1.12% and 10.22% respectively. With no feed water addition, as the initial depth increased from 0.5 cm to 1.5 cm, the daily efficiency declined from 55.91% to 51.56%. At lower initial depths, the thermal heat capacity is low which facilitates better rate of temperature rise, hence improves the daily productivity.

The validation of experimental values were performed by comparing the efficiencies with the theoretical energy balance equations. The energy balance equations from the literature [33] were utilized to predict the temperatures of both tray and condensate cover. The thermal efficiency of the still is the ratio of the energy required to evaporate the daily distillate output to the energy received from the sun for a given period of time. These predicted temperatures were then used to calculate the evaporative heat transfer coefficient and hence the theoretical efficiency. Likewise, the daily efficiency of the still was calculated using the experimental temperature values. Theoretical efficiency of the stepped still was compared with the actual efficiency as shown in Figure 12. Both the theoretical and daily efficiencies followed similar trends. The theoretical efficiency values of the stepped still support the experimental findings explained in this work. The percentage change between the theoretical and actual efficiency is less than 6%. The daily efficiency improved with the addition of feed water quantity, whereas daytime efficiency declined with the increase

in initial depth of water. From Figure 13, it can be recorded that the addition of nocturnal yield improved the efficiency of the stepped still.

Daily Cumulative Yield

The daily cumulative yield without feed water addition for $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ are 5290 ml, 5140 ml and 4800 ml respectively. The daily cumulative yield is the sum of daytime cumulative yield and nocturnal cumulative yield. The daily cumulative yield without feed water addition for an initial depth of 0.5 cm is 3% and 10.2% greater than that of initial depth of 1.0 cm and 1.5 cm respectively as shown in Figure 14. The nocturnal yield produced for $dw_{0.5}$, $dw_{1.0}$ and $dw_{1.5}$ were 30 ml, 80 ml and 260 ml respectively. Despite

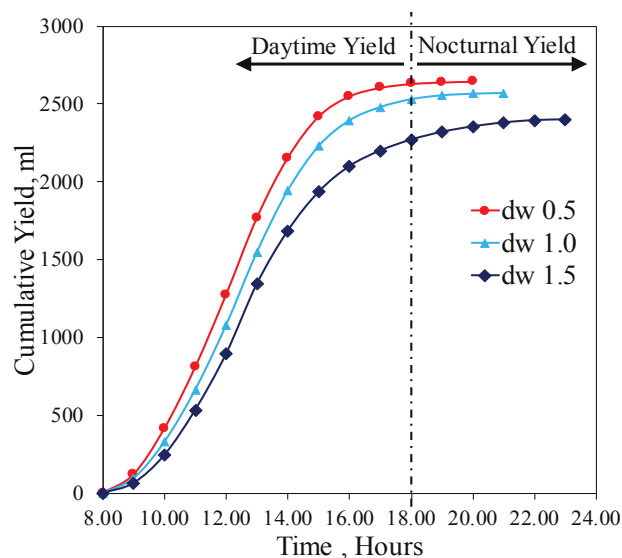


Figure 14. Daily cumulative yield for initial water depth of 0.5 cm, 1.0 cm and 1.5 cm without feed water inlet.

the prolonged nocturnal yields observed with deeper water depths, the cumulative daily yield for $dw_{0.5}$ was 3% and 10.2% higher than $dw_{1.0}$ and $dw_{1.5}$, respectively.

With the addition of feed water into the trays, there was an improvement in the daytime and nocturnal productivity of the stepped still as shown in Figure 15(a-c). Three different depths of water exhibited a uniform trend, wherein the nocturnal productivity increased in correlation with the augmentation of feed water quantity. With the feed water addition of 100 ml and 200 ml for a depth of 0.5 cm, the daytime yield improved by 1.7% and 2.8% respectively, when compared to no feed water addition. Similarly the nocturnal yield also improved from 30 ml to 60 ml and

190 ml with the addition of 100 ml and 200 ml feed water respectively. With the addition of feed water, the daily yield improved in comparison to no feed water addition.

For each initial water depth, the maximum cumulative yield has achieved when 200 ml of feed water was added, followed by 100 ml and 0 ml. Adding 100 ml per tray to 0.5 cm, 1.0 cm and 1.5 cm water depth improved the day time yield by 1.7%, 0.8% and 1.7% respectively, compared to no feed water addition. Similarly for 200 ml feed water addition per tray to an initial depth of 0.5 cm, 1.0 cm and 1.5 cm improved the day time yield by 2.8%, 2.7% and 3.5% respectively. The nocturnal productivity for $dw_{0.5-100}$, $dw_{1.0-100}$ and $dw_{1.5-100}$ with 100 ml water addition improved by 60 ml/m²/day, 170

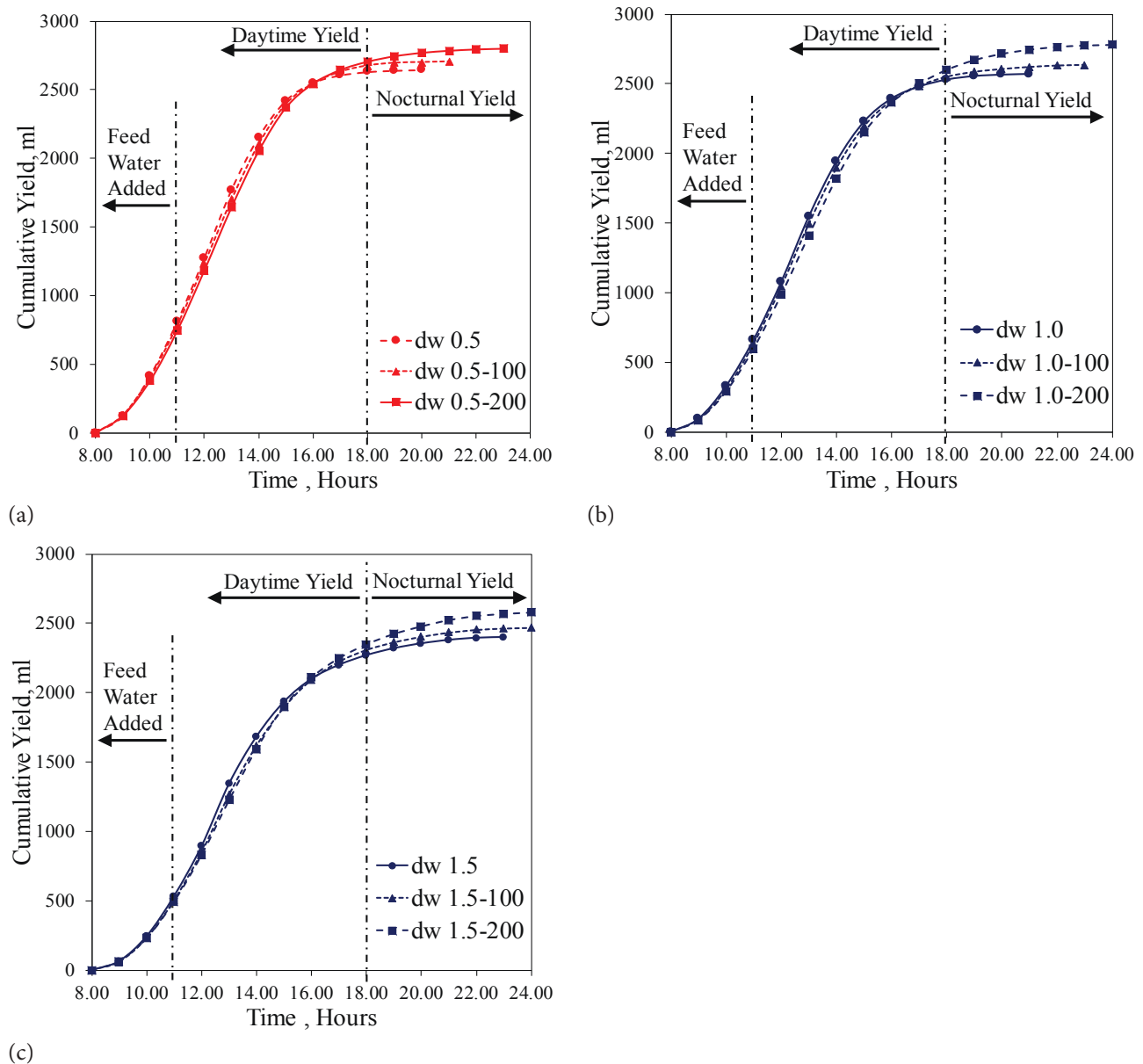


Figure 15. Daily cumulative yield for initial water depth of (a) 0.5 cm, (b) 1.0 cm and (c) 1.5 cm with an input feed water of 100 and 200 ml.

ml/m²/day, 320 ml/m²/day respectively. Similarly for $dw_{0.5-200}$, $dw_{1.0-200}$, and $dw_{1.5-200}$ with 200 ml feed water addition the nocturnal yield improved by 190 ml/m²/day, 360 ml/m²/day and 460 ml/m²/day respectively. Compared to the condition of no feed water addition, the nocturnal yield improved by 100%, 112.5% and 23% for $dw_{0.5-100}$, $dw_{1.0-100}$, and $dw_{1.5-100}$ respectively. While for 200 ml feed water addition, the nocturnal yield improved by 530%, 350% and 77% respectively for $dw_{0.5-200}$, $dw_{1.0-200}$, and $dw_{1.5-200}$.

The stepped still with minimum initial depth of 0.5 cm and maximum feed water inlet of 200 ml per tray produced

the highest distillate output of 5600 ml/m²/day. Lowest productivity of 4800 ml/m²/day was produced by the still having the maximum initial depth of water 1.5 cm and when no feed water was added. Low initial depth of water during the commencement of experiment improved the temperature rise due to the low volume of water in the trays. Gradual addition of feed water into the trays in an hourly manner augmented the water in the trays without compromising much on the temperature rise of water. The increased volume of water in the trays aggravated the energy stored in the form of sensible heat, which again improved the

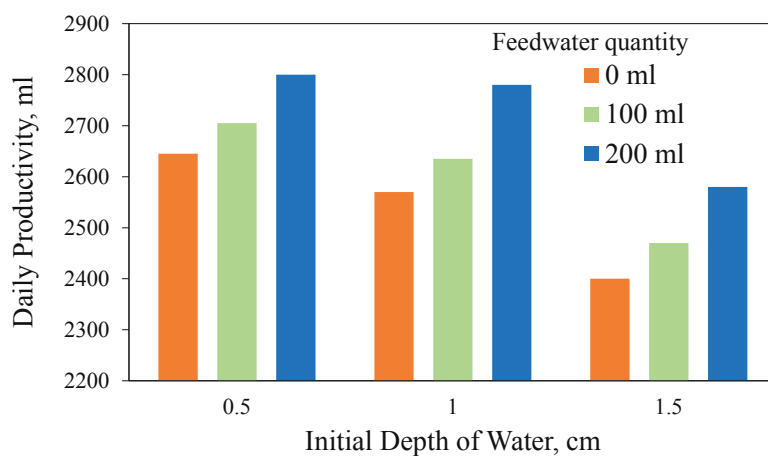


Figure 16. Daily distillate output for water depth of 0.5 cm, 1.0 cm and 1.5 cm with different feed water inlets of 0, 100 and 200 ml.

Table 5. Comparison of cumulative daily yield of various solar stills with different heat storage elements reported in literature

Sl. No.	Still Type	Energy Storage Material	Specific Heat Capacity (J/Kg°C)	Yield (ml/m ² /day)
1	Inclined stepped still [10]	Copper scraps	398	4890
2	Stepped still [11]	Aluminum fillings	900 [52]	5400
3	Stepped still [13]	Fins	460	8900
4	Pyramid solar still [43]	Palmately leaf -Natural fibers	-	5160
5	Stepped double slope [44]	Linen wicks	-	3260
6	Conventional still with V-shaped floating wicks[41]	Black jute cloth	-	6200
7	Stepped concave trays [7]	Sand	830	3780
8	Stepped still [6]	Black sand	830	5750
9	Trays solar still [45]	Black sand	830	5650
10	Conventional still [46]	Fine basalt stones	1230	1075
11	Conventional still [47]	Pumice stones	870	2000
12	Conventional still with preheating [39]	Coarse aggregate	860	4210
13	Stepped concave trays [7]	Concrete pieces	1000	4060
14	Stepped concave trays [7]	Brick	800	3650
15	Proposed work-stepped extended type trays	Water	4179	6330
16	Stepped Still – 120 mm wide tray (extended type) [25]	Water	4179.6	5630
17	Stepped Still – 120 mm wide tray (extended type) [26]	Saline Water	4190	5840
18	Stepped Still – 120 mm wide tray (extended type) with mirrors [26]	Saline Water	4190	6350
19	Inclined stepped Copper Trays [40]	Water	4179.6	4383

nocturnal output. This implies that with the minimum initial depth of 0.5 cm and maximum feed water input of 200 ml improved both the daytime and nocturnal productivity as seen in Figure 16.

As seen from Table 5, different types of heat sensible materials were used in designs of still to improve its productivity. The daily productivity of the stepped still with initial depth of water 0.5 cm and 400 ml feed water addition was 6330 ml.

Economic analysis of freshwater

The projected area of the stepped still is 0.5 m², and the cost parameters considered for the economic analysis is provided in Table 6. The capital recovery factor of the solar still depends on the initial investment made and its changing value of money during its lifetime. Considering the monsoons of Kerala, it is forecasted to receive 300 sunlight days, and the average productivity of the still to be 3000 ml.

It is speculated that, aside from the scrap value of the GI trays, no other material used have some value at the end of 6 years. The Salvage value of the still is considered as the scrap value of the trays used, and is taken as 30% of its initial value. The annual maintenance cost shall include the purchase of paint, sawdust and sealing material and shall be accounted for 15% of the initial annual cost. The total annual cost is given by [48],

$$TAC = IIF + AMC - ASV \quad (12)$$

Cost analysis and comparison studies by [48], indicated that the cost per liter of water produced using still desalination techniques is between 0.0066\$ and 0.2696\$. The market rate of drinking water in India is expected between

0.12\$ and 0.24\$, considering the dollar exchange rate to be 0.012\$/INR.

An economic comparative study was performed to analyze the cost of water produced from different types of stills and is presented in Table 7. From Table 7, the stills having longer salvage period and lower investment costs produced water at a lower price. The CPL for stills using sand as energy storage material is 0.016\$ [6] and 0.015\$ [45], while in this study the CPL is 0.013\$. When no heat storage materials were used the CPL stood slightly higher at 0.021\$ [23] and 0.047\$ [40]. Even with the addition of fins and pebbles to a stepped tray for treating effluent water with a CPL of \$0.32 [49], there was no decrease observed in the CPL. Even though [22] had lesser daily yield compared to this work, the CPL is only 0.01\$/L, due to higher absorber area of the trays, higher solar radiation intensity, higher ambient temperatures, more days of working and longer salvage period.

Considering the hot and wet climates of Kerala, and the material used for the construction of the still box, the salvage period was fixed at 6 year. The low salvage period of the still box had an impact on the price of water produced. Despite of this, the cost of water produced is less than 10% of the market price prevailing in the region. The current market price of bottled water is Rs.20, while the distilled water produced in this study is only Rs 1.08.

CONCLUSION

An extended type stepped solar still was tested with different initial depths of water and various feed water inputs to identify the influence of sensible heat storage capacity

Table 6. Cost Analysis Parameters [50,51] [There is no table legends comprise of referrals to other articles. So the written permission is not required in this case.]

1	Input Parameters considered		
1.1	Asset Cost	P	Rs. 8837.5
1.2	Salvage Cost (30 % of tray material cost)	S	281.25
1.3	Period of operation	n	6 years
1.4	Bank Interest Rate	i	8%
1.5	Production Days Annually	d	300 days
1.6	Mean Annual Distillate Productivity	M	900 litres
2	Parameters Calculated		
2.1	Capital Recovery Factor	CRF	0.22
2.2	Sinking Fund Factor	SFF	0.14
2.3	Initial investment for first year	IIF	1944.25
2.4	Annual Maintenance Cost	AMC	Rs. 291.64
2.5	First Annual Salvage Value	ASV	Rs. 1237.25
2.6	Total Annual Cost	TAC	Rs. 998.64
2.7	Cost of Water per litre	CPL	Rs. 1.11/ Litre

Table 7. Economic analysis of different designs of stills

Sl. No.	Still Type	Energy Storage Material	Total Fixed Cost (\$)	Salvage Period (years)	Days of productivity per year	Yield (ml/m ² /day)	Cost of water per liter (\$/L)
1	Stepped Still [Present work]	Water	106.05	6	300	6330	0.013
2	Stepped Still [6]	Black Sand	187	20	340	5750	0.016
3	Inclined Stepped Solar Still [40]	Nil	1200	10	340	4383	0.047
4	Tray Solar Still [45]	Black Sand	157	20	340	5650	0.015
5	Tray Solar Still [23]	Nil	154	10	340	4800	0.021
6	Multi-tray evaporator still [22]	Nil	85.47	10	340	5820	0.01
7	Stepped Double Slope [44]	Linen Wicks	89	10	270	3260	0.0221
8	Stepped Still with Fins [49]	Pebbles	160	-	-	1370	0.32
9	Pyramid Solar Still [43]	Palmately leaf -Natural Fibers	865	10	-	5160	0.081
10	Conventional Still [46]	Fine Basalt stones	275	10	340	1075	0.017
11	Conventional still with V-shaped floating wicks [41]	Black Jute Cloth	129.41	10	250	6200	0.022
12	Conventional Still with preheating [39]	Coarse aggregate	147.12	10	-	4210	0.0618

of water in the trays. These following conclusions were derived from the experiments conducted:

1. In the stepped stills, with no feed water addition an increase in the initial depth of water results in the decrease of distillate output. As the initial depth of water increased from 0.5cm to 1.0cm and 1.0cm to 1.5cm, the productivity decreased by 3% and 7% respectively.
2. The addition of feed water led to a significant improvement in daily distillate production. Notably, adding 400 ml of feed water to an initial water depth of 0.5 cm resulted in a 19.7% increase in yield.
3. Increasing the quantity of feed water added resulted in a nearly twofold increase in nocturnal yield.
4. The highest distillate yield of 6330 ml/m²/day, was produced for 0.5 cm initial depth of water and 400 ml feed water addition per tray for the stepped solar still.
5. The daily efficiency of the still improved with the addition of feed water, reaching a maximum efficiency of 66.45%.

NOMENCLATURE

A_t	Total area of the tray, m ²
B	Average number of observations
C_{pb}	Specific heat of basin in solar still (J/kg °C)
C_{pg}	Specific heat of glass in solar still (J/kg °C)
C_{pw}	Specific heat of water in solar still (J/kg °C)
dw	Initial depth of water, cm
dw_{x-y}	x gives the initial depth of water (cm) and y the quantity of feed water added per tray, ml

h_{cnv}	Convective heat transfer coefficient between water and glass, W/m ² °C
h_{evp}	Evaporative heat transfer coefficient, W/m ² °C
h_{tbw}	Total heat transfer coefficient between basin and water, W/m ² °C
h_{tga}	Total heat transfer coefficient between glass and ambient, W/m ² °C
h_{twg}	Total heat transfer coefficient between water and glass, W/m ² °C
$I(t)$	Solar radiation intensity, W/m ²
$I_b(t)$	Fraction of solar radiation intensity absorbed by basin, W/m ²
$I_g(t)$	Fraction of solar radiation intensity absorbed by glass, W/m ²
$I_w(t)$	Fraction of solar radiation intensity absorbed by water, W/m ²
IIF	Initial investment for first year
L_{ev}	Latent Heat of Vaporization of water, kJ/Kg
m_b	mass of trays (kg)
m_g	mass of glass (kg)
m_s	mass of stored water in the trays (kg)
m_w	Hourly yield, kg
M_w	Cumulative yield, kg
n	Total number of observations
n_o	Number of observations in one set
P_g	Partial saturated vapor pressures at glass cover temperature (N/m ²)
P_w	Partial saturated vapor pressure at a basin water temperature (N/m ²)
t	Time interval, seconds
T_a	Ambient Temperature, °C

T_g	Temperature of glass cover, °C
T_{sky}	Ambient Sky Temperature, °C
T_w	Temperature of Water in the trays, °C
U_{ba}	Total heat transfer coefficient between basin and ambient, W/m ² °C
V_w	Wind velocity, m/sec

Greek symbols

η	Hourly thermal efficiency of the still
ϵ_{eff}	Effective emissivity between water surface and glass cover
ϵ_g	Emissivity of glass cover

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Sadigov R. Rapid growth of the world population and its socioeconomic results. *Sci World J* 2022;2022:8110229. [\[CrossRef\]](#)
- [2] Cruz-Pérez N, Santamarta JC, Gamallo-Paz I, Rodríguez-Martín J, García-Gil A. A comparison between carbon footprint of water production facilities in the Canary Islands: groundwater resources vs seawater desalination. *Sustain Water Resour Manag* 2022;8:1–9. [\[CrossRef\]](#)
- [3] Heihsel M, Lenzen M, Malik A, Geschke A. The carbon footprint of desalination: An input-output analysis of seawater reverse osmosis desalination in Australia for 2005–2015. *Desalination* 2019;454:71–81. [\[CrossRef\]](#)
- [4] Nassrullah H, Anis SF, Hashaikeh R, Hilal N. Energy for desalination: A state-of-the-art review. *Desalination* 2020;491:114569. [\[CrossRef\]](#)
- [5] Hussien HM, Younes MM, Alawee WH, Abdullah AS, Mohammed SA, Atteya TEM, et al. An experimental comparison study between four different designs of solar stills. *Case Stud Therm Eng* 2023;44:102841. [\[CrossRef\]](#)
- [6] Atteya TEM, Abbas F. Testing a stepped solar still with different sand beds and reflectors. *Case Stud Therm Eng* 2023;43:102782. [\[CrossRef\]](#)
- [7] Jathar LD, Ganesan S. Assessing the performance of concave type stepped solar still with brick, sand, and concrete pieces. *Int J Ambient Energy* 2020;43:1–36. [\[CrossRef\]](#)
- [8] Chauhan MK, Chauhan AK, Khan Y, Singh AP. Experimental and theoretical investigation of thermal efficiency and productivity of single slope basin type solar distillation system using honey-comb. *J Therm Eng* 2023;9:1559–1571. [\[CrossRef\]](#)
- [9] Dhivagar R, Shoeibi S, Parsa SM, Hoseinzadeh S, Kargarsharifabad H, Khiadani M. Performance evaluation of solar still using energy storage bio-material with porous surface: An experimental study and environmental analysis. *Renew Energy* 2023;206:879–889. [\[CrossRef\]](#)
- [10] Ramalingam VK, Karthick A, Jeyalekshmi MPV, Decruz AMMAJ, Manokar AM, Sathyamurthy R. Enhancing the fresh water produced from inclined cover stepped absorber solar still using wick and energy storage materials. *Environ Sci Pollut Res* 2021;28:18146–18162. [\[CrossRef\]](#)
- [11] Abdullah AS. Improving the performance of stepped solar still. *Desalination* 2013;319:60–65. [\[CrossRef\]](#)
- [12] Ali C, Rabhi K, Nciri R, Nasri F, Attyaoui S. Theoretical and experimental analysis of pin fins absorber solar still. *Desalin Water Treat* 2015;56:1705–1711. [\[CrossRef\]](#)
- [13] Muftah AF, Sopian K, Alghoul MA. Performance of basin type stepped solar still enhanced with superior design concepts. *Desalination* 2018;435:198–209. [\[CrossRef\]](#)
- [14] Lilley D, Yu P, Ma J, Jain A, Prasher R. Thermal fluids with high specific heat capacity through reversible Diels–Alder reactions. *IScience* 2022;25:103540. [\[CrossRef\]](#)
- [15] Arunkumar T, Wang J, Dsilva Winfred Rufuss D, Denkenberger D, Kabeel AE. Sensible desalting: Investigation of sensible thermal storage materials in solar stills. *J Energy Storage* 2020;32:1–27. [\[CrossRef\]](#)
- [16] Siddula S, Stalin N, Mahesha CR, Dattu VSNCH, S H, Singh DP, et al. Triangular and single slope solar stills: Performance and yield studies with different water mass. *Energy Reports* 2022;8:480–488. [\[CrossRef\]](#)

- [17] Kabeel AE, El-Agouz SA, Sathyamurthy R, Arunkumar T. Augmenting the productivity of solar still using jute cloth knitted with sand heat energy storage. *Desalination* 2018;443:122–129. [\[CrossRef\]](#)
- [18] Hammoodi KA, Dhahad HA, Alawee WH, Omara ZM. A detailed review of the factors impacting pyramid type solar still performance. *Alexandria Eng J* 2023;66:123–154. [\[CrossRef\]](#)
- [19] Hoque A, Abir AH, Paul Shourov K. Solar still for saline water desalination for low-income coastal areas. *Appl Water Sci* 2019;9:104. [\[CrossRef\]](#)
- [20] Zanganah P, Soltani A, Ayatollahi S, Feilizadeh M. Efficiency improvement of solar stills through wettability alteration of the condensation surface: An experimental study. *Appl Energy* 2020;268:1–11. [\[CrossRef\]](#)
- [21] D'Cotha JS, Sajeesh P, Suresh PR. Influence of initial water depth of extended stepped tray on the productivity while working beyond sunshine hours. *Mater Today Proc* 2023;217:267–275. [\[CrossRef\]](#)
- [22] Saadi Z, Rahmani A, Lachtar S, Soualmi H. Performance evaluation of a new stepped solar still under the desert climatic conditions. *Energy Convers Manag* 2018;171:1749–1760. [\[CrossRef\]](#)
- [23] Abdullah AS, Younes MM, Omara ZM, Essa FA. New design of trays solar still with enhanced evaporation methods – Comprehensive study. *Sol Energy* 2020;203:164–174. [\[CrossRef\]](#)
- [24] Abujazar MSS, Fatihah S, Lotfy ER, Kabeel AE, Sharil S. Performance evaluation of inclined copper-stepped solar still in a wet tropical climate. *Desalination* 2018;425:94–103. [\[CrossRef\]](#)
- [25] Kabeel AE, Khalil A, Omara ZM, Younes MM. Theoretical and experimental parametric study of modified stepped solar still. *Desalination* 2012;289:12–20. [\[CrossRef\]](#)
- [26] Omara ZM, Kabeel AE, Younes MM. Enhancing the stepped solar still performance using internal reflectors. *Desalination* 2013;314:67–72. [\[CrossRef\]](#)
- [27] D'Cotha JS, Sajeesh P, Suresh PR, Jithu J. Inherent configuration characteristics altering the distillate enhancement of passive stepped solar still: A review. *J King Saud Univ Eng Sci* 2021;8:684–693. [\[CrossRef\]](#)
- [28] Thakur VK, Gaur MK, Sagar MK, Tiwari GN. A study on heat and mass transfer analysis of solar distillation system. *J Therm Eng* 2021;7:1184–1205. [\[CrossRef\]](#)
- [29] Rani A, Kant R, Suresh S, Kumar A. Experimental investigation on thermal behavior of hybrid single slope solar still. *J Therm Eng* 2021;7:677–689. [\[CrossRef\]](#)
- [30] Abdullah AS, Alawee WH, Shanmugan S, Omara ZM. Techniques used to maintain minimum water depth of solar stills for water desalination – A comparative review. *Results Eng* 2023;19:101301. [\[CrossRef\]](#)
- [31] Alwan NT, Ali BM, Alomar OR, Abdulrazzaq NM, Ali OM, Abed RM. Performance of solar still units and enhancement techniques: A review investigation. *Heliyon* 2024;10:e37693. [\[CrossRef\]](#)
- [32] Nougriaya SK, Chopra MK, Gupta B, Baredar P, Parmar H. Influence of basin water depth and energy storage materials on productivity of solar still: A review. *Mater Today Proc* 2021;44:1589–1603. [\[CrossRef\]](#)
- [33] Agrawal A, Rana RS, Srivastava PK. Heat transfer coefficients and productivity of a single slope single basin solar still in Indian climatic condition: Experimental and theoretical comparison. *Resour Technol* 2017;3:466–482. [\[CrossRef\]](#)
- [34] Gawande JS, Bhuyar LB. Effect of shape of the absorber surface on the performance of stepped type solar still. *Energy Power Eng* 2013;5:489–497. [\[CrossRef\]](#)
- [35] Omara ZM, Abdullah AS, Kabeel AE, Essa FA. The cooling techniques of the solar stills' glass covers – A review. *Renew Sustain Energy Rev* 2017;78:176–193. [\[CrossRef\]](#)
- [36] Dunkle R. Solar water distillation; the roof type still and a multiple effect diffusion still. *Dev Heat Transf ASME Proc Heat Transf Part V* 1961;895.
- [37] Jamil B, Akhtar N. Effect of gap between absorber plate and condenser cover on the performance of a solar still. *Green Energy Technol* 2016;PartF2:161–173. [\[CrossRef\]](#)
- [38] Jamil B, Akhtar N. Effect of specific height on the performance of a single slope solar still: An experimental study. *Desalination* 2017;414:73–88. [\[CrossRef\]](#)
- [39] Dhivagar R, Mohanraj M, Hidouri K, Belyayev Y. Energy, exergy, economic and enviro-economic (4E) analysis of gravel coarse aggregate sensible heat storage-assisted single-slope solar still. *J Therm Anal Calorim* 2021;145:475–494. [\[CrossRef\]](#)
- [40] Abujazar MSS, Fatihah S, Kabeel AE. Seawater desalination using inclined stepped solar still with copper trays in a wet tropical climate. *Desalination* 2017;423:141–148. [\[CrossRef\]](#)
- [41] Agrawal A, Rana RS. Theoretical and experimental performance evaluation of single-slope single-basin solar still with multiple V-shaped floating wicks. *Heliyon* 2019;5:e01525. [\[CrossRef\]](#)
- [42] El-Sebaai AA. Effect of wind speed on active and passive solar stills. *Energy Convers Manag* 2004;45:1187–1204. [\[CrossRef\]](#)
- [43] Alshqirate AA, Awad AS, Al Alawin A, Essa MA. Experimental investigation of solar still productivity enhancement of distilled water by using natural fibers. *Desalination* 2023;553:116487. [\[CrossRef\]](#)
- [44] Sharshir SW, Eltawil MA, Algazzar AM, Sathyamurthy R, Kandeal AW. Performance enhancement of stepped double slope solar still by using nanoparticles and linen wicks: Energy, exergy and economic analysis. *Appl Therm Eng* 2020;174:115278. [\[CrossRef\]](#)

-
- [45] Hadj-Taieb L, Abdullah AS, Aljaghtham M, Alkhudhiri A, Omara ZM, Essa FA. Improving the performance of trays solar still by using sand beds and reflectors. *Alexandria Eng J* 2023;71:659–668. [\[CrossRef\]](#)
- [46] Mohamed AF, Hegazi AA, Sultan GI, El-Said EMS. Enhancement of a solar still performance by inclusion of basalt stones as a porous sensible absorber: Experimental study and thermo-economic analysis. *Sol Energy Mater Sol Cells* 2019;200:109958. [\[CrossRef\]](#)
- [47] Bilal A, Jamil B, Haque NU, Ansari MA. Investigating the effect of pumice stones sensible heat storage on the performance of a solar still. *Groundw Sustain Dev* 2019;9:100228. [\[CrossRef\]](#)
- [48] El-Bialy E, Shalaby SM, Kabeel AE, Fathy AM. Cost analysis for several solar desalination systems. *Desalination* 2016;384:12–30. [\[CrossRef\]](#)
- [49] Velmurugan V, Naveen Kumar KJ, Noorul Haq T, Srithar K. Performance analysis in stepped solar still for effluent desalination. *Energy* 2009;34:1179–1186. [\[CrossRef\]](#)
- [50] Panchal HN. Life cycle cost analysis of a double-effect solar still. *Int J Ambient Energy* 2017;38:395–399. [\[CrossRef\]](#)
- [51] Rahbar N, Gharaiian A, Rashidi S. Exergy and economic analysis for a double slope solar still equipped by thermoelectric heating modules – An experimental investigation. *Desalination* 2017;420:106–113. [\[CrossRef\]](#)
- [52] Furbo S. Using water for heat storage in thermal energy storage (TES) systems. Cambridge, UK: Woodhead Publ Ltd; 2015. [\[CrossRef\]](#)



Research Article

Experimental adsorption studies of atmospheric water vapor by clay based composite sorbents in vertical fluidized bed

C. R. HIREMATH^{1,*}, Ravikiran KADOLI², R. J. TALAPATI³

¹Department of Mechanical Engineering, BLDEA's V. P. Dr. P. G. Halakatti College of Engineering and Technology, Vijayapur-Karnataka, 586103, India

²Department of Mechanical Engineering, National Institute of Technology Karnataka, Surathkal, 575025, India

³Department of Mechanical Engineering, KLS Vishwanathrao Deshpande Institute of Technology, Haliyal, 581329, India

ARTICLE INFO

Article history

Received: 21 August 2024

Revised: 14 December 2024

Accepted: 18 December 2024

Keywords:

Adsorption; Clay; Calcium chloride; Fluidization; Horse dung; Humidity ratio; Sawdust

ABSTRACT

Transient adsorption characteristics of atmospheric water vapor with clay and clay-additives-based calcium chloride composite desiccants in vertical fluidized beds are studied experimentally. The different clay-based composite sorbents considered in the study are clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride adsorbents. The influence of bed mass and air superficial velocity on transient change in bed inlet air humidity ratio and air temperature for different desiccant clay-based composite sorbents is experimentally studied. It was noted that the thermo-physical properties of different adsorbents, bed mass and air inlet velocity significantly influence the fluidized bed performance in the adsorption system. The linear porosity distribution in fluidization shows the availability of increased adsorption capacity for clay-based desiccants in comparison to clay-additive desiccants, as indicated by higher total adsorption capacity. For the similar conditions of bed weight (300 g) and inlet air velocity (2 m/s), the total quantity of water adsorbed is 30.09 g, 21.84 g and 27.02 g for clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride fluidized adsorbent beds. The results reveal heat load reduction of clay-calcium chloride vertical fluidized bed dehumidification system is 62% and 52% higher compared to clay-horse dung-calcium chloride and clay-sawdust-calcium chloride desiccant dehumidification systems. For all the dehumidification systems, the reduction in latent load is higher than the increase in sensible heat load, which will reduce the overall heat load of the cooling system.

Cite this article as: Hiremath CR, Kadoli R, Talapati RJ. Experimental adsorption studies of atmospheric water vapor by clay based composite sorbents in vertical fluidized bed. J Ther Eng 2025;11(5):1374–1391.

*Corresponding author.

*E-mail address: chandra.hiremath@yahoo.com

This paper was recommended for publication in revised form

Editor-in-Chief Ahmet Selim Dalkilic.



INTRODUCTION

The demand for economical, environmentally friendly, energy-efficient air conditioning and refrigeration systems is increasing daily. Using refrigerants in traditional air conditioning and refrigeration systems, such as hydro chlorofluorocarbons and chlorofluorocarbons, endangers the ozone layer and contributes to global warming. Because of this, sorption-based air conditioning systems are becoming more and more critical. In desiccant cooling, the air is dehumidified and then sensibly cooled before being supplied to the conditioned space, a relatively newer air conditioning technique. The capability of the adsorbent material to adsorb water vapor is a critical factor in dehumidification performance. The desiccant-based system is used in applications such as heating, ventilation, air conditioning and refrigeration (HVACR) system, energy and environment, drying of grains and cereals, food preservation, freshwater production, filtration, purification processes, dehumidification, pollution control, wastewater treatment and thermal energy storage system [1-7]. The parameters influencing the desiccant-based system's effectiveness are desiccant bed configuration and desiccant material properties. The conventional packed bed system is commonly employed in sorption-based systems, and it has been noted that flow through the packed bed is not entirely uniform. At any instant in time, the adsorbents above or below the mass transfer zone do not participate in the mass transfer process. Lv bo et al. [8] investigated liquid-solid fluidized bed's fluidization properties and adsorption efficiency using batch, cyclic and continuous adsorption characteristics. The fluidized bed layer's external diffusion rate increased due to high-frequency contact between adsorbent and adsorbate. The results showed that a fluidized bed is more effective than a packed bed in adsorption. Krzywanski et al. [9] investigated the fluidized bed-to-surface heat transfer and fixed beds of adsorption chillers. Compared to a conventional packed bed, a fluidized bed of adsorbents showed a considerable increase in bed-to-wall heat transfer coefficient. Further, it was revealed that the proposed concept of a fluidized bed will reduce the chiller's size, eliminating the fundamental barrier to broader adoption of adsorption cooling technology.

Hamed [10] experimentally investigated the desorption and adsorption behavior of an inclined fluidized bed with silica gel as a desiccant. It was observed that the adsorption rate was highly affected by inlet air velocity. Horibe et al. [11] experimentally investigated sodium polyacrylate sorbent's fluidized bed sorption behavior with multiple cooling pipes. It was revealed that numerous cooling pipes resulted in a more excellent sorption ratio and a shorter sorption process completion time. Ye et al. [12] developed soybean seeds' contact sorption-fluidized bed drying system from the perspective of energy efficiency and product quality. The drying rate was improved with gas velocity and mass ratio of silica gel. Experimental results showed an improvement

in the quality of dried seeds since drying is carried out in a consistent environment of low humidity. Sobrino et al. [13] experimentally studied the effect of rotational distributor plates on vertical cylindrical bed behavior. The proposed bed design promotes the radial distribution of particles, reducing high-concentration zones and preventing temperature gradients like hot spots in a fluidized bed. Hamed et al. [14] studied the transient adsorption/desorption behavior of spherical particles of silica gel in a fluidized bed. The maximum decrease in air humidity was reported at the beginning of adsorption. Wang et al. [15] conducted an experimental study on the performance of fluidized bed adsorber/desorber containing a working pair of adsorption refrigeration systems with activated carbon. The experiments were performed for different masses of activated carbon, and enhancements were observed for 2 Kg and 3 Kg of activated carbon adsorbent. However, the effectiveness of 4 Kg adsorbent in fluidized adsorber and desorber was low due to incomplete fluidization of activated carbon. Horibe et al. [16] experimentally investigated sorption desorption characteristics of organic sorbent HU720PR in two-chamber connected fluidized beds. It was reported that dehumidification performance was highly influenced by the initial bed height of the sorbent and the temperature difference between sorption and desorption chambers. Zettl et al. [17] developed a rotating drum filled with zeolite to avoid the non-reactive zones. Due to the moving reaction bed, the whole volume will contribute to adsorption dehumidification in distinction to fixed bed vessels. Li et al. [18] proposed a novel compact micro fluidized bed (CMFB) to remove carbon dioxide from indoor air using resin-based polyethyleneimine solid adsorbent in HVAC. CMFB's saturation adsorption time was 17% shorter than a fluidized bed (FB). A fluidized bed provides a means for achieving intimate gas-sorbent contacting and mixing. The results showed that a fluidized bed is more effective than a packed bed in the adsorption process. Fluidization might be very useful in increasing the adsorption-desorption characteristics of sorbents in a desiccant column.

Studies on chemical desiccants and their composites have been studied in packed and fluidized bed configurations. Still, only few studies have been published on biomass-based and environmentally friendly natural desiccants [19]. Impregnation of sodium chloride deliquescent salt within a super-porous hydrogel polymer matrix improved desiccant performance threefold [20]. The driving force behind this improved performance is the porous structure of the polymer matrix. The potential use of natural materials and their composites as desiccants for water vapor extraction and evaporative cooling was investigated.

Kumar et al. [21] investigated saw wood-calcium chloride composite desiccant on water production from atmospheric air. Freshwater production from ambient air depends on the amount of calcium chloride. Hiremath et al. [22] investigated clay's transient heat and mass transfer behavior as the desiccant carrier with sawdust and horse

dung as additives. Liang et al. [23] reported the effect of geometrical factors on an improved circulating inclined fluidized bed for an HVAC duct system. The optimal ICIFB arrangement resulted in minimum pressure drop and high energy factor compared to circulating erect fluidized bed (CEFB). Thus, an optimal ICIFB system was beneficial for lowering energy consumption. Timsina et al. [24] reported the influence of different particle sizes on fluidization characteristics. Compared to the experiment with only large particles, the lowest fluidization velocity with 20% tiny and 80% large particles was reduced by 60.8%.

Singh et al. [25] investigated moisture desorption and sorption behavior of cocopeat, sawdust and dried cow dung in the evaporative cooling system. Experimental results showed that dried cow dung and coco-peat had the potential to replace chemical desiccants. Dorothea and Dimitrios [26] explored the application of spruce sawdust processed with sulfuric acid and diethylene glycol to remove hexavalent chromium from wastewater. The rate of adsorption capacity was found to increase by 84% for sawdust pretreated with 50% diethylene glycol and by 89% for sawdust pretreated with sulphuric acid. Dasar et al. [27] investigated dried cow dung's sorption and desorption behavior with Polyvinyl Pyrrolidone and clay as a binder. It was reported that dried cow dung with Polyvinyl Pyrrolidone could replace chemical desiccants due to its exergy efficiency and moisture absorption capability. One of the benefits of a fluidized bed system over a packed bed system is the high heat and mass transfer rate among gas and particles, as reported in the literature study.

Previous studies were published using commercially available silica gel in fluidized bed configurations and their sorption performances. A few studies on clay and clay-based sorbents in fluidized beds were discussed. Besides desiccant and bed configurations, literature on using calcium chloride as the desiccant and biomaterial is minimal [28]. To fill this research gap, the current study explores the effect of clay, clay-based sorbents in dehumidifying the atmospheric air using vertical fluidized bed arrangement.

Most of the earlier research was done on chemical desiccants and their mixtures. Although there are few studies on using natural desiccants in the open literature, dried cow dung has a very promising moisture uptake capacity; however, during the desorption process, the binding strength is lost, resulting in a mass reduction of the desiccant. Using clay and clay-additive desiccants in vertically packed beds releases heat of adsorption, increasing the bed exit air temperature and bringing an additional heat load [29]. The increase in outlet air again becomes a source of energy consumption that requirements to be resolved. For this reason, sorbent-based fluidized beds with high heat and mass transfer rates are promising alternatives suitable for air-conditioning systems. The fluidized bed system has a lot of benefits regarding the energy savings of an air-conditioning system with effective dehumidification performance diminishing the scale of the system, low extra-sensible heat

load reducing the system load [30]. The research aims to study the adsorption characteristics of atmospheric water vapor with desiccants based on clay in a vertical fluidized bed (VFB). Naturally available clay and hygroscopic inorganic salts are employed to prepare sorbents. Low-cost biomaterials like horse dung and sawdust are utilized as additives. The experimental study was conducted to understand the influence of bed mass and air inlet velocity on transient change in bed air humidity ratio and air temperature for different desiccant clay-based composite sorbents. The different clay-based composite sorbents considered in the study are clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride adsorbents. Furthermore, heat load calculations and exergy analyses are performed to measure the capabilities of these novel desiccant composites.

DESICCANT MATERIALS AND EXPERIMENTAL SYSTEM

Desiccant Materials

In the present study, composite desiccants made of clay are employed. The desiccants are prepared from naturally available clay, biomaterials like horse dung and sawdust as additives and commercially available hygroscopic calcium chloride inorganic compound. The clay is procured from a local pot maker, which prepares baked pots and oil lamp fixtures. Horse dung, popularly employed by local pot makers to prepare porous pots, is selected as an additive. Sawdust, a sawmill residue, is another additive with horse dung. The cleaned clay and segregated additives are combined in the proportion of 1:20 by mass. The mixture of clay and additives is thoroughly wetted by water, and separate pasty material of clay, clay-horse dung and clay-sawdust are obtained. The ready pasty material is then extruded through a syringe. The syringe has a needle of 2 mm diameter. The extrusion process produces an extended cylindrical-shaped structure of 2 mm diameter. The overlong cylinder is then cut to a length of 2 mm along its length. Thus, cylindrical-shaped desiccants of size 2 mm × 2 mm are prepared. clay lumps and desiccants based on clay additions are heat treated to 500°C in a muffle furnace. The heat treatment removes the initial desiccant moisture and induces strength and rigidity. The photographs of prepared desiccants before saturation with calcium chloride hygroscopic solution are shown in Figure 1.

A calcium chloride solution with a concentration of 50% is prepared by dissolving granules of calcium chloride in distilled water. To avoid catching atmospheric moisture, the solution is kept in an airtight jar. The calcium chloride desiccant solution is prepared by impregnating lumps of heat-treated clay and clay-additive desiccants. The duration of impregnation is 24 hours. Following that, soaked particles are filtered and later dried in the furnace. Further, to ensure the impregnation and depth of penetration of calcium chloride into the pores of desiccants, scanning electron microscopy

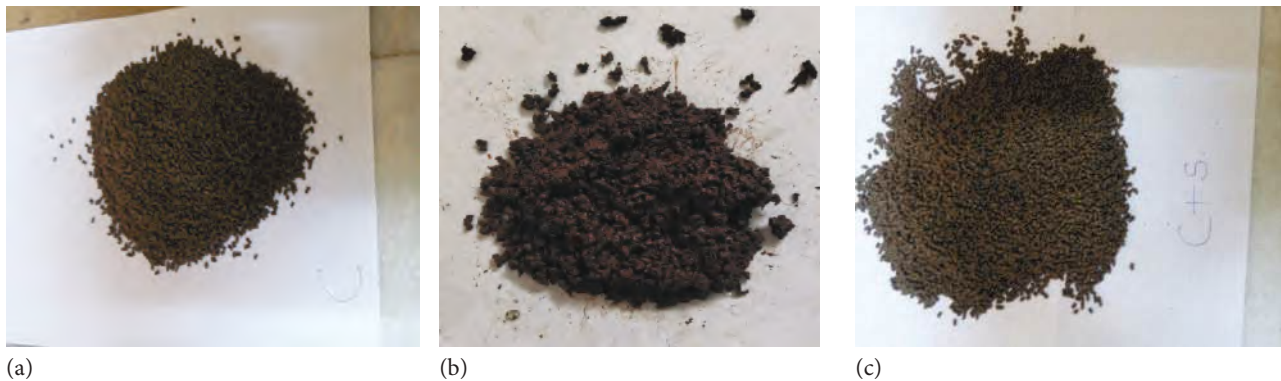


Figure 1. Photographs of heat treated (a) clay (b) clay-horse dung and (c) clay-sawdust desiccants.

(SEM) image analysis is carried out. The images of the halved portion of clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride at 2000-X magnification are shown in Figure 2.

The surface examination of images reveals a porous nature with interconnected voids and valleys. Further, it shows a smoother surface (see Figure 2(a) and 2(b)) appearance for clay additives composites as a rough surface for clay without additives (see Figure 2(c)). The white flaky and spongy appearance qualitatively ensures the depth and penetration of calcium chloride onto the surface. The surface pattern of clay-calcium chloride shows the existence of inter-particle macro-pores, whereas clay-horse

dung-calcium chloride and clay-sawdust-calcium chloride indicate dispersed micro-pores

The thermo-physical properties of prepared desiccants are shown in Table 1 [22, 31]. The higher porosity and water content decrease the thermal conductivity of clay-horse dung-calcium chloride and clay-sawdust-calcium chloride by 16.75 % and 29.56% pertaining to the clay-calcium chloride desiccant. Maximum thermal diffusivity values are being reported for clay-based composite desiccants. Lower porosity in clay-calcium chloride desiccants signifies higher density than clay-additives-based desiccants. The higher water content in clay-additives-based desiccants contributes to higher specific heat capacities.

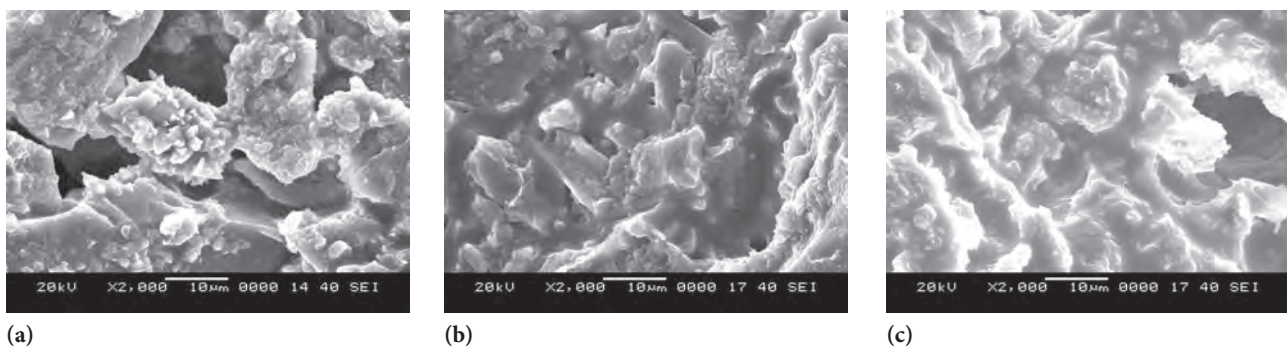


Figure 2. Scanning electron microscope images of clay composite desiccants showing cut-sections (a) clay-calcium chloride (b) clay-horse dung- calcium chloride (c) clay-sawdust-calcium chloride.

Table 1. Properties of different solid composite adsorbent

Adsorbent	Thermal diffusivity (m ² /s)	Specific heat (J/kg K)	Thermal conductivity (W/m ² K)	Density (Kg/m ³)	Porosity (%)	Initial water content (g water/kg adsorbent)
Clay- calcium chloride	6.54×10 ⁻⁵	832.5	2.03	2372	13	63.79
Clay-horse dung- calcium chloride	4.54×10 ⁻⁵	1392	1.69	1456	26	94.44
Clay-sawdust-calcium chloride	3.30×10 ⁻⁵	1461	1.43	1114	23	86.03

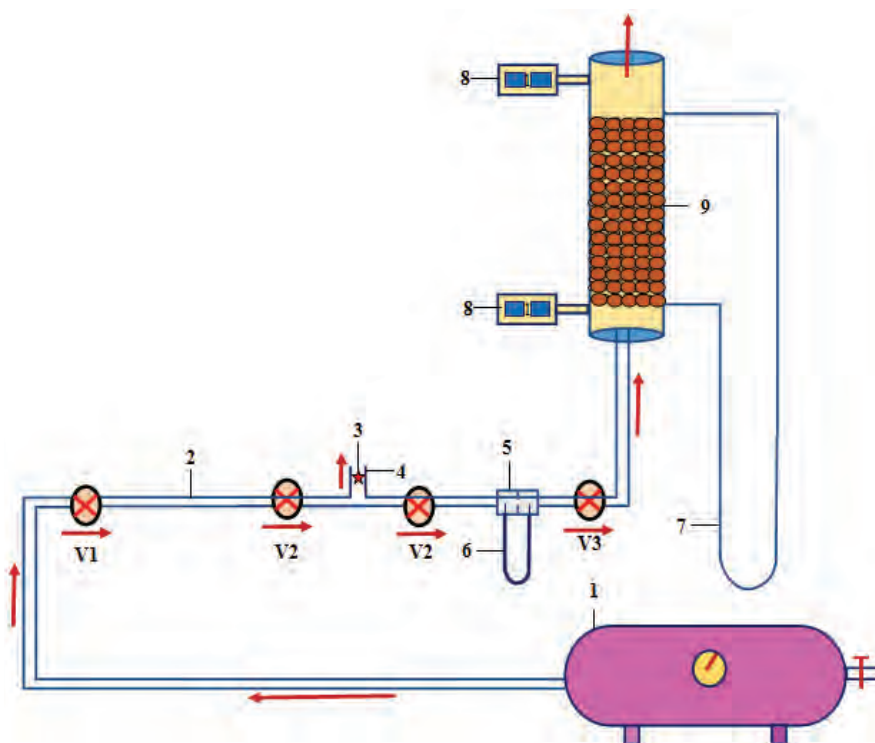
Experimental Methodology

The present experimental setup is the same as reported in Hiremath et al. [22] and is used to conduct dehumidification experiments. The components of the experimental arrangement are an interconnected air compressor, air heating unit and vertical desiccant column, shown in Figure 3. The system is a single-blow, open-loop adsorption experiment. Air from the compressor is directed through a galvanized iron pipe of 25 mm diameter to the vertical bed column. Flow control valves (V1, V2) regulate the process airflow direction for adsorption experiments. To visualize the fluidized bed behavior, the vertical cylindrical desiccant test section is constructed with transparent acrylic material. The schematic diagram and photograph image of various phases of fluidization are shown in Figure 4 and Figure 5. The size of a single cylindrical adsorbent and regeneration bed is 700 mm in height, 50 mm in diameter and 4 mm thick. A flange connects the vertical bed and the experiment's primary pressure line with the nut and bolts system. For uniform distribution of air stream, a fine wire mesh of stainless steel is fixed at the lower part of a bed. The bed's bottom and flange are properly sealed to prevent the leakage of desiccants and air stream. The vertical bed height (700 mm) is sufficient to prevent the loss of desiccants from the upper part of the bed, as the maximum fluidization height attained during experimentation is 650 mm. The

vertical column has taps for placing the hygro-transmitter at the inlet and outflow of the vertical bed column and pressure taps. During the initial stages of fluidization, the air moves through the bed, gets loosened, and dense phase fluidization occurs. The clay composite desiccant moves away from each other, creating voids for the passage of air. This indicates the desiccant bed begins to experience smooth fluidization (see Figure 5(a)). The voids created due to the separation of the desiccant provide passage for airflow because air flows at fluidization velocity, and the desiccant is further taken into a state of buoyancy. This increased the bed height, and the fluidization type switched to bubbling fluidization (see Figure 5(b)).

Once the desiccants start to bubble, the momentum of air is such that the desiccant is carried along with the air stream, achieving complete fluidization. The bed behavior changes from bubbling fluidization to a turbulent fluidization regime (see Figure 5(c)). All desiccants are well circulated, the bed reaches the maximum height and all desiccants are entrained into the vertical bed.

The minimum superficial velocity (v) to overcome gravitational energy required for fluidization should be known and is estimated from the Leva equation [32]. The minimum value of fluidization velocity estimated using Leva equation for clay-calcium chloride and clay - additives - calcium chloride desiccants is checked and validated during experiments.



1. Air compressor 2. Pressure line 3. Thermocouple 4. Pressure line vent 5. Orifice meter 6, 7. U-Tube manometer 8. Hygrometer 9. Fluidized bed. 10. Control valves (V1, V2, V3, V4).

Figure 3. Experimental set up layout.

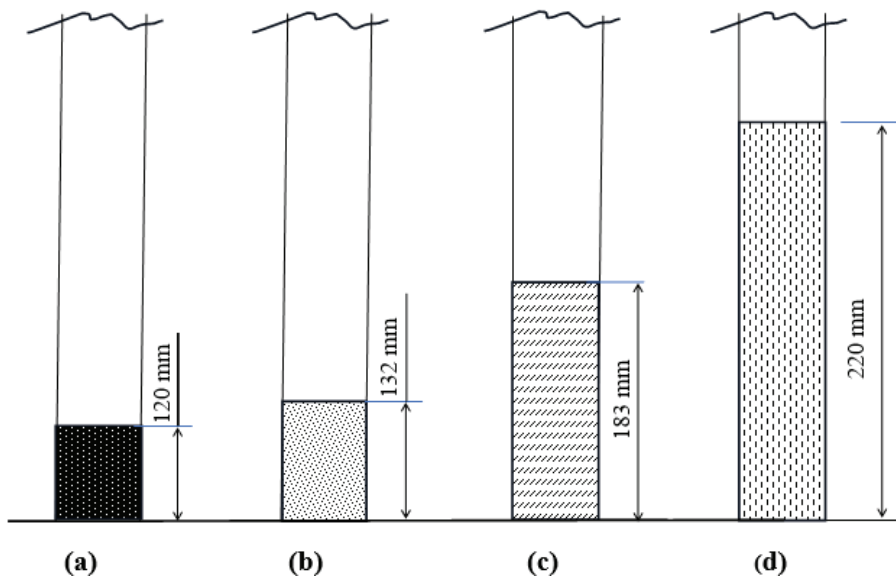


Figure 4. Schematic diagram of various phases of fluidization of clay - horse dung - calcium chloride for $m=200$ g, $v=1.5$ m/s. (a) Packed bed (b) Dense phase fluidization (c) Bubbling fluidization (d) Turbulent fluidization.

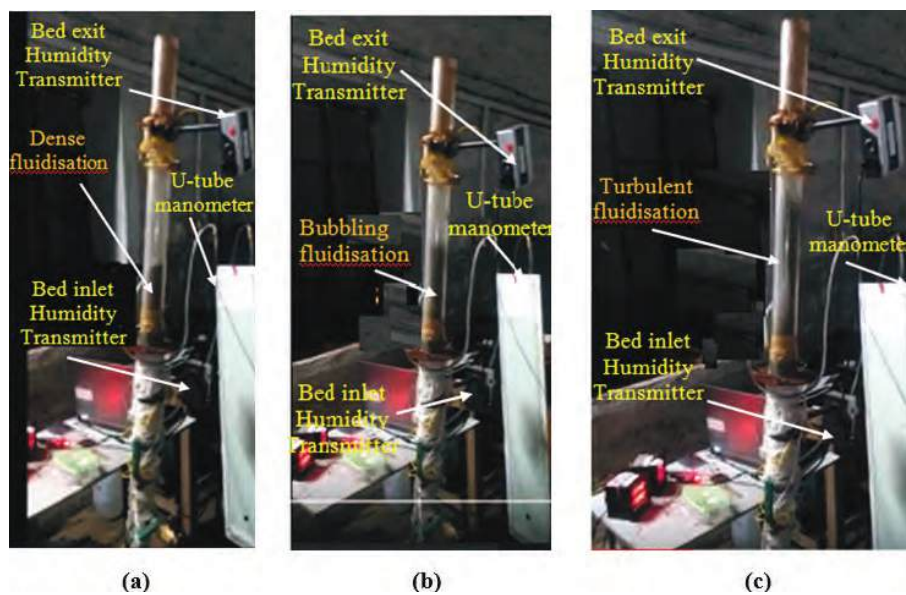


Figure 5. Photograph showing regimes of fluidization (a) Dense fluidization (b) Bubbling fluidization and (c) Turbulent fluidization.

Table 2. Validation of minimum fluidization velocity

Desiccant bed	Bed weight (g)	Minimum fluidization velocity (Experimental, m/s)	Minimum fluidization velocity (Theoretical, m/s)
Clay-calcium chloride	200	1.26	1.87
Clay - sawdust-calcium chloride	200	1.70	1.20
Clay - horse dung-calcium chloride	200	1.74	1.18

Before the fluidization experiments the minimum fluidization velocity for bed weight 200 g was measured and is shown in Table 2. The experimentally tested minimum fluidization velocity for bed mass of 200 g bed weight is about 1.26 m/s to 1.74 m/s. The theoretically estimated velocity for clay-calcium chloride and clay-additives-calcium chloride desiccant ranges from 1.20 m/s to 1.87 m/s.

Data reduction

The actual volume flow rate of air through orifice flow meter is given by following equation: [33];

$$Q_{act} = \frac{C_d A_1 A_2 (2gh)^{0.5}}{\left((A_1)^2 - (A_2)^2\right)^{0.5}} \quad (1)$$

The density of process air is obtained from ideal gas relation by using equation (2) [33];

$$\rho_a = \frac{P_a}{R_a T_a} \quad (2)$$

Equation (3) calculates the mass flow rate of air [33];

$$\dot{m} = \rho_a \times Q_{act} \quad (3)$$

The process air superficial velocity at bed inlet is estimated by equation (4) [33];

$$v_a = \frac{\dot{m}}{\left(\rho_a \times \frac{\pi}{4} d_b^2\right)} \quad (4)$$

The vapor pressure (P_w) at desiccant bed entry and exit as a function of relative humidity (ϕ) and saturation pressure (P_{sat}) is given by equation (6). The saturation pressure at bed inlet and exit as a function of temperature of air is given by equation (5) [34];

$$P_{sat} = 610.78 \times e^{\frac{(17.2694 \times T_a)}{(T_a + 273)}} \quad (5)$$

$$P_w = \phi \times P_{sat} \quad (6)$$

The humidity ratio (H_R) of process air at bed inlet and exit is given by equation (7) [34];

$$H_R = \frac{0.622 \times P_w}{P_{total} - 0.378 \times P_w} \quad (7)$$

The linear variation of bed porosity (ε_z) along the bed length in fluidization is given by equation (8) [34];

$$\varepsilon_{b_z} = \left(\varepsilon_{\min}\right) + \left(\frac{z}{L}\right) \left(\varepsilon_{\max} - \varepsilon_{\min}\right) \quad (8)$$

The total adsorption rate ($M_{sat}(t)$) is the amount of water adsorbed during a specific period and is obtained by taking integral of transient adsorption rate with process time as given by equation (9) [35];

$$M_{ads}(t) = \dot{m} \int_0^t (H_{R_i}(t) - H_{R_e}(t)) \Delta t \quad (9)$$

Adsorption heat released during moisture uptake increases the temperature of air. This heat is undesirable in evaporative cooling but can be indirectly cooled to condition the room. The total heat load includes both sensible and latent heat load. The decrease in sensible heat load for unit mass flow rate due to decrease in temperature of the air can be estimated by equation (10) [27];

$$Q_S = C_{p_a} (T_i - T_e) \quad (10)$$

The decrease in latent heat load because of a reduction in moisture content from the air. Latent heat load for unit mass flow rate is given by equation (11) [27];

$$Q_L = h_{fg_v} (H_{R_i} - H_{R_e}) \quad (11)$$

The sensible heat factor (SHF) is given by equation (12) [27];

$$SHF = \frac{Q_S}{(Q_S + Q_L)} \quad (12)$$

Exergy of process air comprises thermal exergy and chemical exergy. The change in thermal exergy is due to heat and mass transfer between desiccant and the process air. The change in chemical exergy is due to variation in mass of Calcium chloride contained in the pores of desiccant. The exergy of process air at the desiccant bed inlet is given by equation (13) [36];

$$E_{X_i} = \left(C_{p_a}\right) + \left(H_{R_i} C_{p_v}\right) T_o \left[\frac{T_i}{T_o} - 1 - \ln \frac{T_i}{T_o}\right] + R_o T_o \left[1 + 1.608 H_{R_i} \ln \frac{(1+1.608 H_{R_o})}{(1+1.608 H_{R_i})} + 1.608 H_{R_i} \ln \frac{H_{R_i}}{H_{R_o}}\right] \quad (13)$$

The exergy of process air at the desiccant bed outlet is given by equation (14) [36];

$$E_{X_e} = \left(C_{p_a}\right) + \left(H_{R_e} C_{p_v}\right) T_o \left[\frac{T_e}{T_o} - 1 - \ln \frac{T_e}{T_o}\right] + R_o T_o \left[1 + 1.608 H_{R_e} \ln \frac{(1+1.608 H_{R_o})}{(1+1.608 H_{R_e})} + 1.608 H_{R_e} \ln \frac{H_{R_e}}{H_{R_o}}\right] \quad (14)$$

The exergy destruction (E_Y) in water vapor adsorption is given by equation (15) [36];

$$E_Y = E_{X_i} - E_{X_e} \quad (15)$$

The exergy efficiency of desiccant dehumidifier is given by equation (16) [36];

$$\eta = \left[\frac{E_{X_i} - E_Y}{E_{X_i}} \right] \quad (16)$$

$$\pm \sqrt{\left(\frac{\partial v}{\partial x} \right)^2 \times (\omega x)^2} \quad (18)$$

Measurements and Instrument Uncertainty

The experimental setup shown in Figure 3 is equipped with the following instrumentation to measure the parameters in adsorption experiments:

- Two Hygroflex hydro transmitters are fixed at the inlet and outlet of the vertical bed column to measure unsteady values of relative humidity and air temperature with an accuracy of $\pm 1\%$ RH for 0 % to 100 % relative humidity and $\pm 0.2^\circ\text{C}$ for temperatures range of -40°C to 85°C .
- Pressure drops across the orifice meter, and the vertical fluidized bed is obtained using two water-filled U-Tube manometers. The accuracy of the manometers is 1 mm.
- The volume flow rate of air is estimated with an orifice flow meter of 10.12 mm diameter, and the coefficient of discharge (C_d) is 0.68.
- The mass is measured using a digital balance with a range of 0. of 2 g to 300 g with a resolution of 0.01 g.
- The air temperature in the pressure line is measured using a calibrated K-type thermocouple (Chromel - Alumel) having a temperature range of -5°C to 1260°C .

The uncertainty value for the dependent parameter humidity ratio (H_R) is uncertainty in the measurement and calculation of saturation pressure (P_{sat}), vapor pressure (P_w) and relative humidity (ϕ). The superficial velocity (v_a) is a function of manometer deflection (x). Therefore, uncertainty analysis is carried out to evaluate the error or deviation in calculated parameters by the error propagation [37, 38]. Substituting measured and calculated data in equations (17) and (18) calculates the uncertainty in the experimental parameters like humidity ratio and superficial velocity.

$$\pm \sqrt{\left(\frac{\partial H_R}{\partial P_w} \right)^2 \times \left(\left(\left(\frac{\partial P_w}{\partial \phi} \right)^2 \times (\omega \phi)^2 \right) + \left(\left(\frac{\partial P_w}{\partial P_{sat}} \right)^2 \times (\omega P_{sat})^2 \right) \right)} \quad (17)$$

Table 3. Uncertainties in the measurement of experimental parameters

Parameter	Uncertainty (%)	
	m = 200 g	m = 300 g
Saturation pressure (P_{sat})	± 0.64	± 0.92
Vapor pressure (P_v)	± 0.78	± 1.36
Humidity ratio (H_R)	± 0.78	± 1.39
Superficial velocity (v_a)	$\pm 0.24\%$	$\pm 0.40\%$
Mass of bed (m)	$\pm 0.005\%$	$\pm 0.003\%$
Temperature of air (T_a)	± 1	± 1

The parameters of the bed's weight (m) and the air's temperature (T_a) are measured directly with calibrated devices. The maximum values of uncertainty of the physical parameters, which cannot be measured directly, and the parameters that are measured directly in the experimental runs are presented in Table 3.

RESULTS AND DISCUSSION

Adsorption Performance of Clay and Clay Based Calcium Chloride Composite Adsorbents in a Vertical Fluidized Bed

The transient variation of fluidized bed air humidity ratio, air temperature at inlet and exit conditions for different air velocities, and bed mass are shown in Figures 6 - 7. For every adsorption experimental result, the humidity ratio trends and exit air temperature are similar. Maximum reductions in humidity and temperature of bed exit air relative to inlet air temperature are seen during the first phase of fluidization. The decrease in humidity results from higher initial dryness of the bed. Apart from dryness, lower vapor pressure on the desiccant surface and the higher interfacial area between the desiccant surface and surrounding air lowers the humidity of air leaving the bed [39]. The adsorption rate is higher in the early stages of the dehumidification process and gradually decreases over time. When the process starts first, the vapor pressure on the bed's surface is lower than the vapor pressure in the air. The wall portions in pores provide a mass transfer area. The distances over which moisture travels are decreased by the increase in mass transfer area between process air and desiccants owing to fluidization. The pores are filled with water vapor with time, decreasing the effective interfacial area available for mass transfer. Filling the pores with vapor decreases the calcium chloride concentration, increasing the vapor pressure on the desiccant's surface and directly affecting the potential mass transfer from air to bed. The change in calcium chloride concentration affects the potential mass transfer from the start of the experiment to the end state [39]. The transient change of the bed's exit air humidity and air temperature of 200 g and 300 g clay-calcium chloride desiccants at the bed's air inlet velocity for 1.5 m/s, 2 m/s is shown in Figure 6(a) - 6(b) and Figure 7(a)-7(b). The maximum reduction in humidity ratio for 200 g clay-calcium chloride bed mass exposed to the bed's air inlet velocity of 1.5 m/s and humidity of 5.65 g/kg is 44.04% (see Figure 6(a)). In contrast to the bed's entrance air temperature of 28.86°C , the average temperature of the bed's exit air is 27.24°C (see Figure 7(a)). With a bed mass of 300 g and an air velocity of 2 m/s, the maximum reduction in humidity is around 43.74% (see Figure 6(b)), and exit air attains a maximum temperature of 27.84°C relative to an inlet air temperature

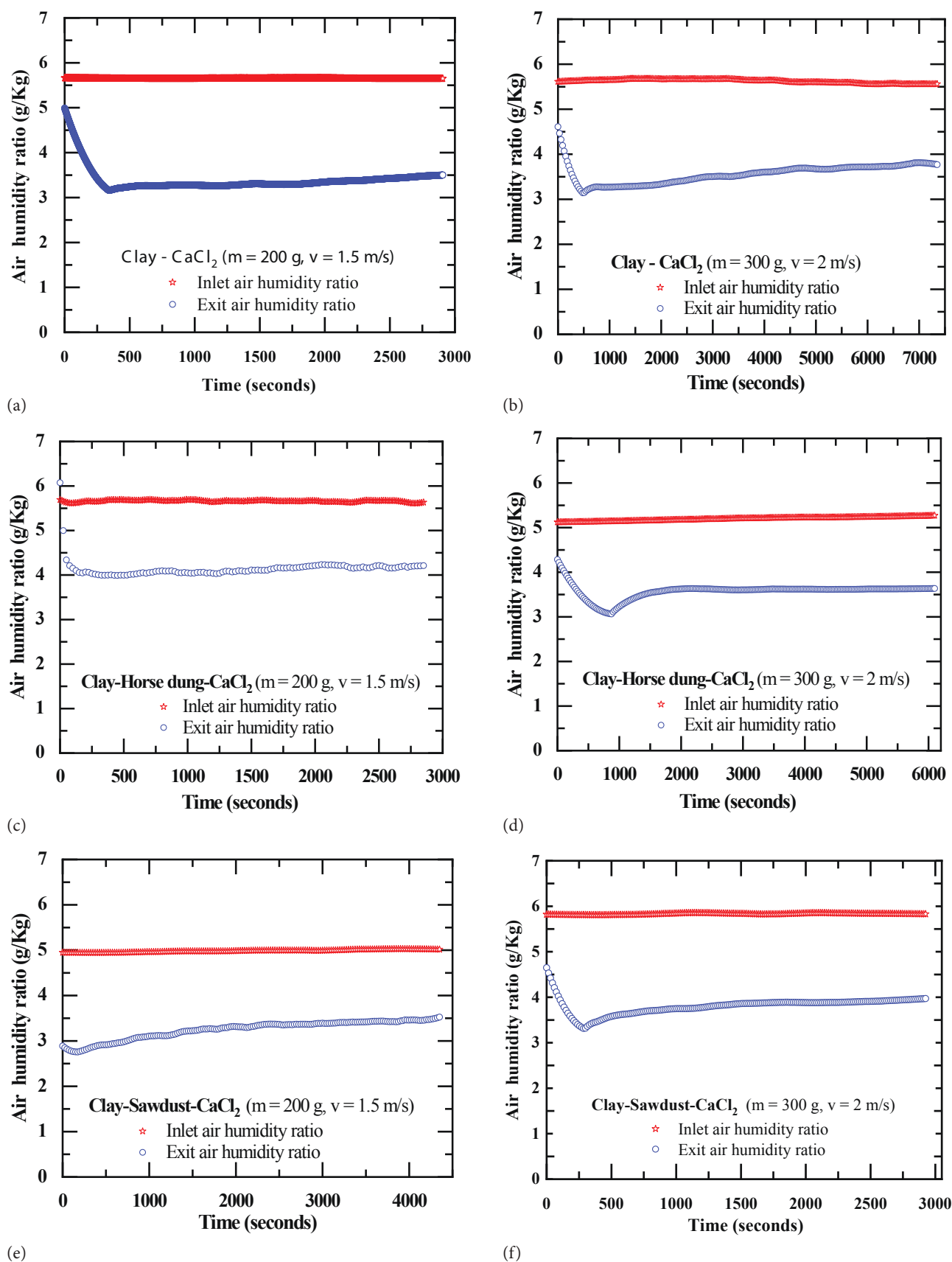


Figure 6. Air humidity ratio characteristics of different desiccants for adsorption process.

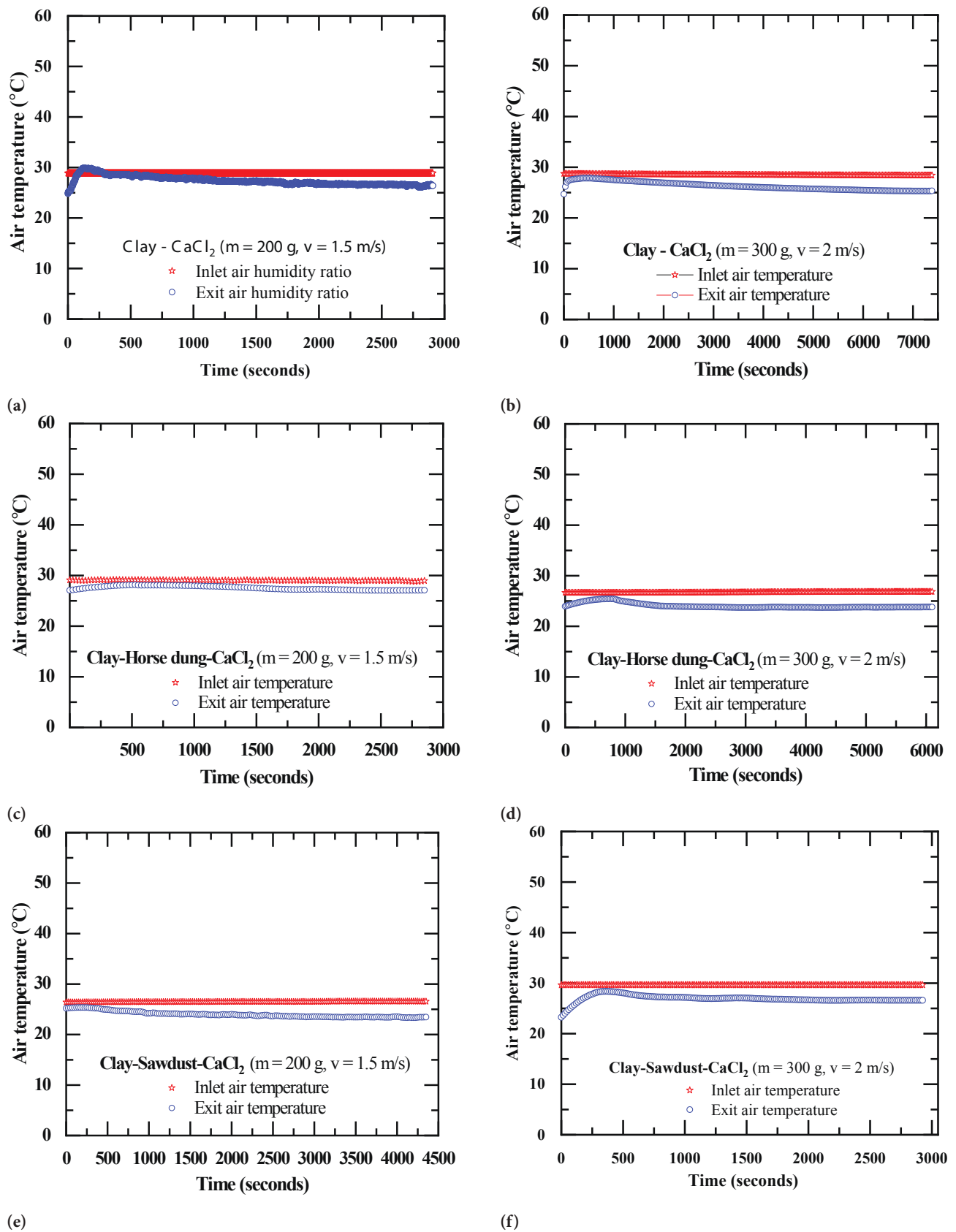


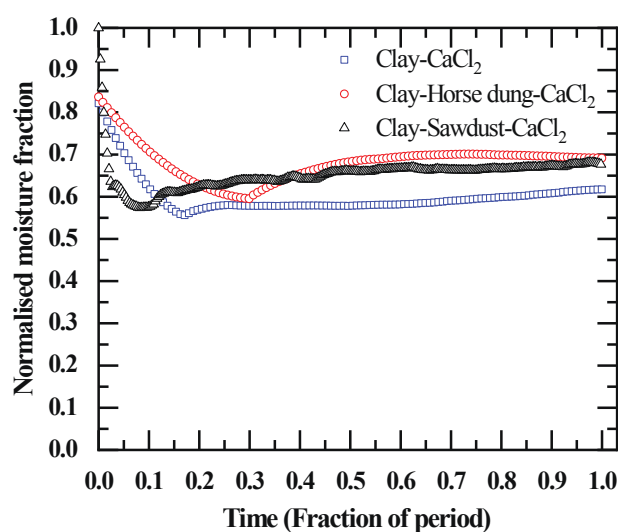
Figure 7. Air temperature characteristics of different desiccants for adsorption process.

of 28.74°C (see Figure 7(b)). For 0.0045 kg/s mass flow rate of process air, pressure drop varies from 0.34 to 0.29 m of water column. Once the process had finished, the bed mass increased by 5.84% and 3.4%, respectively, with dry desiccant bed masses of 200 g and 300 g. The transient change of the bed's exit air humidity ratio and air temperature of clay-horse dung-calcium chloride for different bed masses and air velocities is shown in Figure 6(c) - 6(d) and Figure 7(c) - 7(d). During the initial period of the process, the maximum decrease in humidity ratio for 200 g bed mass is 29.55% (see Figure 6(c)). In contrast, for 300 g, the reduction in humidity ratio is 41.89% (see Figure 6(d)). The maximum increase in temperature for 200 g bed mass is 3.42% (see Figure 7(c)), whereas for 300 g, the increase is 5.64% (see Figure 7(d)). The maximum decrease in humidity and increase in temperature is due to a higher surface area of 0.93 m² for 300g, whereas the available surface for 200 g bed mass is 0.62 m². The increased air velocity and bed surface accelerate the adsorption rate. With time, the adsorption rate decreases and steadily progresses for 200 g and 300 g bed masses. At the end, bed mass is increased by 4% and 2.93% for 200 g and 300 g bed masses at 1.5 m/s and 2 m/s inlet air velocities of bed, respectively. The transient change of the bed's exit air humidity and air temperature for 200 g and 300 g clay-sawdust-calcium chloride desiccants at the bed's inlet air velocity of 1.5 m/s, 2 m/s is shown in Figure 6(e) - 6(f) and Figure 7(e) - 7(f). The humidity ratio declines to 55% of bed inlet air humidity during the early stages of the process for 200 g bed mass. It rises to a maximum of 70% of bed inlet air humidity. The bed's exit air temperature drops to a minimum of 88% of the bed's inlet air temperature and rises to a maximum of 25% of the bed's inlet air temperature. For 300 g bed mass, the humidity ratio attains

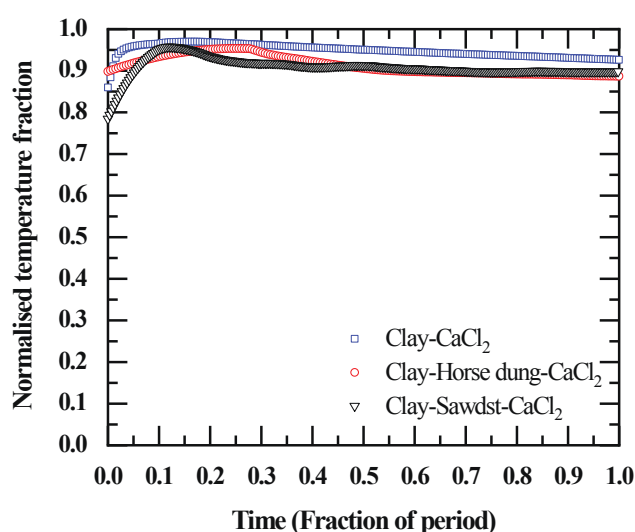
a maximum drop of 57% of bed inlet air humidity. When the process is finished, the humidity reaches 67% of inlet air humidity. The bed exit air temperature exceeds 95% of the temperature of bed intake air and reaches a minimum of 89% of air inlet temperature. The bed mass increases by 4.45% for 200 g and 2.22% for 300 g bed masses during the adsorption process, respectively.

The results in Figures 6 and 7 indicate a maximum drop in exit air humidity ratio and a maximum rise in exit air temperature within 500 s of the process. Within this period, the adsorption intensity is higher for all the clay and clay-additives-based desiccant beds. Though there is an increase in the temperature of bed exit air, it drops below the inlet air temperature. The drop in bed exit air temperature is less and remains steady compared to bed inlet air temperature. The results thus reveal the isothermal adsorption behavior of clay and clay-additives-based desiccants in fluidization.

In the adsorption process within the fluidized bed, we observed that the exit air humidity ratio is lower than the inlet air humidity ratio. This is expected as the sorbent material adsorbs moisture from the air, reducing the moisture content in the exit air. Regarding the temperature behavior, although adsorption is an exothermic process that theoretically releases heat known as the heat of adsorption, the observed decrease in the exit air temperature can be attributed to the unique dynamics of the fluidized bed. In fluidized bed systems, the constant mixing of particles and air promotes efficient heat transfer and allows the sorbent to dissipate the released heat effectively. Thus, hot spots are effectively dissipated, resulting in isothermal adsorption conditions. Additionally, the airflow in the bed has a cooling effect, which counterbalances the heat released during adsorption, resulting in a lower exit air temperature.



(a)



(b)

Figure 8. Normalized (a) humidity ratio and (b) temperature fraction of process air for different fluidized beds at $m = 300$ g, $v = 2$ m/s.

Thus, the observed reduction in exit humidity and exit temperature can be explained by the combined effects of adsorption and the cooling influence of the airflow within the fluidized bed, leading to effective moisture removal with moderate temperature control. Apart from hot spot dissipation and cooling effect, lower residence time for air in bed and higher heat capacities [see Table 1] of clay and clay-additives-based composite desiccants control the increase in bed exit air temperature concerning inlet air temperature in fluidization.

Performance comparison of clay and clay-based calcium chloride adsorbents in vertical fluidized beds for dehumidification

The comparison of clay-additives-calcium chloride composite desiccants water vapor and process air temperature is shown in Figures 8 (a) and (b). It is reported that the normalized humidity ratio for clay-calcium chloride is 0.56, clay-sawdust-calcium chloride is 0.57 and clay-horse dung-calcium chloride fluidized desiccant beds is 0.60 (see Figure 8(a)). Clay-calcium chloride desiccants with the lowest value of humidity ratio exhibit a higher rate of adsorption than clay-additives-calcium chloride desiccants. The higher adsorptivity is due to higher dryness due to lower initial bed water content. Clay-based desiccants have a higher density than additives-based ones, resulting in higher residence and contact time for process air and higher heat and mass transfer rates. Composite desiccants made with clay additions have less adsorptivity due to the shorter contact duration. The higher porosity of clay-additives-based adsorbents reduces the thermal conductivity and diffusivity of adsorbents, reducing the heat transfer rate in the clay-additives-based fluidized beds [40, 41]. Even though clay-based desiccants have low porosity, they have a higher adsorption rate because of their more considerable pore volume and higher interfacial area of interconnected macro-pores, as observed in Figure 2. With time, the adsorptivity of the clay-calcium chloride adsorbent bed is higher than the adsorptivity of the clay-additives-based composite desiccant bed [see Figure 8(a)]. Adsorption heat released with increased adsorption rate increases the bed outlet air temperature. The maximum value of normalized temperature is 0.97 for clay-based desiccants compared to clay-additives-based composite desiccants (see Figure 8 (b)). For all fluidized beds, the point of maximum temperature coincides with the point of the lower humidity value. The maximum temperature attained by the outlet air for clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride beds are 27.84°C, 25.42°C and 28.37°C respectively. The lower exit temperature of the adsorbed air stream may be attributed to using inert materials like clay, horse dung and sawdust in desiccant preparation [22, 42]. Applying additives of horse dung and sawdust with natural clay increases heat capacity and simultaneously decreases the thermal conductivity and thermal diffusivity of clay-additive desiccants (see Table 1).

Compared to clay desiccants, the thermo-physical properties of additives-based composite desiccants lessen the dissipation of adsorption heat to fluidized air. For clay-additive desiccants, the bed water uptake capacity immediately falls from the point of higher absorptivity forward, whereas, for clay beds, the absorptivity progressively declines. Once the process has finished, the bed water amount is increased by 3.4% for clay-calcium chloride, 2.93% for clay-horse dung-calcium chloride and 2.22% for clay-sawdust-calcium chloride beds.

The lower adsorptive nature of low-density clay-additives-based calcium chloride desiccants is due to reduced residence time for air in fluidization. Low-density adsorbents may experience uneven fluidization, which causes channeling and bypassing, where air flows through preferential paths, further reducing the effective active interfacial mass transfer area with the desiccants and thus limiting the adsorption of water vapor.

In the case of clay-horse dung-calcium chloride and clay-sawdust-calcium chloride composites, the adsorbed water vapor may form a layer on the surface of the desiccants in less time as compared to heavier clay-calcium chloride desiccants. The water vapor layer over the desiccant particles minimizes the available surface area for water vapor adsorption. Since adsorption is a surface-driven process, any reduction in the effective surface area significantly lowers the desiccant's performance.

The linear variation of porosity along the fluidized bed height is shown in Figure 9(a). The minimum and maximum bed lengths in fluidization with bed mass of 300 g and velocity of air at 2 m/s are 150 and 250 mm, 210 and 450 mm, 250 and 650 mm for clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride desiccant beds. The estimated minimum and maximum porosity distribution values are 0.56 and 0.72, 0.50 and 0.76, 0.45 and 0.78 for clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride desiccant beds. For identical bed masses of 300 g, additive-based desiccant has lower density and higher bed heights than clay-based fluidized desiccant beds. The axial porosity distribution reveals more adsorption sites in clay-additives-based beds than higher active adsorption sites in clay-based desiccant beds. The higher intensity of fluidization and a greater number of active adsorption sites (see Figure 2(a)) results in higher absorptivity as indicated by higher total adsorption capacity for clay-calcium chloride fluidized bed. Apart from fluidization, the higher clay content and inter-particle macro-pores [see Figure 2] in clay-calcium chloride desiccants result in higher water uptake and increased adsorption rate [43]. As shown in Figure 9(b), the calculated total adsorption quantities for identical 300 g bed mass are 30.09 g, 21.84 g and 27.02 g for clay-calcium chloride, clay-horse dung-calcium chloride and clay-sawdust-calcium chloride fluidized adsorbent beds. Due to more pores with a higher quantity of calcium

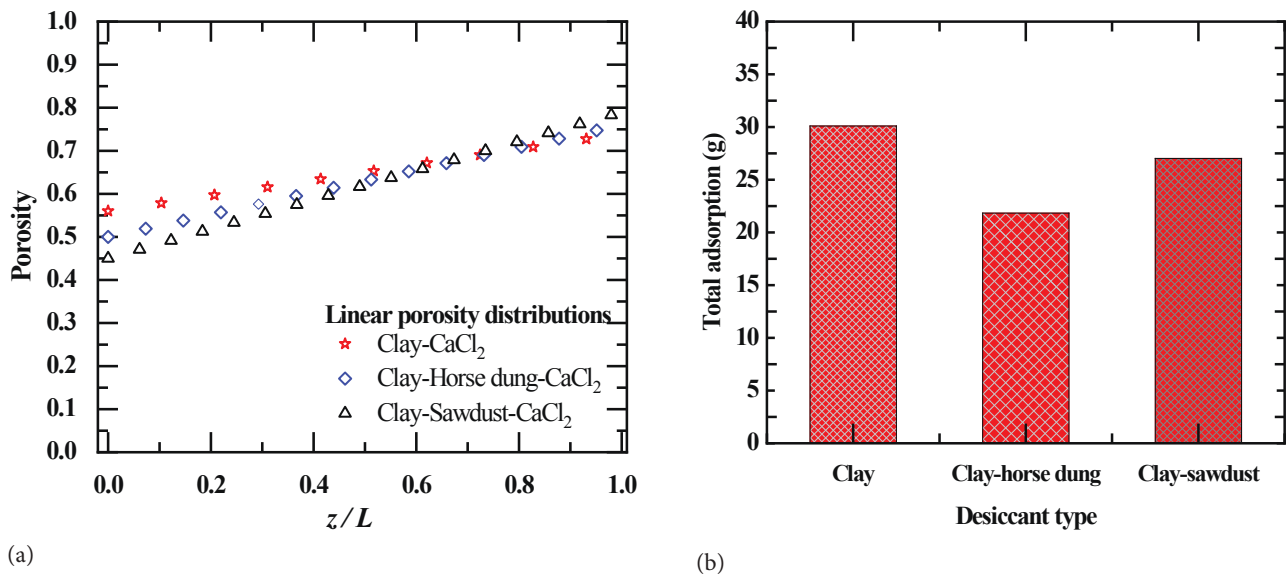


Figure 9. (a) Porosity variation along the bed length and (b) Total adsorption rate for identical mass of fluidized beds at $m = 300$ g, $v = 2$ m/s.

chloride impregnated, the clay-calcium chloride fluidized bed has a higher overall adsorption capacity.

Sensible heat factor and Exergy efficiency

The sensible and latent heat loads are calculated for the weights of desiccant beds, which are calculated for 200 g and 300 g, and the values are tabulated in Table 4. For all the cases, the results reveal that the average latent heat load decrease is higher than the sensible heat load increase. Hence, the overall heat load of the cooling system will decrease. It is observed that total heat load reduction is higher for clay-calcium chloride than for clay-additives-calcium chloride composite desiccants. Higher heat load reduction is due to higher moisture uptake, as Figure 9(b) indicates. Due to lower moisture uptake in clay-additive-calcium chloride beds, the heat load reduction is lesser, as shown in Table 4.

The sensible heat factor, which is a function of total heat load, is considered when designing the air conditioning process. Figure 10 shows the sensible heat factor for clay

and clay-additives-based calcium chloride composite desiccant for different bed masses and bed inlet air velocities. For the identical bed masses of 200 g, the decrease in sensible heat and latent heat loads are 21% and 79% for clay-calcium chloride bed, 29% and 71% for clay-horse dung bed and 34% and 66% for clay-saw dust-calcium chloride beds, respectively. With the increase in bed mass and air velocity, the sensible heat factors increase due to the availability of more bed surface area for dehumidification of process air. The result shows that the decrease in latent heat load is higher for clay-calcium chloride beds as compared to clay-additives-calcium chloride beds, whereas a reduction of sensible heat load is higher for clay-additives-calcium chloride beds. The higher decrease in sensible heat load in additive-based desiccants is attributed to the higher heat capacity of composite desiccants. The higher heat capacity, initial water content and lower thermal conductivity (see Table 1) of additive-based desiccants decrease the sensible heat load. The results indicate an average sensible heat

Table 4. Total heat load reduction by clay and clay-additives based composite desiccants

Type of desiccant	Weight of bed (m), g	Sensible heat (Q_s), J/kg	Latent heat (Q_L), J/kg	Net decrease in total heat load ($Q_L - Q_s$), J/kg
Clay-calcium chloride	200	1631.92	5074.75	3442.83
	300	1912.19	5510.35	3598.16
Clay-horse dung-calcium chloride	200	858.32	1866.23	1007.91
	300	2563.67	3943.62	1379.95
Clay-sawdust-calcium chloride	200	2435.02	4170.79	1735.77
	300	2527.94	4271.75	1743.81

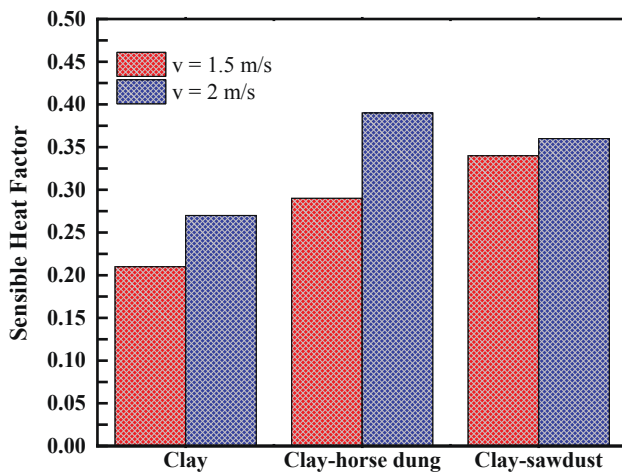


Figure 10. Sensible heat factor for different fluidized beds in dehumidification.

factor of 0.30 for clay and clay-additives-based dehumidification systems. The portion of the latent heat load handled by these dehumidification systems is 70 % compared to a sensible heat load of 30% [Table 4]. The results show the potential of clay and clay-additives-based desiccants in providing thermal comfort in hot and humid climates for sorption-based air conditioning systems.

The variation of exergy efficiency with time for different bed masses and air velocity are shown in Figure 11. The difference in exergy efficiency during dehumidification process by the fluidized clay and clay-additives based calcium chloride composite desiccants are seen in Figure.11. The estimated average rate of total exergy at bed inlet and exit are 369.4-191 kJ/s, 327-163 kJ/s and 638.3 - 264 kJ/s

for 200 g clay-calcium chloride, clay-horse dung calcium chloride and clay-sawdust-calcium chloride desiccant beds. While for 300 g bed mass, the average exergy values at desiccant bed inlet and exit are 273.49 -107.04kJ/s, 612.23-213.38 kJ/s and 433.36-120.52kJ/s respectively. From Figures 11 (a) and (b) it is clear that, the rate of exergy efficiency was higher in the initial period of dehumidification process. It reaches its peak value between 500 s to 1000 s of process time and after that the exergy efficiency decreases and reaches its minimum value. The peak value is reached up to the point of release of heat of adsorption. The released heat due to adsorption of moisture increases the exergy losses which lowers the exergy efficiency. From Figure.11 (b) it is observed that point of peak value of efficiency for clay-calcium chloride bed leads the peak point for clay-horse dung-calcium chloride bed followed by clay-saw dust-calcium chloride bed. The delayed attainment of peak value for clay based composite desiccant is attributed due to micro-pore volume, higher density, lower porosity and higher dryness (See Table 1).

The attainment of higher exergy efficiency in clay-based desiccants is due to lower pressure drop through the bed. The lower pressure drop minimizes the work input to the system. The pressure drop through the bed of clay-calcium chloride is 372.78 Pa, whereas for clay-horse dung-calcium chloride, it is 539.55 Pa, and for clay-walled dust-calcium chloride, the bed is 1226.25 Pa. The lower exergy efficiency values for clay-additives-based composite desiccant beds reveal the slower adsorptive nature compared to clay-based composite desiccant beds. The higher latent heat load controlled by clay-calcium chloride composite desiccants increases the exergy efficiency of process air exiting the bed. The results reveal that a higher decrease in latent heat increases exergy efficiency. In contrast, a higher reduction

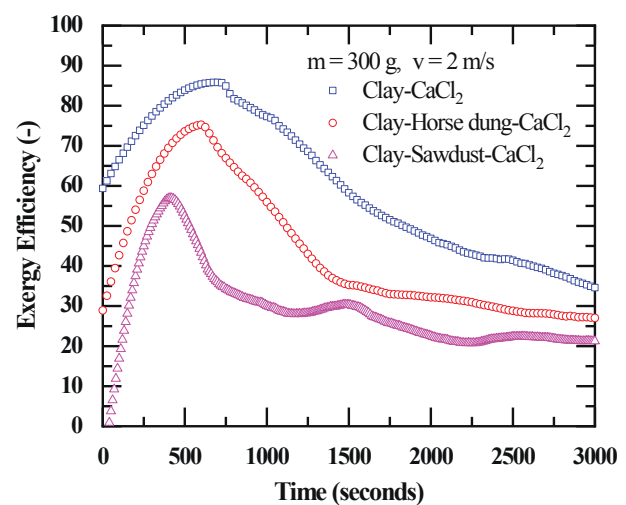
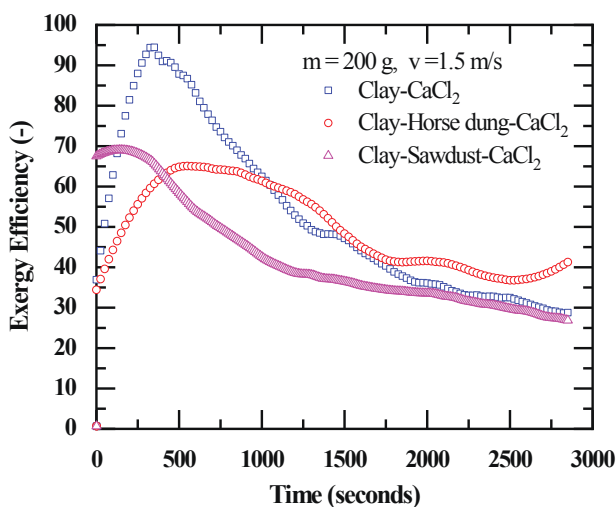


Figure 11. Transient variation of exergy efficiency for fluidized clay and clay-additives composite desiccants in dehumidification.

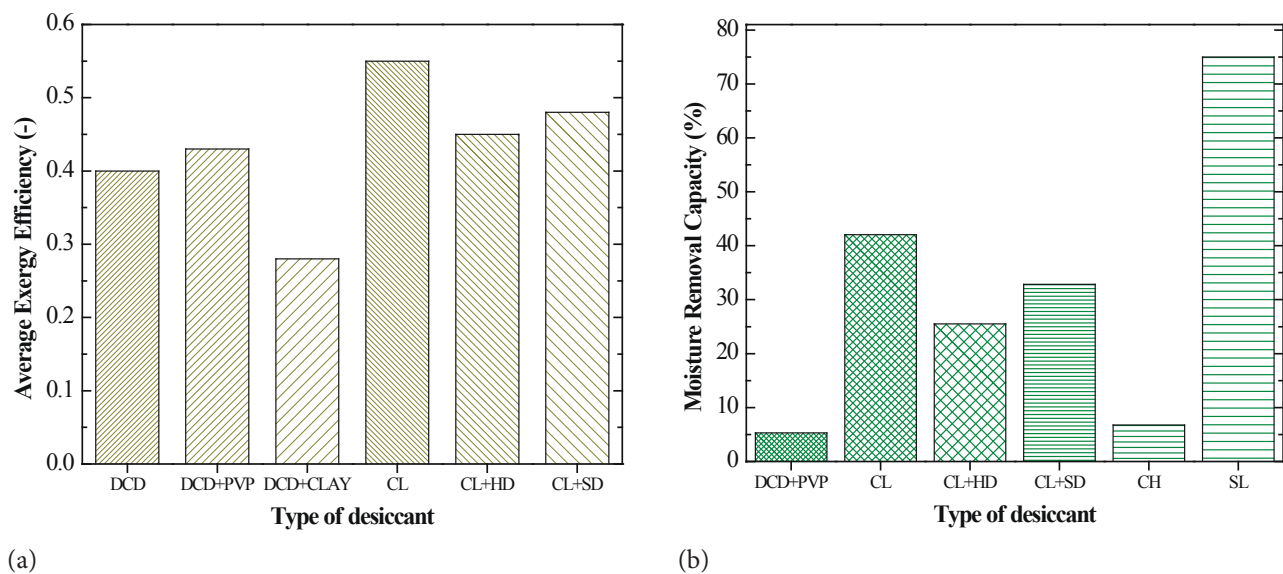


Figure 12. Assessment of (a) exergy efficiency and (b) moisture removal capacity for different dehumidification systems

in sensible heat lowers the exergy efficiency in clay additives-based composite desiccant beds.

The average exergy efficiency of the present work clay-calcium chloride (CL), clay-horse dung-calcium chloride (CL+HD) and clay-sawdust-calcium chloride (CL+SD) are plotted in Figure 12(a) and are compared with the literature results. The moisture removal capacity is calculated by knowing the bed inlet and exit air humidity ratio and is plotted in Figure 12(b). The average exergy efficiencies for the dried cow dung (DCD), cow dung combined with polyvinyl pyrrolidone (DCD+PVP), and dried cow dung with clay (DCD+CLAY) dehumidification systems are 0.40, 0.43, and 0.35, respectively. Notably, the exergy efficiency of the DCD+PVP system is 1.8% and 3.6% higher than that of the DCD and DCD+clay systems. Correspondingly, the moisture removal capacity of DCD+PVP dehumidification systems is 5.3 % and 18.37% greater than DCD and DCD+clay systems [27]. Higher moisture removal capacity in the clay-calcium chloride dehumidification system means higher exergy efficiency presented in Figure 12(a). The results reveal lower exergy efficiency in the clay-additives-calcium chloride dehumidification system than in the clay-calcium chloride system. Lesser exergy efficiency is attributed to lesser moisture removal capacity, as presented in Figure 12(b). The moisture removal capacity of present work dehumidification systems is compared with DCD+PVP, clay-calcium chloride (CH) [39] and silica gel (SL) [44] dehumidification systems available in literature. Silica gel dehumidification system has a higher % moisture removal capacity of 75% compared to clay and clay-additives-based calcium chloride composite desiccants. However, silica gel exhibits higher dehumidification capacity but significantly increases the air temperature.

Clay-based dehumidification systems offer advantages with respect to moisture removal and temperature control.

Besides dehumidification, the advantage of clay-based desiccants is the lower temperature of the air that leaves the desiccant bed. The process air leaves the clay-based composite desiccants at much lower temperatures than the silica gel desiccant bed. The average drop in temperature of clay and clay-additives composite desiccant bed exit air concerning inlet air temperature is 0.80°C (fluidized bed) and 1.80 °C (packed bed) [39]. The average silica gel bed exit air temperature increase is 17.10°C concerning inlet air temperatures [44]. Lower temperatures prevailing in clay and clay-additives composite desiccant systems as compared to silica gel will promote cooling energy saving in sorption-based systems.

CONCLUSION

In this work, an experimental study was conducted to understand the adsorption characteristics of a new composite desiccant prepared from natural materials. Utilization of naturally available clay and biomaterials such as horse dung and sawdust, which are considered as wastes added value in the preparation of clay-based composite desiccants. Horse dung traditionally employed in the pot making finds relevance in desiccant preparation. The following conclusions can be drawn from the experimental study on transient adsorption characteristics of atmospheric water vapor with clay and clay-additives based calcium chloride composite desiccants in vertical fluidized bed:

1. The scanning electron microscope micrographs reveal the presence of interparticle macro-pores and higher mass of calcium chloride in clay-calcium chloride adsorbents.

2. The fluidization of higher porosity and low-density desiccants decreases the mass transfer area involved in adsorption. For the set period of the process, the lower active interfacial area may lead to a slower adsorptive nature of clay-additives-calcium chloride desiccants compared to clay-calcium chloride adsorbents in fluidization.
3. The linear porosity distribution in fluidization demonstrates higher total adsorption capacity for clay-based desiccants, which is 38% and 12 % higher than that of clay-horse dung- calcium chloride and clay-sawdust-calcium chloride desiccant dehumidification systems.
4. Clay-calcium chloride vertical fluidized bed dehumidification system achieves higher total heat load reduction of 62% and 52% as compared to clay-horse dung-calcium chloride and clay-sawdust-calcium chloride desiccant dehumidification systems. As a result the overall heat load of the cooling system will significantly decrease.
5. The exergy efficiency analysis reveals that the clay- calcium chloride fluidization dehumidification system has a better exergy efficiency of 55% as compared to 45% for clay-additives-calcium chloride composite desiccants.
6. Using waste material like horse dung and sawdust with clay in desiccant preparation leads to a new family of desiccants making the technology more eco-friendly and helping promote sustainable goal development.

NOMENCLATURE

Symbol Meaning

A_1	Cross sectional area of pipe (m^2)
A_2	Cross sectional area of orifice meter (m^2)
C_p	Specific heat ($J/kg\ K$)
C_d	Coefficient of discharge of orifice (-)
d	Diameter (m)
E_X	Exergy (J)
E_Y	Exergy destruction (J)
g	Acceleration due to gravity (m/s^2)
h	Manometer deflection (m)
h_{fg}	Latent heat of vapor (J/kg)
L	Length of bed (m)
m	Mass flow rate (kg/s)
m	Weight of desiccant bed (g)
M	Total adsorption rate (g/s)
P	Pressure (N/m^2)
Q_{act}	Actual volume flow rate (m^3/s)
Q_S	Sensible heat load (J/kg)
Q_L	Latent heat load (J/kg)
R	Gas constant ($J/kg\ K$)
H_R	Humidity ratio of air (kg/kg of dry air)
t	Time (s)
T	Temperature ($^{\circ}C$)
v	Superficial velocity (m/s)
x	Manometer deflection (m)
z	Step length of the bed (m)

Greek symbols

ε	Porosity (-)
ρ	Density (kg/m^3)
η	Exergy efficiency (-)
ϕ	Relative Humidity (-)
ω	Overall uncertainty (-)
∂	Differential operator (-)
Δ	Small change (-)

Subscripts

a	Process air
ads	Adsorption
b	Bed
e	Exit
sat	Saturation
min	Minimum
max	Maximum
mf	Minimum fluidization
i	Inlet
o	Ambient
w	Vapor

Abbreviations

$CaCl_2$	Calcium chloride
GI	Galvanized iron
RH	Relative humidity
SEM	Scanning electron microscope
SHF	Sensible heat factor
VFD	Vertical fluidized bed

ACKNOWLEDGEMENT

The authors acknowledge the administrative and technical support provided by V. P. Dr. P. G. Halakatti College of Engineering and Technology Vijayapur-586103, Karnataka state, India, to carry out this research study.

CONFLICTS OF INTEREST

The authors declare no conflict of interest.

CREDIT AUTHORSHIP CONTRIBUTION STATEMENT

C R Hiremath: Conceptualization, Investigation, Methodology, Writing - original draft. **Ravikiran Kadoli:** Supervision, Writing - review and editing. **R J Talapati:** Writing - review and editing

REFERENCES

- [1] Zhang Y, Wang W, Zheng X, Cai J. Recent progress on composite desiccants for adsorption-based dehumidification. *Energy* 2024;302:131824. [CrossRef]
- [2] Kaushik SC, Verma A, Tyagi SK. Advances in solar absorption cooling systems: an overview. *J Therm Eng* 2024;10:1044-1067. [CrossRef]

- [3] Sharafian A, Bahrami M. Assessment of adsorber bed designs in waste-heat driven adsorption cooling systems for vehicle air conditioning and refrigeration. *Renew Sustain Energy Rev* 2014;30:440-451. [\[CrossRef\]](#)
- [4] Subramanyam N, Maiya MP, Murthy SS. Application of desiccant wheel to control humidity in air-conditioning systems. *Appl Therm Eng* 2004;24:2777-2788. [\[CrossRef\]](#)
- [5] Enteria N, Mizutani K. The role of the thermally activated desiccant cooling technologies in the issue of energy and environment. *Renew Sustain Energy Rev* 2011;15:2095-2122. [\[CrossRef\]](#)
- [6] Hiremath CR, Ravikiran K. Experimental analysis of low-temperature grain drying performance of vertical packed clay and clay-additives composite desiccant beds. *Sadhana* 2021;46:1-6. [\[CrossRef\]](#)
- [7] Ji JG, Wang RZ, Li LX. New composite adsorbent for solar-driven fresh water production from the atmosphere. *Desalination* 2007;212:176-182. [\[CrossRef\]](#)
- [8] Lv B, Zhao Z, Deng X, Fang C, Xing B, Dong B. Hydrodynamics and adsorption performance of liquid-solid fluidized bed with granular activated carbon for removal of copper ions from wastewater. *J Clean Prod* 2021;328:129627. [\[CrossRef\]](#)
- [9] Krzywanski J, Grabowska K, Sosnowski M, Zylka A, Kulakowska A, Czakiert T, et al. Heat transfer in adsorption chillers with fluidized beds of silica gel, zeolite, and carbon nanotubes. *Heat Transf Eng* 2021;43:172-182. [\[CrossRef\]](#)
- [10] Hamed AM. Experimental investigation on the adsorption/desorption processes using solid desiccant in an inclined-fluidized bed. *Renew Energy* 2005;30:1913-1921. [\[CrossRef\]](#)
- [11] Horibe A, Husain S, Inaba H, Haruki N, Tu P. An experimental investigation of sorption process in fluidized bed with cooling pipe. *J Heat Transf* 2008;130:114509. [\[CrossRef\]](#)
- [12] Ye J, Luo Q, Li X, Xu Q, Li Z. Sorption drying of soybean seeds with silica gel in a fluidized bed dryer. *Int J Food Eng* 2008;4. [\[CrossRef\]](#)
- [13] Sobrino C, Almendros-Ibáñez JA, Santana D, De Vega M. Fluidization of Group B particles with a rotating distributor. *Powder Technol* 2008;181:273-280. [\[CrossRef\]](#)
- [14] Hamed AM, Abd El Rahman WR, El-Emam SH. Experimental study of the transient adsorption/desorption characteristics of silica gel particles in fluidized bed. *Energy* 2010;35:2468-2483. [\[CrossRef\]](#)
- [15] Wang Q, Gao X, Xu JY, Maiga AS, Chen GM. Experimental investigation on a fluidized-bed adsorber/desorber for the adsorption refrigeration system. *Int J Refrig* 2012;35:694-700. [\[CrossRef\]](#)
- [16] Horibe A, Haruki N, Hiraishi D. Continuous sorption and desorption of organic sorbent powder in two connected fluidized beds. *J Therm Sci Technol* 2012;7:563-576. [\[CrossRef\]](#)
- [17] Zettl B, Englmaier G, Somitsch W. An open sorption heat storage concept and materials for building heat supply. *Energy Procedia* 2015;73:297-304. [\[CrossRef\]](#)
- [18] Li X, Wang L, Jia L, Cai W. Numerical and experimental study of a novel compact micro fluidized beds reactor for CO₂ capture in HVAC. *Energy Build* 2017;135:128-136. [\[CrossRef\]](#)
- [19] Chairunnisa, Yu H, Saren S, Miksik F, Conte P, Miyazaki T, et al. A review of recent advances in sustainable preparation of high-performing activated carbon for dehumidification technology. *J Mater Sci* 2024;59:1-36. [\[CrossRef\]](#)
- [20] Mittal H, Al Alili A, Alhassan SM. Development of high efficacy super-porous hydrogel composites-based polymer desiccants to capture water vapors from moist air. *Adsorption* 2024;30:841-857. [\[CrossRef\]](#)
- [21] Kumar M, Yadav A. Composite desiccant material "CaCl₂/Vermiculite/Saw wood": A new material for fresh water production from atmospheric air. *Appl Water Sci* 2017;7:2103-2111. [\[CrossRef\]](#)
- [22] Hiremath CR, Kadoli R, Katti VV. Experimental and theoretical study on dehumidification potential of clay-additives based calcium CaCl₂ desiccants. *Appl Therm Eng* 2018;129:70-83. [\[CrossRef\]](#)
- [23] Liang JD, Hsu CY, Hung TC, Chiang YC, Chen SL. Geometrical parameters analysis of improved circulating inclined fluidized beds for general HVAC duct systems. *Appl Energy* 2018;230:784-793. [\[CrossRef\]](#)
- [24] Timsina R, Thapa RK, Moldestad BME, Eikeland MS. Effect of particle size on flow behavior in fluidized beds. *Int J Energy Prod Manag* 2019;4:287-297. [\[CrossRef\]](#)
- [25] Singh A, Kumar S, Dev R. Studies on cocopeat, sawdust and dried cow dung as desiccant for evaporative cooling system. *Renew Energy* 2019;142:295-303. [\[CrossRef\]](#)
- [26] Politi D, Sidiras D. Modified spruce sawdust for sorption of hexavalent chromium in batch systems and fixed-bed columns. *Molecules* 2020;25:5156. [\[CrossRef\]](#)
- [27] Dasar SR, Boche AM, Yadav AK. Sorption-desorption characteristics of dried cow dung with PVP and clay as composite desiccants: Experimental and exergetic analysis. *Renew Energy* 2023;202:394-404. [\[CrossRef\]](#)
- [28] Suranjan Salins S, Reddy SK, Kumar S, Nair PS. Application of biomass-based wood shaving packing in a liquid desiccant dehumidification system - a source of sustainable energy. *Int J Sustain Energy* 2024;43:2287783. [\[CrossRef\]](#)
- [29] Hiremath CR, Kadoli R. Experimental studies on heat and mass transfer in a packed bed of burnt clay impregnated with CaCl₂ liquid desiccant and exploring the use of gas side resistance model. *Appl Therm Eng* 2013;50:1299-1310. [\[CrossRef\]](#)

- [30] Chen H, Peng YH, Wang YL. Thermodynamic analysis of hybrid cooling system integrated with waste heat reusing and peak load shifting for data center. *Energy Convers Manag* 2019;183:427-439. [\[CrossRef\]](#)
- [31] Hiremath CR, Katti VV, Kadoli R. Experimental determination of specific heat and thermal conductivity of clay+additives-CaCl₂ composite desiccant. *Procedia Mater Sci* 2014;5:188-197. [\[CrossRef\]](#)
- [32] Wu SY, Baeyens J. Effect of operating temperature on minimum fluidization velocity. *Powder Technol* 1991;67:217-220. [\[CrossRef\]](#)
- [33] Rachayya HC. Studies on dehumidification potential of clay with additives and impregnated with CaCl₂ composite desiccants [dissertation]. Surathkal (IN): National Institute of Technology Karnataka; 2019.
- [34] Ramzy A, Kadoli R. Modified PGC model and its validation by experiments for heat and moisture transfer analysis in a vertical fluidized desiccant bed. *Appl Therm Eng* 2015;81:83-91. [\[CrossRef\]](#)
- [35] Liang JD, Hsu CY, Hung TC, Chiang YC, Chen SL. Geometrical parameters analysis of improved circulating inclined fluidized beds for general HVAC duct systems. *Appl Energy* 2018;230:784-793. [\[CrossRef\]](#)
- [36] Sarker MSH, Ibrahim MN, Aziz NA, Punan MS. Energy and exergy analysis of industrial fluidized bed drying of paddy. *Energy* 2015;84:131-138. [\[CrossRef\]](#)
- [37] Kumar R, Mishra DR, Dumka P. Improving solar still performance: A comparative analysis of conventional and honeycomb pad augmented solar stills. *Solar Energy* 2024;270:112408. [\[CrossRef\]](#)
- [38] Moffat RJ. Describing the uncertainties in experimental results. *Exp Therm Fluid Sci* 1988;1:3-17. [\[CrossRef\]](#)
- [39] Hamed AM. Theoretical and experimental study on the transient adsorption characteristics of a vertical packed porous bed. *Renew Energy* 2002;27:525-541. [\[CrossRef\]](#)
- [40] Demir H, Mobedi M, Ülkü S. Effects of porosity on heat and mass transfer in a granular adsorbent bed. *Int Commun Heat Mass Transf* 2009;36:372-377. [\[CrossRef\]](#)
- [41] Zheng X, Wang LW, Wang RZ, Ge TS, Ishugah TF. Thermal conductivity, pore structure and adsorption performance of compact composite silica gel. *Int J Heat Mass Transf* 2014;68:435-443. [\[CrossRef\]](#)
- [42] Majumdar P, Sarwar MK. Performance of a desiccant dehumidifier bed with mixed inert and desiccant materials. *Energy* 1994;19:103-116. [\[CrossRef\]](#)
- [43] Wang T, Tian S, Li G, Sheng M, Ren W, Liu Q, et al. Experimental study of water vapor adsorption behaviors on shale. *Fuel* 2019;248:168-177. [\[CrossRef\]](#)
- [44] Pesaran AA, Mills AF. Moisture transport in silica gel packed beds-II. Experimental study. *Int J Heat Mass Transf* 1987;30:1051-1060. [\[CrossRef\]](#)



Research Article

Performance optimization of low-head vertical axis impulse turbine runners for nozzle angle, nozzle diameter, and nozzle standoff distance using response surface methodology

H. N. LAKDAWALA^{1,*}, V. K. PATEL², G. P. BAKHRU³, D. N. MODI⁴

¹Department of Mechanical Engineering, Dr. S. & S.S. Ghandhy Government Engineering College, Surat, Gujarat, 395001, India

²Department of Mechanical Engineering, Sardar Vallabhbhai National Institute of Technology (SV NIT), Surat, Gujarat, 395007, India

³Department of Mechanical Engineering, Government Engineering College, Valsad Gujarat, 396001, India

⁴Department of Mechanical Engineering, University of Windsor, Windsor, Ontario, N9B 3P4, Canada

ARTICLE INFO

Article history

Received: 27 August 2024

Accepted: 25 December 2024

Keywords:

3D Printed Buckets; Hydraulic Nozzles; Pico Hydro; Response Surface Methodology; Turgo Turbine

ABSTRACT

Pico-scale turbines, such as Turgo and Pelton, are site-specific and show variable performance under various operating and design parameters. Experimental investigation and response surface methodology were combined to optimize the geometrical parameters of Pelton and Turgo turbine runners in the present study. Low-cost, lightweight hybrid runners with 3D-printed buckets and aluminum runner discs with a vertical turbine axis were designed for low-head applications. Multivariate statistical evaluation and response surface methodology were conducted using Design Expert 13.0 software, with a central composite design applied to analyze results and optimize parameters through 80 test runs. Quadratic models describing the hydraulic efficiency characteristics of impulse turbine runners were developed via ANOVA. The study examined nozzle diameters (10-14 mm), angles (90°-95°), numbers (1 or 2), and standoff distances (40-60 mm). The optimized condition for the Turgo turbine runner was achieved with a 95° nozzle angle, 14 mm nozzle diameter, and a 40 mm standoff distance. The hydraulic efficiency of 66.19% is found, with significant model parameters having P-values below 0.0001. The findings indicated a maximum efficiency of 66.55% under optimized conditions, closely matching the proposed model with an error of 0.54%. Thus, the Turgo turbine is highly efficient and suitable for Pico hydro off-grid applications.

Cite this article as: Lakdawala HN, Patel VK, Bakhru GP, Modi DN. Performance optimization of low-head vertical axis impulse turbine runners for nozzle angle, nozzle diameter, and nozzle standoff distance using response surface methodology. J Ther Eng 2025;11(5):1392–1419.

INTRODUCTION

With a permanent magnet, a Pico turbine combines a 220V single-phase alternator with a hydraulic turbine

or propeller. “Pico” denotes the power range of the alternator. Pelton waterwheel for the high heads is seated, Francis turbine is horizontal, and Run of the River Kaplan

*Corresponding author.

*E-mail address: hnlgecs@gmail.com

This paper was recommended for publication in revised form by
Assigned Editor-in-Chief Ahmet Selim Dalkilic



is a vertical Pico hydro turbine, one of the three hydraulic turbine varieties. The hydro impulse turbine is a popular, dependable, and sustainable option in hydroelectric plants, especially well-suited for moderate and high head applications. Technologies exploiting low and ultra-low head water potential sites have become popular due to their minimal environmental impact. Furthermore, putting small, creative water-related projects into action offers underdeveloped nations a cost-effective way to produce renewable energy.

A high-pressure water jet with kinetic energy is key in producing impulses. The performance of the impulse turbine suffers due to alterations in jet characteristics. The jet size and shape affect the performance of the impulse turbine. The turbine's efficiency was mainly influenced by the nozzle opening diameter or the jet diameter, speed ratio, blade count, and jet angle or the nozzle tilt angle [1]. The variability in the length-to-diameter ratio of the jet provides a method for controlling jet surface roughness over a substantial range [2]. The exact calculation of a Turgo turbine's wheel-to-nozzle diameter ratio, or D/d , is necessary since it influences wheel velocity, which influences efficiency and performance [3]. Smaller jet diameters and higher flow rates can be used to obtain the ideal operating state since the maximum brake power increases with increased discharge due to an increase in torque [4]. Higher water heads lead to increased power generation, while electric power increases linearly with nozzle diameter enlargement [5]. In descending order, the circular water jet exhibits the highest peak pressure at identical inlet pressures, followed by the common fractal shapes like square, triangular, cross-shaped, and elliptical water jets [6]. At the minimum standoff distance of the nozzle, axial flow predominates in the water jet; otherwise, radial flow is more dominant [7]. The main jet diffusion factors influencing jet quality are eccentricity-induced pressure fluctuations and jet velocity imbalance. [8]. Apart from the primary flow in the bucket, the weak secondary flows from bends or bifurcations in the Pelton turbine distributor cause minor disturbances, affecting the type of jet shape, alignment of the jet with the bucket, and shifting of the water jet core from the nozzle centerline [9]. The efficiency of Turgo and Pelton turbine runners, used in Pico hydro applications, suffers from variations in speed ratio and jet misalignment [10]. The hydrodynamically designed buckets of the Turgo impulse turbine with an optimum jet angle are required for maximum power with the best efficiency [11]. According to numerical modeling, the needle tip's position changed the velocity distribution of a Pelton turbine's free jet [12].

The variation in available head and water discharge through the turbine affects the hydraulic performance of turbine runners. In a novel way, the Turgo impulse turbine boosts a hydraulic impulse turbine's capacity while regulating flow with the nozzle and spear injector mechanism of the Pelton turbine [13]. The single-jet Turgo turbine was found to have improved performance over the

baseline design, with optimized parameters by multi-criteria analysis [14,15]. Also, for a Turgo impulse turbine, the discharge plays a vital role; as the discharge increases, the specific speed rises, while with a higher net head, the specific speed decreases. Additionally, increasing the net head boosts power output [16], and reducing the water head decreases water discharge, decreasing rotational speed, torque, brake power, and efficiency [17]. The water flow rate is the only factor affecting pressure, the power produced at the alternator, and the sufficient torque applied to the shaft [18]. The Pico hydro Turgo turbine has hill charts similar to the regular turbine for low-head applications at the constant head for varying flow conditions and optimized jet angle for remote area applications [19,20]. Nowadays, researchers are attracted to the low-head operation of the Pelton runner. With a flow rate of 33 liters per minute and a static pressure head (very low) of 2.5 meters, a model of micro Pelton turbine was designed to operate at maximum efficiency using a revolutionary hydraulic system [21], and a modified Pico-sized Pelton wheel model was adapted to utilize the velocity of rainwater harvested from rooftops to supplement energy generation [22]. Micro-hydro Pelton turbines can be designed, modeled, and analyzed for remote site-specific conditions [23]. A Pelton micro hydropower turbine with a maximum head of 15 meters in rural Ethiopia could produce 32 kilowatts of electricity utilizing available water [24]. The novel design of the Turgo turbine was tested and operated with competitive geothermal efficiency to get maximum yield from steam [25]. Optimized conditions are required to operate low-head Turgo and Pelton wheel turbines for maximum hydraulic efficiency, as they are site-specific and geometry-specific [26-28].

Three-dimensional unstable flow is used to analyze flow through buckets, which is essential to the runners' performance. The fluid flow through the Turgo turbine bucket is simpler than that of the Pelton turbine. Computational Fluid Dynamics (CFD) analysis on a Pelton turbine using the RANS (Reynolds-averaged Navier-Stokes) -based Eulerian scheme predicted the complex flow patterns, jet interactions, and post-impact water interference with the buckets [29]. The instability of free surface fluid flow (sheets) in rotating buckets in three dimensions was reproduced by the study methods of the two-phase homogeneous model. [30], and with a non-orthogonal curvilinear coordinate-based boundary-fitted grid (BFG) [31], can predict the dynamic flow patterns and torque fluctuations. However, roughness on the bucket surface hinders free surface flow within the bucket, significantly impacting Pelton turbine performance and hydraulic losses, and the level of roughness magnitude influences it [32]. The bucket cutout design with the "u" type cutout and simple, improved design provides the best performance for a Pico hydro Pelton turbine [33,34]. A runner of a Pelton turbine should have the optimum number of moving buckets to operate efficiently and prevent jet interference [35]. The

enhanced hydraulic efficiency of the runner of the Turgo model can be achieved by parameterizing and numerically tuning stochastic optimization [36]. Flaws and unsuccessful designs can be detected early in the process using a 3D-printed Pelton turbine runner [37]. Coconut shell is suggested as a viable material for Turgo turbine buckets due to its robust strength and ability to endure water pressure effectively [38]. Also, buckets of the aluminum alloy, austenitic and martensitic stainless steel, and grey cast iron material have shown enough structural integrity with different bucket angles upon evaluation by numerical simulation for equivalent stress distribution and total deformation [39].

Response Surface Methodology (RSM) is a valuable tool for optimizing the turbine parameters. The Turgo turbine is effective for low-head applications (3.5 m) and can efficiently operate across various flow conditions with varying jet-to-diameter ratios. Studies have shown that RSM helps maximize energy extraction in Turgo impulse turbines, as studied by Gallego et al. [40] and for optimizing the housing basin and fluid-laden inlet channel of the gravitational water vortex hydraulic turbines by Velásquez et al. [41]. Betancour et al. [42] successfully applied RSM with the Central Composite Design technique to design a 3D-printed Gravitational Vortex Turbine (GVT) runner, with satisfactory results across a range of parameters. Guerra et al. [43] also utilized RSM to optimize the number of vanes (blades) and turbine hub-to-rotor outer diameter ratio, parameters that significantly impact the hydraulic performance of a siphon turbine. Apart from radial flow, RSM is also useful for optimizing the design of an AST (Archimedes screw turbine, an axial flow turbine) for hydrokinetic power generation applications [44]. Beyond turbine design, RSM applies to all mechanical systems to effectively optimize performance parameters. RSM has also been applied to improve heat pipes' thermal performance, as Singh et al. studied [45], and it is employed to optimize the blowdown operation of double-effect vapor absorption refrigeration systems studied by Gambhir et al. [46], and to evaluate single-effect vapor absorption systems for the dairy industry, carried out by Solanki et al. [47]. Ali [48] also demonstrated the utility of optimization for performance analysis in internal combustion spark ignition engines by adjusting low-octane gasoline with Methyl tert-butyl ether additives.

The literature review shows fewer studies on low-head vertical-axis Turgo turbines than Pelton turbine runners. Various bucket materials, such as coconut shells, aluminum, and SS 304, have been explored for their performance. However, sustainable materials like Polylactic Acid (PLA) as bucket material, combined with an aluminum runner disc, have not been reported in hybrid material runners so far. While other Response Surface Methodology (RSM) methods have been explored, the Central Composite Design (CCD) method is introduced to compare optimal operating conditions for achieving

maximum hydraulic performance. Although non-dimensional numbers and ratios have been studied in past studies, the nozzle standoff distance, which considers the jet expansion effect, has not been included. This study thus bridges the gap between design and performance by optimizing most operating parameters that affect the establishment of Pico hydro off-grid power production facilities. This study is dedicated to enhancing the hydraulic efficiency of the designed runners by applying response surface methodology. It emphasizes key operational factors that play a crucial role in turbine performance, aiming to refine and improve overall functionality. The flow chart depicts the optimization methodology for the present work, as described in Figure 1.

The design of identical runners with fixed parameters is followed by manufacturing. The analysis of variance for subsequent optimization and validation tests follows the Response Surface Methodology (RSM) with a Central Composite Design (CCD).

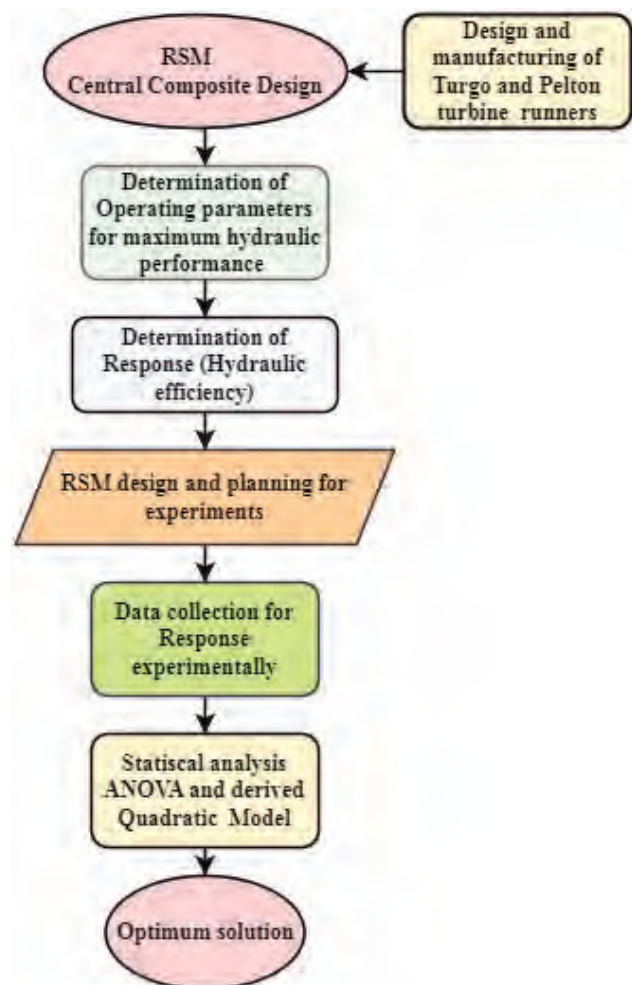


Figure 1. Flow chart for the work carried out.

The primary objective of the current experimental investigation includes

- 1) To study the effect of different nozzle diameters (d_j), the nozzle standoff distance (N_{sof}), number of nozzles (n), and nozzle angle (α) on the Pelton and Turgo impulse turbine runner.
- 2) Optimize the operating parameters for the current study using Response Surface Methodology (RSM) and Central Composite Design (CCD) to improve maximum hydraulic performance for the proposed runners.
- 3) To compare the optimized performance of runners with similar previous work.

METHOD AND MATERIALS

Test Configuration

The delivery line comprises 4-inch galvanized iron pipes, and a flexible hose connects to the nozzle spear mechanism assembly and header tapping. The sucked water through the collecting water sump at the water by the pump is discharged to the delivery line with a high head in an upward direction. The valve position is regulated, allowing water to flow further in the test rig. The head of 10 m is kept constant with the help of pressure gauges mounted on the header pipe. The pressure gauge measures the pressure before delivery to the flexible hose. The flow line is facilitated with overflow protection and controlled water flow.

The test rig setup is depicted in Figure 2, with all the instruments for measuring quantities using different runners and with the same pitch circle diameter (PCD), i.e., 100 mm. The turbine runners are struck with a tangentially arranged set of nozzles placed at PCD. The nozzles of 10, 12,

and 14 mm of aluminum have been attached to the nozzle hub. The variable flow condition is achieved with the help of the spear mechanism. The spear mechanism and the body were made from SS 304. The square acrylic frame of 1000 mm in length and width houses the vertical shaft and bearings. The runner is kept with a vertical turbine axis configuration. Also, the steel shaft has a pulley of 50 mm effective diameter fixed midway above the transparent acrylic casing to measure shaft power with absorption-type dynamometer elements connected to it. The rope, pulley, and dead weight constitute the absorption type diameter for torque measurement. The non-contact type tachometer was attached to the frame to measure runner speed. Flow measurement was achieved with the Electromagnetic type flow meter (Flowtech Digital Water Flow Meter, Model: MAG EMFM-0634, maximum process pressure 25 kg/cm²)

Table 1. Characteristic parameters of the centrifugal pump used in the experimental setup

Description of parameter	Value
Model	Kirloskar KS-516+, Monoblock
Power rating	3.5 kW
Phase, Voltage	Three-phase, 300-400 V
Flowrate	23.7 - 13.2 LPS
Head	12-18 m
Suction lift	7.5 m
Impeller	Cast iron

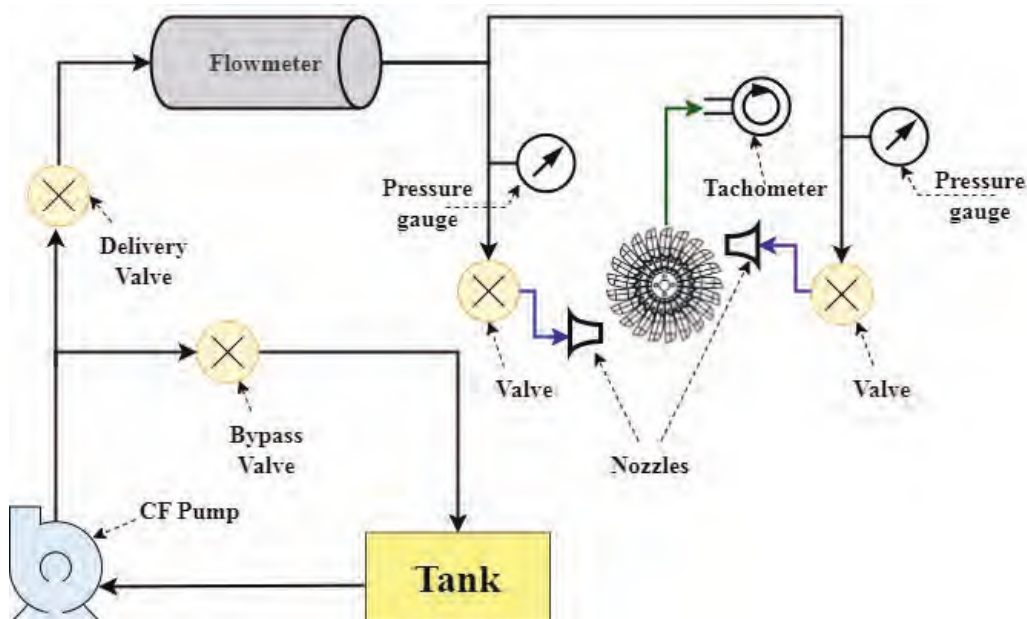


Figure 2. Line diagram for the experimental setup for testing runners with instruments measuring the process parameters.

with a measuring capacity of ± 0.00069 . The nozzle angle variation was set up with a protractor for each run.

The centrifugal pump attached to the bottom of the experimental setup has the following characteristic parameters depicted in Table 1. The pump has a cast iron motor body and an IP44 rating for protection. The pump has a minimum variation in efficiency over the entire operating range.

The Turgo and Pelton turbine runners were designed for experimentation. The material and specifications for the buckets and runner are depicted in Table 2 below. The runners are designed and identically sized for PCD, as mentioned in Appendix A1, and the bucket material used is the same.

The aluminum runner disc, which is 66 mm in diameter and 4 mm in thickness, has radially drilled holes for bolting buckets with 2 nos. M4 size bolts firmly, as shown in Figure 3. The runner material, aluminum, was selected for its lightweight and manufacturing ease for the present study.

The close conduit of the test setup consists of a flow measuring device, a pressure sensor, and a spear nozzle assembly at the end. An electric motor-driven impeller pump circulates the water in the entire test rig. The buckets of PTR and TTR were designed for micro Pelton turbines and suitable for Pico hydro application for a 10 m head. The buckets were designed and 3D printed using PLA material on a **Prusa i3 FDM 3D Printer** (Bed dimensions 200 x 200 x 200 mm, material layer thickness 0.05 to 0.4mm, article printing speed 20 to 200 mm/s, 3D Printer Standard Extruder

Table 2. Pelton and Turgo turbine runner with the 3D printed Polylactic Acid (PLA) bucket

Sr.No.	Bucket element	Dimensions	
		Pelton Turbine Runner (PTR)	Turgo Turbine Runner (TTR)
1	Bucket width (B_b) x Length (L_b)	36 mm x 34 mm	18 mm x 34 mm
2	Splitter angle (θ_s)	30°	15°
3	Hemispherical vane thickness (t_h)	2 mm	2 mm
4	Overhang length from pitch circle (L_{ov})	36 mm	36 mm
5	Notch radius (R_n)	10 mm	10 mm
6	Spherical vane outer radius (R_v)	10 mm	-----
7	Overall length (L_o)	52 mm	52 mm
8	Connection bolt-hole diameter (d_h)	3.2 mm	3.2 mm
9	Aluminum runner plate diameter (D_r)	66 mm	66 mm
10	PCD of the runner (D_p)	100 mm	100 mm
11	Cutout Edge (B_c)	14 mm	-----
12	Bucket Exit Angle (β_2)	15°	-----

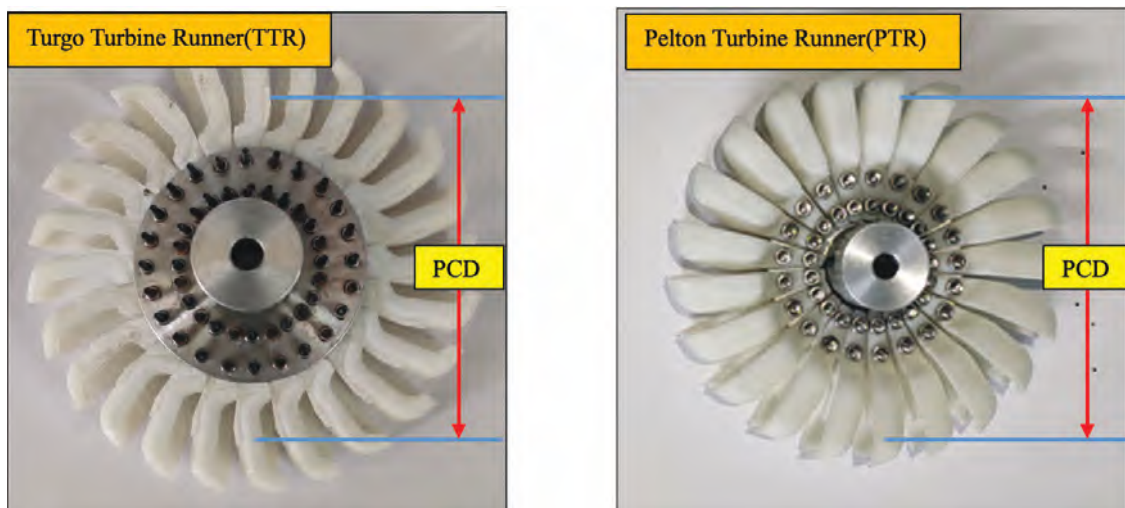


Figure 3. Designed and manufactured Turgo and Pelton turbine runners with 3D-printed buckets.

Diameter: 0.4mm, 3D Printing Software: CURA, Repetier-Host simplify3D, 3D Printing Filament Material: Polylactic Acid (PLA), Acrylonitrile Butadiene Styrene (ABS), High-Impact Polystyrene (HIPS), WOOD, Polycarbonate (PC), Poly Vinyl Chloride (PVC) and more, 3D Printing Material Diameter: 1.75mm, 3D Printing Accuracy: $\pm 0.1\text{mm}$) manufactured by **Make3d** at the AFMFP laboratory, SV NIT Surat.

The steel vertical shaft of 15 mm diameter is stepped to 8 mm diameter for 50 mm length at one end to accommodate the runners, as shown in Figure 4. The hub of the runner accommodates the M4 grub screw to prevent relative motion between the runner and shaft. The aluminum pulley with a rope wound measured the shaft torque. The deadweight and spring balance arrangement was used to measure the rope's differential tensions and shaft power. The optimum buckets for any given PCD are about 19 to 21 because no significant flow separation at the cutout edge occurs. For the experiment, 21 buckets were selected for the Pelton and 24 buckets for the Turgo turbines. The runners'

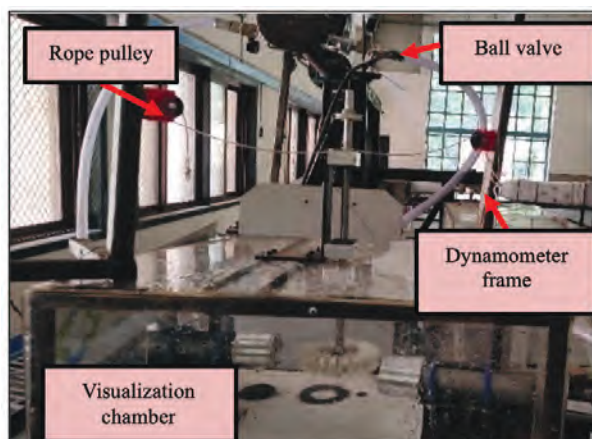
design and buckets were carried as per the micro hydro Pelton wheel turbine manual [49].

Study of Nozzle Standoff Distance and Jet Incidence Angle

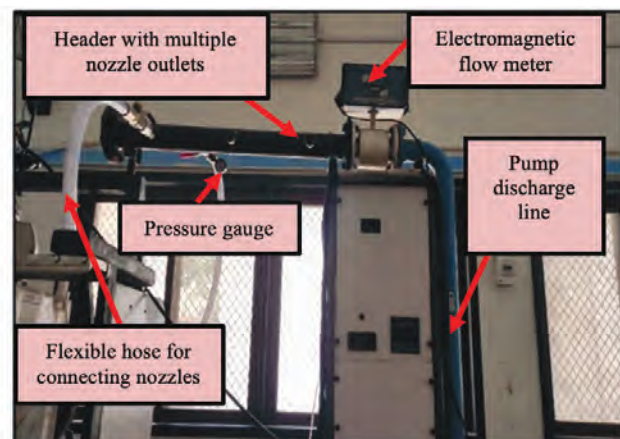
The linear distance between the bucket's splitter edge and the nozzle tip is known as the "nozzle standoff distance" in PTR. On the other hand, the surface inside the bucket has a point where the jet hits in the TTR. The arrangement shown in Figure 5 depicts the jet deviation for three steps (Normal, $+2.5^\circ$, and $+5^\circ$. So, from normal jet incidence (perpendicular to the bucket radial axis) at 90° , it varies between 92.5 and 95 degrees in the present study.

Study of Nozzle Diameter

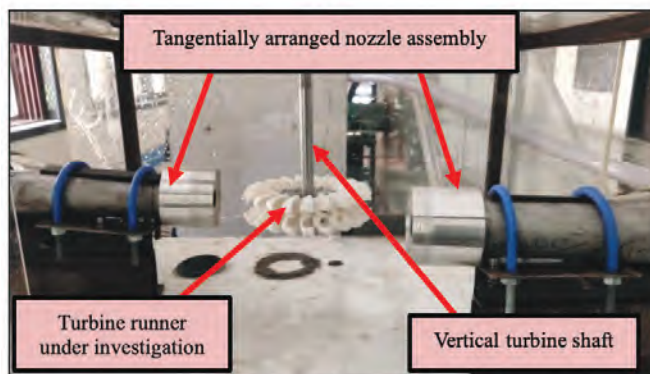
The aluminum nozzle hub is machined to make the dimensions shown in Figure 6(a) to reduce the spear rod assembly weight and ease machining. An orifice plate of thickness 4.77 mm acts as a nozzle with 80 degrees of divergence provided to the different concentrically drilled jet sizes.



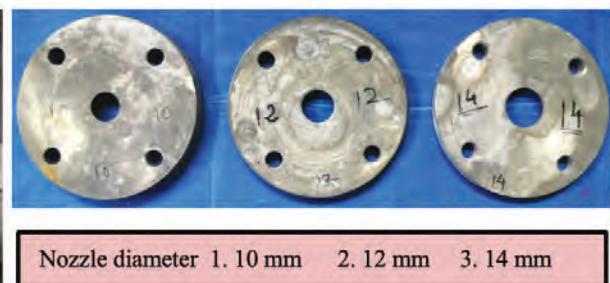
(a)



(b)



(c)



(d)

Figure 4. Experimental setup and tangential nozzle arrangement (a) the torque measuring arrangement, (b) the flow line and flow meter, (c) the test section-runner arrangement concerning nozzles, (d) the tested nozzles of diameters 10, 12, and 14 mm.

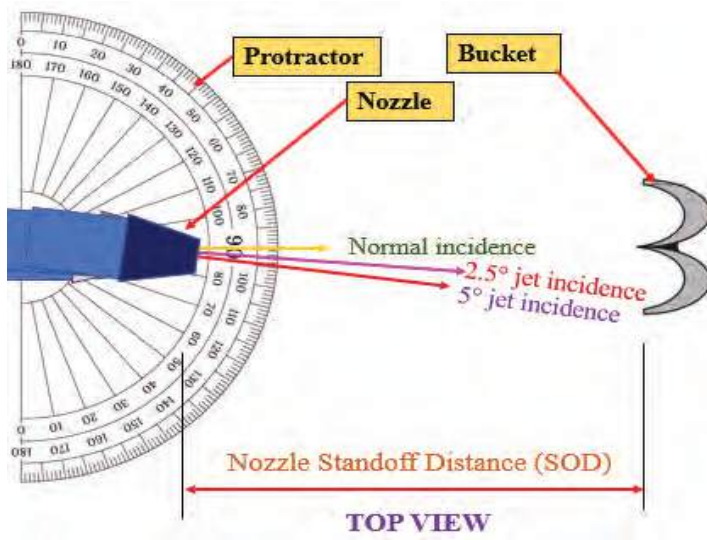


Figure 5. Concept of jet deviation (η_h) and Nozzle standoff distance (N_{sod}).

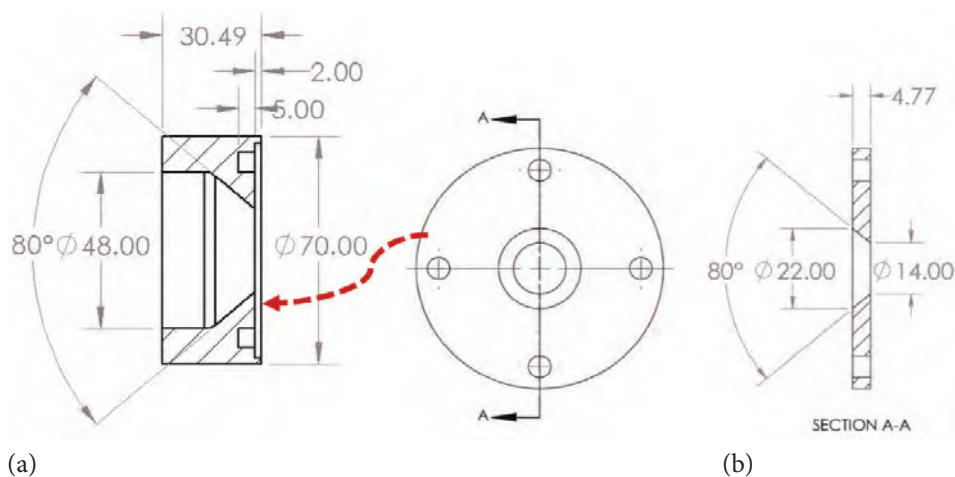


Figure 6. The specifications of (a) Nozzle hub and (b) Adjustable nozzle plate.

The nozzle(jet) sizes used in the current study are 10, 12, and 14 mm, as depicted in Figure 6(b). The sitting position of the orifice plate on the nozzle hub with the help of an M4 screw is shown with the red dotted arrow. The inside threaded side of the nozzle hub can be fitted to the spear rod mechanism. The interchangeable plates of different diameters produce jets of different sizes. The standoff distance, which is typically 4 to 5 d_j between the jet exit (nozzle tip) from the spear assembly and the jet striking point on the bucket, is used.

Investigation of Hydraulic Efficiency (η_h)

The various essential variables from the study's point of view are listed below in Table 3 with operating ranges. The dimensionless quantities compare the runner's operation range under investigation. In the case of the Turgo turbine runner (TTR), a point on the bucket is considered a normal

jet incidence point, and variation in angle is carried with respect to that point located from the bucket's outer edge.

The parameters shown in the above table are available for evaluation. The potential Energy provided to the turbine, or the hydraulic power to the turbine, is the product of the mass flow rate of fluid flowing through it $\dot{m}_f = \rho \times Q$ and the test-specific Energy $E_s = g \times H$. The values for range calculations are based on the direct effect on the evaluated parameter of the incurred parameter in the equations above. The data collected upon experimentation in the present study is reduced using equations mentioned in Table 4 below.

An uncertainty analysis is essential to envisage the correctness of measurements. The instruments measuring various physical quantities always suffer from inherent characteristics leading to false readings. According to the study, uncertainty was found for the head of water, angular

Table 3. Recorded input key global parameters with a range of work

Parameters	Notation	Range	
		PTR	TTR
Speed ratio (λ_s)	$= \frac{V_{jet}}{u}$	0.011-0.66	0.013-0.68
Flow ratio (ϕ_f)	$= \frac{Q}{N \times D_p^3}$	$6 \times 10^{-4} - 18.13 \times 10^{-4}$	$5.4 \times 10^{-4} - 10.14 \times 10^{-4}$
Jet ratio (ψ_j)	$= \frac{D_p}{d_j}$	6 - 10	6 - 10
Nozzle angle (α)	(α)	90-95°	90° - 95°
Energy coefficient (E_p)	$= \frac{g \times H}{N^2 \times D_p^2}$	0.0034-0.035	0.0032-0.027
Power coefficient (P_f)	$= \frac{P_d}{\rho \times N^3 \times D_p^5}$	$1.16 \times 10^{-8} - 7.6 \times 10^{-6}$	$1.16 \times 10^{-8} - 7.6 \times 10^{-6}$
Specific Speed (Ω_s)	$= \frac{\omega \times \sqrt{Q}}{(g \times H)^{\frac{3}{4}}}$	0.002 - 0.166	0.022 - 0.16
Power Specific Speed (Ω_{sp})	$= N \frac{\sqrt{\frac{P_d}{\rho}}}{(g \times H)^{\frac{5}{4}}}$	0.051 - 0.69	0.025 - 1.24
Bucket size ratio(δ_b)	$= \frac{L_b}{B_b}$	0.92	1.85

Table 4. Data Reduction

Parameters	Notation
Torque developed at the runner shaft, N-m.	$T_d = \frac{(W - S) \times D_e \times g}{2}$
Angular velocity(ω) of the runner, rad/s	$\omega = \frac{2 \times \pi \times N}{60}$
Power developed, Watt.	$P_d = T_d \times \omega$
Hydraulic power, Watt	$P_h = \rho \times g \times Q \times H$
Hydraulic efficiency (%)	$\eta_h = \frac{P_d}{P_h} \times 100$

speed, torque developed at the runner shaft, and flow rate for all the tested combinations with full gate opening (100%). Assuming volumetric η_v and mechanical efficiency η_m are approximately unity, the hydraulic efficiency is given by equation (1) as

$$\eta_h = \frac{T_d \times \omega}{\rho \times g \times Q \times H} \quad (1)$$

Since the experiments were carried out with a constant head, but considering frictional head loss and pipe fitting loss incurred to calculate the value of dH, the difference in measured and actual value of torque due to spring balance, which is taken care of by considering the most minor count. The measured value from the speed sensor and flow rate are also affected by the accuracy and subsequent error in measurement.

$$\frac{d\eta_h}{\eta_h} = \sqrt{\left(\frac{dH}{H}\right)^2 + \left(\frac{dT_d}{T_d}\right)^2 + \left(\frac{d\omega}{\omega}\right)^2 + \left(\frac{dQ}{Q}\right)^2} \quad (2)$$

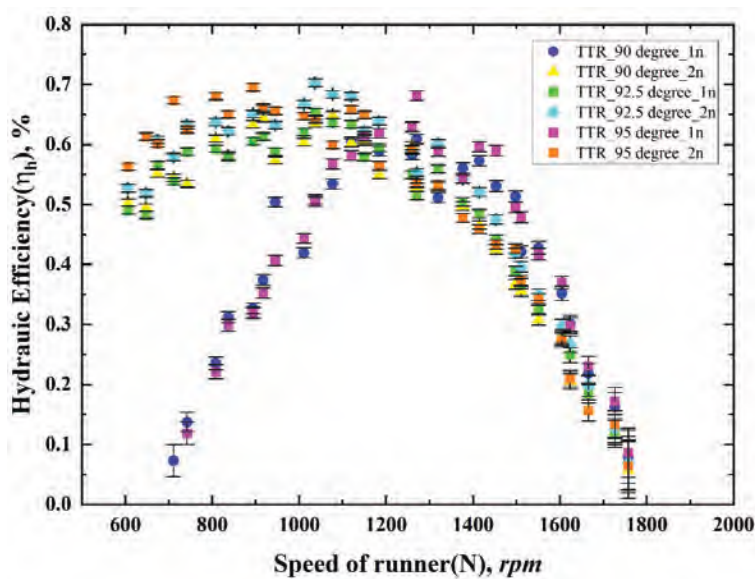
According to the study, a sample calculation for uncertainty with whole gate opening (100%) and nozzle diameter, nozzle angle (Jet deviation) in the case of both a Turgo Turbine Runner (TTR) and regular Pelton turbine runner (PTR) carried out for the available head, torque, angular speed, and the flow rate was $\pm 0.48\%$, $\pm 1.03\%$, $\pm 1.09\%$, and $\pm 0.01\%$ respectively as per equation (2).

Experimental Methodology

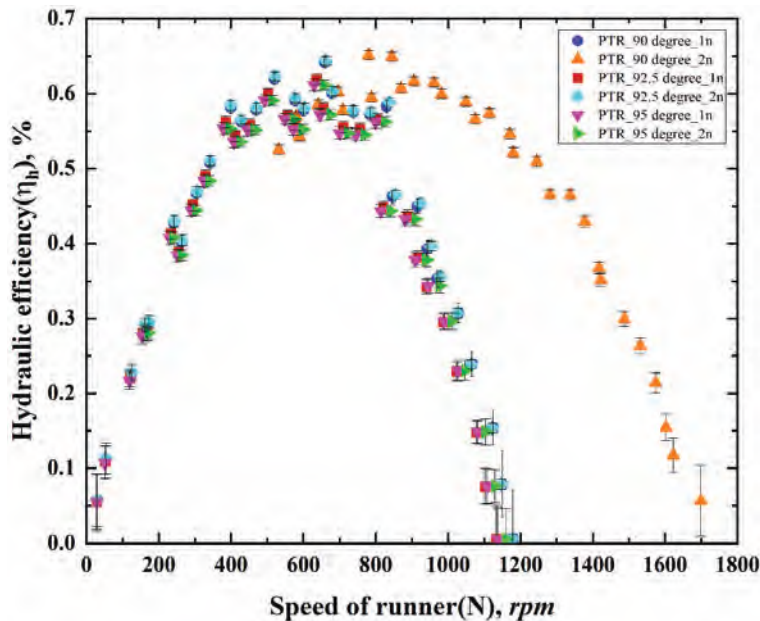
The hydraulic performance of runners is significantly influenced by the flow, speed, and energy coefficients [50],

[51]. The jet incidence angle, number of jets, and nozzle standoff distance are considered in the current study. Continuous experiments were conducted to minimize observational error and reproducibility. The experiments were conducted in a closed-circuit test facility at Advanced Fluid Mechanics and Fluid Power Laboratory, SV NIT, Surat. The maximum head available at the discharge line is 43 m, which is reduced to 10 m in a header for supplying the runner. The electromagnetic flowmeter (Flowtech MAG) is attached to the discharge line for accurate flow measurement. The spear rod mechanism at the nozzle distributes

the water flow with a variable flow area by driving the spear rod in and out. As depicted in Figure 3, the water jet to the runner is supplied with either a single or two nozzles. The different jet diameters 10, 12, and 14 mm were used in the present study with a 10 m head and 100 mm PCD impulse turbine runners, i.e., Turgo and Pelton turbines, in the present work. The observation was noted for the runner's speed, spring balance reading, slotted dead weight, and discharge, and processed further for analysis during each planned experimental run.



(a)



(b)

Figure 7. The performance of (a) Turgo and (b) Pelton turbine runners under variable parameters with Nozzle standoff distance ($N_{sod} = 40$ mm, Nozzle diameter (d_j) = 10 mm.

The experiments were carried out with all the possible combinations of parameters considered in the present study with both runners. The graph in Figure 7 depicts the PTR and TTR hydraulic performance for the 40 mm Nozzle standoff distance and 10 mm nozzle diameter. The graph has been plotted with errors evaluated by the uncertainty in the measurements of the parameters. Upon experiments and subsequent analysis, the TTR was more efficient than the PTR. It was found that the jet angle, jet diameter, and standoff distance had detrimental effects on the turbine performance. These independent parameters were taken for optimization and validation in the present work.

Response Surface Methodology (RSM)

The Response Surface Methodology (RSM) employs statistical and mathematical techniques to fit empirical data from experimental models, optimizing factors by minimizing experimental runs and analyzing factor interactions [52,53]. In the optimization process, the significance of the Response Surface lies in its function as a metamodel that encapsulates the Design of Experiments (DoE) data, enabling the formation of a rapidly executable surrogate model.

The standard RSM analyzed the performance of TTR and PTR, which were designed for Pico hydro applications. The optimization study in RSM can be performed in different stages, as described in Figure 8.

In the current study, three independent variables for the statistical experimental design: Two categorical elements, turbine runner type (PTR, TTR), and the number of nozzles (n , 1 or 2) are shown in Table 5, together with three numerical factors: nozzle angle (α , degrees), nozzle diameter (d_j , mm), and nozzle standoff distance (N_{sod} , mm). These factors' levels and ranges were modified in accordance with the experimental design.

$$Y = f(X_1, X_2, X_3, \dots, X_n) \quad (3)$$

In this study, equation (3), the Y represents the experimental response, specifically hydraulic efficiency, while X_n denotes the independent variables as action factors. The regressor variables are presumed to be continuous and independent controllable within the experimental framework, with negligible errors. The aim is to replicate the true functional relationship between the response surface and the independent variables [54]. The experimental design aimed to optimize the hydraulic efficiency (as a response variable), Y . Therefore, it is crucial to establish a dependable estimation of the relationship between the independent variables and the resulting response to ensure accuracy and reliability. To reduce mistakes and the impact of uncontrollable factors, the trials were randomized and presented by equation (4).

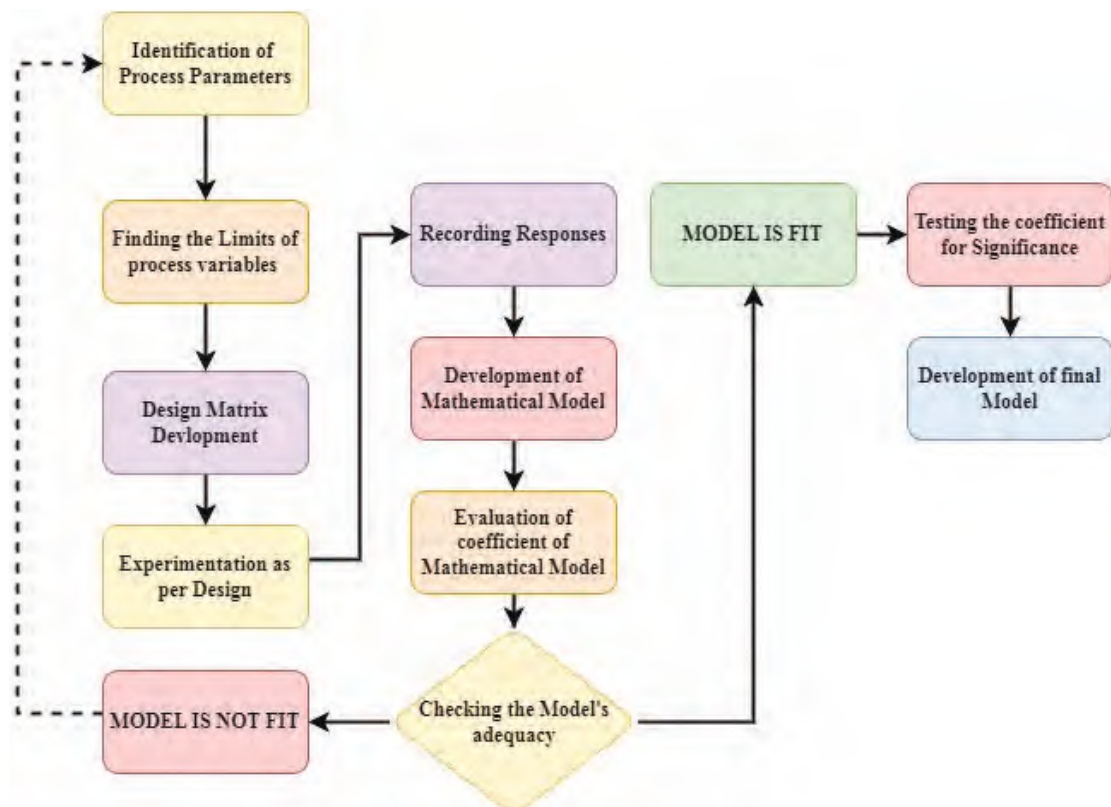


Figure 8. The flowchart for Response Surface Methodology (RSM) with Central Composite Design (CCD).

Table 5. Design of experiments (DoE), Central Composite Design (CCD) input parameters, and response

Independent variables	Notations	Factor type	Range	Coded value	Unit
Nozzle angle (degrees)	α	Numerical factor	90-95°	x_1	degrees
Nozzle diameter (mm)	d_j	Numerical factor	10-14	x_2	mm
Nozzle standoff distance (mm)	N_{sod}	Numerical factor	40-60	x_3	mm
Runner type	PTR, TTR	Categorical factor	----	x_4	----
No. of nozzles (n)	n	Categorical factor	1-2	x_5	----

Number of test runs required, $M = m_t(2^m + 2m + m_c)$ (4)

In the above equation, M is the number of required runs, and m_t is the number of categorical elements. The number of replications is denoted with m_c , while the number of numerical factors is m . So, for the present study, factorial design is 2^m composed of 3 numerical factors $2^3 = 8$; the axial section included $2m(2 \times 3 = 6)$ points, and 3 test replications as center points are considered. Hence, it yields 80 test runs as per the above calculations. A face-centered Central Composite Design (CCD) can be achieved by selecting Alpha = 1.0; each of the five levels at which each element in the research study operates can be altered. This method reduces the complexity of implementation to three tiers for each element. With this arrangement, a quadratic model may be estimated, with the linear (first-order) terms being addressed by the 2^m factorial points and the quadratic terms being accommodated by the $2m$ axial points. A complete factorial design provides a simple, organized approach that thoroughly assesses the primary effects and interactions of variables.

Model And Data Analysis (ANOVA)

Equation (5) shows a second-order (quadratic) model for n parameters. In this equation, β_0 denotes the constant term, β_i signifies the linear coefficient, β_{ii} represents the quadratic coefficient, β_{ij} indicates the interaction coefficient, and ε accounts for the experimental error.

$$Y = \beta_0 + \sum_{i=1}^n \beta_i X_i + \sum_{i=1}^n \beta_{ii} X_i^2 + \sum_{i=1}^n \sum_{j=i+1}^n \beta_{ij} X_i X_j + \varepsilon \quad (5)$$

The Design Expert 13.0 software is used in this study to estimate model coefficients and conduct a statistical analysis of the experimental data. The coefficients of the developed model were determined by this program using independent data and constant variance to distributions, one-way Analysis of Variance (ANOVA). The ANOVA method was employed to evaluate the appropriateness and adequacy of the responses concerning the model's functions and variables. The models that achieved statistical significance have a P-value below 0.05. The model's statistical importance was assessed using the signal-to-noise (F value), and its validity was confirmed using the coefficient of determination (R^2). The residuals and pure errors at duplicate locations were evaluated using a lack of fit test.

The formulas and processes used in the research were used to confirm the model's fit using the anticipated residual sum of squares, or PRESS.

The experiments were performed with the nozzle diameters of 10 mm, 12 mm, and 14 mm at 90° jet incidence (normal incidence), 92.5°, and 95° for both PTR and TTR. The RSM simulation–optimization method optimized the input parameters to maximize hydraulic efficiency. The CCD of RSM techniques developed a model with a standard deviation of 0.053. Regression analysis validated the model, confirming its reliability in determining input parameters for maximum hydraulic efficiency. The value of ANOVA is depicted in Table 7, corresponding to the test in Table 6, with different combinations of parameters studied under the present investigation, representing the quadratic model. The parameters considered for the present study, like nozzle angle (α), nozzle diameter (d_j), nozzle standoff distance (N_{sod}), PTR, TTR, and the number of nozzles (n), are found to affect the hydraulic efficiency (η_h) of turbine runners significantly, as experimentally.

The factor coding and the sum of squares are coded as type III – partial, respectively. The sum of squares provides the total for each variable entered last in the model. In other words, each variable's impact is assessed by considering all other variables. The significance of the model is indicated by its F-value of 90.49. An enormous F-value could only result from noise in 0.01% of cases. Significant model terms are indicated by P-values below 0.0500. Important model terms include A, B, C, D, E, AB, AC, BC, BD, CD, A^2 , and C^2 . If a model term's value is more than 0.1000, it is not meaningful. If the model has many superfluous terms (aside from those required to preserve hierarchy), model reduction may improve the model. The lack of fit, F-value of 0.97, indicates that the lack of fit is minimal compared to the pure error. The likelihood that noise is the cause of a significant lack of fit F-value is 54.54%. A non-significant lack of fit is preferred because the model must fit. According to Appendix A2, the proposed model in the present study is quadratic. Fitting a multivariate linear regression model with all independent variables included is the first step in the backward elimination process. The model is then refitted after the variable with the greatest p-value is eliminated. This iterative process is repeated until all of the model's remaining

Table 6. Design of Experiments (DoE) Central Composite Designs (CCD) matrix and results

Test Run (M)	Nozzle angle (α), degrees	Nozzle diameter (dj), mm	Nozzle standoff distance (Nsod), mm	C.F. 1 Runner type (PTR, TTR)	C.F. 2 Number of nozzles (n)	Results for hydraulic efficiency (η_h) Regression model		
						Exp.	Pred.	Residual
1	95	12	50	PTR	2n	0.6415	0.6383	0.0032
2	90	14	40	TTR	1n	0.6855	0.6903	-0.0048
3	90	14	60	PTR	2n	0.6755	0.6772	-0.0017
4	92.5	12	40	PTR	2n	0.6589	0.6612	-0.0023
5	92.5	12	50	TTR	2n	0.6735	0.6682	0.0053
6	90	12	50	PTR	2n	0.6795	0.6822	-0.0027
7	95	14	40	TTR	1n	0.6495	0.6458	0.0037
8	95	10	40	TTR	1n	0.6595	0.6624	-0.0029
9	92.5	10	50	TTR	1n	0.6535	0.6504	0.0031
10	90	10	60	TTR	1n	0.6695	0.6735	-0.004
11	90	10	40	TTR	2n	0.7165	0.7187	-0.0022
12	90	14	40	PTR	1n	0.6285	0.6283	0.0002
13	90	14	40	PTR	1n	0.6765	0.6711	0.0054
14	95	10	60	TTR	1n	0.6385	0.6362	0.0023
15	92.5	12	40	TTR	1n	0.6425	0.6378	0.0047
16	92.5	12	50	TTR	1n	0.6515	0.6466	0.0049
17	90	10	60	TTR	2n	0.7015	0.7006	0.0009
18	92.5	12	50	TTR	2n	0.6645	0.6682	-0.0037
19	92.5	12	50	PTR	2n	0.6525	0.649	0.0035
20	92.5	14	50	TTR	1n	0.6345	0.6372	-0.0027
21	92.5	10	50	PTR	1n	0.6315	0.6232	0.0083
22	92.5	10	50	PTR	2n	0.6435	0.6486	-0.0051
23	92.5	12	50	PTR	1n	0.6245	0.6264	-0.0019
24	90	10	40	PTR	2n	0.6815	0.6865	-0.005
25	90	14	60	TTR	2n	0.6905	0.6853	0.0052
26	95	12	50	TTR	2n	0.6545	0.6567	-0.0022
27	90	12	50	TTR	2n	0.7055	0.7024	0.0031
28	92.5	12	60	PTR	2n	0.6375	0.6455	-0.008
29	90	10	60	PTR	1n	0.6455	0.6505	-0.005
30	92.5	12	50	PTR	2n	0.6445	0.649	-0.0045
31	90	12	50	PTR	1n	0.6565	0.6572	-0.0007
32	92.5	12	50	PTR	1n	0.6215	0.6264	-0.0049
33	95	14	60	TTR	1n	0.6125	0.6117	0.0008
34	92.5	12	50	TTR	1n	0.6375	0.6466	-0.0091
35	92.5	12	50	PTR	2n	0.6585	0.649	0.0095
36	90	10	40	TTR	1n	0.6915	0.6921	-0.0006
37	95	12	50	TTR	1n	0.6365	0.6375	-0.001
38	95	14	40	PTR	2n	0.6415	0.6454	-0.0039
39	95	10	40	PTR	2n	0.6555	0.6539	0.0016
40	92.5	12	60	TTR	2n	0.6545	0.6597	-0.0052

Table 6. Design of Experiments (DoE) Central Composite Designs (CCD) matrix and results (continued)

Test Run (M)	Nozzle angle (α), degrees	Nozzle diameter (dj), mm	Nozzle standoff distance (Nsod), mm	C.F. 1 Runner type (PTR, TTR)	C.F. 2 Number of nozzles (n)	Results for hydraulic efficiency (η_h) Regression model		
						Exp.	Pred.	Residual
41	95	10	40	TTR	2n	0.6835	0.6842	-0.0007
42	92.5	12	50	PTR	2n	0.6505	0.649	0.0015
43	90	14	60	PTR	1n	0.6515	0.6547	-0.0032
44	95	10	60	TTR	2n	0.6615	0.6585	0.003
45	92.5	12	50	TTR	2n	0.6725	0.6682	0.0043
46	92.5	12	40	PTR	1n	0.6315	0.6388	-0.0073
47	92.5	12	50	PTR	1n	0.6325	0.6264	0.0061
48	92.5	12	50	PTR	1n	0.6315	0.6264	0.0051
49	95	10	60	PTR	1n	0.6125	0.615	-0.0025
50	95	14	60	TTR	2n	0.6235	0.6283	-0.0048
51	92.5	12	50	PTR	2n	0.6545	0.649	0.0055
52	92.5	14	50	TTR	2n	0.6515	0.656	-0.0045
53	95	14	40	TTR	2n	0.6655	0.6619	0.0036
54	92.5	12	50	TTR	2n	0.6615	0.6682	-0.0067
55	92.5	10	50	TTR	2n	0.6815	0.6748	0.0067
56	92.5	12	50	PTR	1n	0.6215	0.6264	-0.0049
57	92.5	12	50	PTR	2n	0.6515	0.649	0.0025
58	92.5	14	50	PTR	2n	0.6415	0.6437	-0.0022
59	90	10	60	PTR	2n	0.6835	0.6786	0.0049
60	92.5	12	50	TTR	1n	0.639	0.6466	-0.0076
61	95	14	60	PTR	1n	0.6025	0.6044	-0.0019
62	92.5	12	50	TTR	2n	0.6655	0.6682	-0.0027
63	90	14	40	TTR	2n	0.7125	0.7112	0.0013
64	92.5	12	40	TTR	1n	0.6715	0.6641	0.0074
65	90	10	40	PTR	1n	0.6655	0.6589	0.0066
66	95	10	60	PTR	2n	0.6345	0.6383	-0.0038
67	90	12	50	TTR	1n	0.6735	0.6784	-0.0049
68	95	10	40	PTR	1n	0.6255	0.6311	-0.0056
69	90	14	40	PTR	2n	0.6925	0.6929	-0.0004
70	92.5	12	50	PTR	1n	0.6315	0.6264	0.0051
71	92.5	12	40	TTR	2n	0.6915	0.6855	0.006
72	92.5	12	60	PTR	1n	0.6255	0.6226	0.0029
73	95	14	60	PTR	2n	0.6295	0.622	0.0075
74	92.5	12	50	TTR	1n	0.6435	0.6466	-0.0031
75	95	12	50	PTR	1n	0.6215	0.6181	0.0034
76	92.5	12	50	TTR	2n	0.6615	0.6682	-0.0067
77	92.5	12	50	TTR	1n	0.6515	0.6466	0.0049
78	92.5	14	50	PTR	1n	0.6185	0.6239	-0.0054
79	90	14	60	TTR	1n	0.6715	0.6638	0.0077
80	92.5	12	50	TTR	1n	0.6475	0.6466	0.0009

Table 7. Present work - ANOVA for response surface quadratic model for the hydraulic efficiency of runners

Source	SOS	df	M.S	F-value	p-value	Remarks
Model	0.0456	18	0.0025	90.49	< 0.0001	significant
A-Nozzle angle (α)	0.018	1	0.018	642.26	< 0.0001	significant
B-Nozzle diameter (d_j)	0.0008	1	0.0008	29.26	< 0.0001	significant
C-Nozzle SOD (N_{sod})	0.0044	1	0.0044	157.85	< 0.0001	significant
D-Runner type PTR, TTR	0.0078	1	0.0078	278.07	< 0.0001	significant
E-No. of nozzles (n) 1n,2n	0.0098	1	0.0098	349.69	< 0.0001	significant
AB- ($\alpha \times d_j$)	0.0004	1	0.0004	15.81	0.0002	---
AC- ($\alpha \times N_{sod}$)	0.0001	1	0.0001	4.15	0.0459	---
AD - ($\alpha \times PTT / TTR$)	8.10E-06	1	8.10E-06	0.2894	0.5926	---
AE- ($\alpha \times 1n / 2n$)	0.0001	1	0.0001	2.06	0.1565	---
BC- ($d_j \times N_{sod}$)	0.0001	1	0.0001	4.43	0.0394	---
BD- ($d_j \times PTT / TTR$)	0.0005	1	0.0005	17.26	0.0001	---
BE- ($d_j \times 1n / 2n$)	0.0001	1	0.0001	2.9	0.0936	---
CD- ($N_{sod} \times PTT / TTR$)	0.0003	1	0.0003	9.22	0.0035	---
CE- ($N_{sod} \times 1n / 2n$)	7.84E-07	1	7.84E-07	0.028	0.8676	---
DE- ($PTT / TTR \times 1n / 2n$)	4.95E-06	1	4.95E-06	0.1768	0.6756	---
A ² - (α^2)	0.0014	1	0.0014	50.33	< 0.0001	---
B ² - (d_j^2)	0.0001	1	0.0001	3.1	0.0834	---
C ² - (N_{sod}^2)	0.0002	1	0.0002	7.49	0.0081	---
Residual	0.0017	61	0	--	--	---
Lack of Fit	0.0011	41	0	0.9733	0.5454	Not significant
Pure Error	0.0006	20	0	--	--	---
Cor Total	0.0473	79	--	--	--	---

*1n= Single nozzle, 2n= Double nozzle, df = Degree of freedom, SoS= Sum of squares, MS= Mean square

variables show p-values below a given threshold—typically set at 0.05.

In RSM with CCD, normal probability plots and externally studentized residuals are essential for model diagnostics and validation Figure 9(a). Figure 9(b) compares the predicted values from the model to the actual observed values for model accuracy and validation, detecting outliers and identifying patterns like non-linearity. Additionally, the internally studentized, as shown in Figure 9(c), also describes the outliers and pattern; moreover, it checks the assumption of constant variance (homoscedasticity). Also, describe the heteroscedasticity of the spread of residuals with a change in run number. Figure 9(d) shows an incorrect functional form and missing variables in the systematic pattern in the residuals.

RESULTS AND DISCUSSIONS

Measurement of Hydraulic Efficiency (η_h)

The hydraulic performance was measured by measuring various parameters like runner speed (N), flow rate through nozzles(Q), dead weight (W), and spring balance

reading (S). The experimental work is carried out with varying flow conditions through the spear nozzle mechanism. An absorption-type rope brake dynamometer was used to measure the torque available on the shaft. The experiments were performed continuously for the various parameters, and different combinations of variables were considered for the present study. The results found with different configurations are given in terms of range in Table 5.

Comparison of PTR and TTR Based on Hydraulic Efficiency (η_h)

According to theoretical analysis, hydraulic efficiency (η_h) depends on parameters like flow rate, head, and fluid density. However, density remains constant for water as a fluid, so it merely depends on the remaining parameters. The runners are geometrically identical except for the bucket. The effect of independent parameters considered for the present study on the hydraulic efficiency of both runners is as follows.

Effect of Nozzle Angle (Jet Incidence Angle, α)

In the present study, the nozzle angle varied from 90 to 95 degrees in two steps of 2.5 degrees due to limitations. The normal jet incidence is desirable in the case of PTR. In

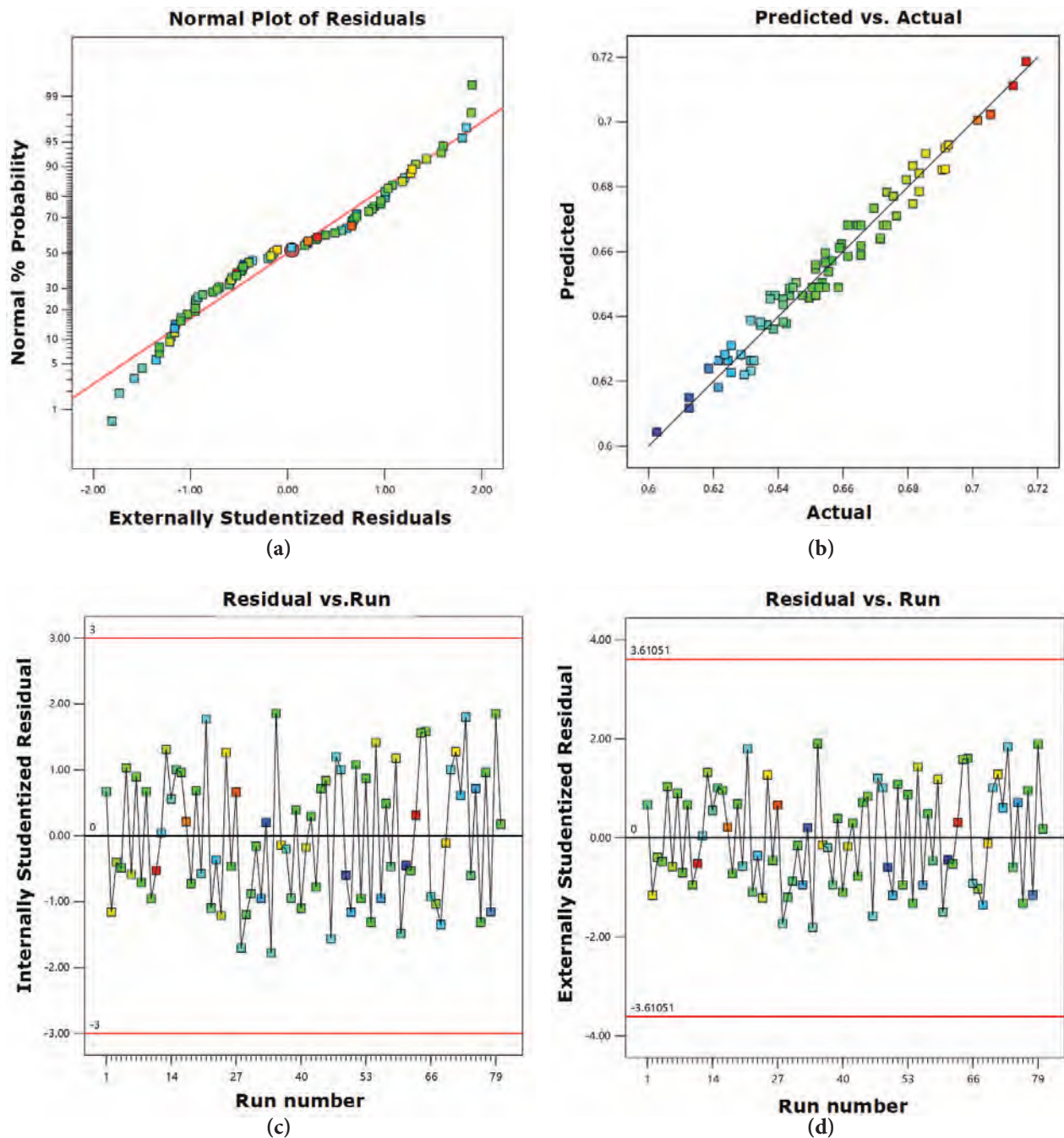


Figure 9. The graph for (a) Normal probability vs. Externally studentized residuals, (b) Predicted efficiency vs. Actual efficiency, (c) Internally studentized residual vs. Run number, and (d) Externally studentized residual vs. Run number.

contrast, TTR's hydraulic performance changes with nozzle angle variation. The larger the jet incidence angle variation from the design location, the greater the hydraulic efficiency (η_h) within a specific limit of jet incidence angle. For the present study, equations (6) to (9) revealed that the negative coefficient associated with nozzle angle tends to reduce the runner efficiency in both the runners considered here with either a single or double nozzle arrangement. The present

study describes the situations where the jet misalignment contributes less to torque generation.

Effect of Nozzle Diameter (Jet Diameter, d_j)

The jet of three different sizes, 10 mm, 12 mm, and 14 mm, equal to the nozzle diameter, strikes the runner bucket in different inclinations and distances, yielding different torques on the vertical shaft attached to the runner. With an increase in jet diameter, the fluid flow rate also increases.

Conversely, the jet is more focused with a smaller diameter and more diffusion with surroundings found with a 14 mm diameter. The hydraulic efficiency (η_h) increases with a decrease in nozzle diameter (d_j) in the case of PTR with a single nozzle (1n). It is found more with a double nozzle (2n), keeping the nozzle standoff distance (N_{sod}) constant but a low value of nozzle angle (α). For TTR, compared to PTR, the hydraulic efficiency (η_h) is found to be higher with low inlet angle (design point) and nozzle diameter (d_j). From equations (6) and (7), the nozzle diameter (d_j) is found to have a positive coefficient in the case of PTR for both single and double nozzle operations. Still, its contribution is less due to the minor terms involved than the nozzle angle (α). From equations (8) and (9), the nozzle diameter (d_j) is found to have a negative coefficient in the case of TTR for both single and double nozzle operations, with less effect on hydraulic performance than the minor terms involved, such as nozzle angle (α).

Effect of Nozzle Standoff Distance (N_{sod})

For an impulse turbine, the nozzle tip distance from the runner is crucial for jet incidence. Typically, this distance is maintained between $4d_j$ and $6d_j$ to prevent flow separation at the jet surface. It must be optimized to avoid interference with water from the bucket. In this study, the nozzle standoff distance (N_{sod}) was varied between 40 and 60 mm in 10 mm increments. Quadratic equations (6) to (9) indicate that a 40 mm standoff distance is more effective, as it has a negative coefficient for both runners investigated. However, standoff distance (N_{sod}) is more negatively dominant for TTR than for PTR. The standoff distance is kept constant for the further evaluation of other parameters as attributed by RSM.

The Response Surface Methodology (RSM) with the CCD model was adopted to systematically investigate the influence of independent variables on hydraulic efficiency (η_h). This study examined three critical numerical parameters: the angle of the jet ($\approx$ nozzle angle) (A), the diameter of the jet ($\approx$ nozzle diameter) (B), and nozzle standoff distance (C) alongside a two categorical parameter turbine runner type (D) and number of nozzles (E), designated as independent variables. The output hydraulic efficiency (Y) was selected as the dependent variable (response).

The experimental design includes two categorical and three numerical components. To ensure the readings were repeatable, 80 experimental tests were conducted thrice, with eight test points and three replications for each category. Table 6 lists the experimental design elements and related experimental responses for the constructed model. An adequate regression model was constructed by analyzing the experimental data. An Analysis of Variance (ANOVA) was performed, a response surface model was constructed, and the Central Composite Design (CCD) responses were evaluated using “Design Expert 13.0” software. According to the different turbine types, the software determined the

most important models, as explained below: Pelton Turbine Runner (PTR)-Single nozzle(1n)

$$\begin{aligned}\eta_h = & +0.626362 - 0.019550 \times \alpha + 0.000375 \times d_j \\ & - 0.008110 \times N_{sod} - 0.003719 \times \alpha \times d_j \\ & - 0.001906 \times \alpha \times N_{sod} - 0.001969 \times d_j \times N_{sod} \\ & + 0.011317 \times \alpha^2 - 0.002808 \times d_j^2 + 0.004367 \times N_{sod}^2\end{aligned}\quad (6)$$

Pelton Turbine Runner (PTR)-Double nozzle(2n)

$$\begin{aligned}\eta_h = & +0.648982 - 0.021950 \times \alpha + 0.002475 \times d_j \\ & - 0.007830 \times N_{sod} - 0.003719 \times \alpha \times d_j \\ & - 0.001906 \times \alpha \times N_{sod} - 0.001969 \times d_j \times N_{sod} \\ & + 0.011317 \times \alpha^2 - 0.002808 \times d_j^2 + 0.004367 \times N_{sod}^2\end{aligned}\quad (7)$$

Turgo Turbine Runner (TTR)-Single nozzle(1n)

$$\begin{aligned}\eta_h = & +0.646587 - 0.020450 \times \alpha - 0.006575 \times d_j \\ & - 0.013190 \times N_{sod} - 0.003719 \times \alpha \times d_j \\ & - 0.001906 \times \alpha \times N_{sod} - 0.001969 \times d_j \times N_{sod} \\ & + 0.011317 \times \alpha^2 - 0.002808 \times d_j^2 + 0.004367 \times N_{sod}^2\end{aligned}\quad (8)$$

Turgo Turbine Runner (PTR)-Double nozzle(2n)

$$\begin{aligned}\eta_h = & +0.668212 - 0.022850 \times \alpha - 0.009425 \times d_j \\ & - 0.012910 \times N_{sod} - 0.003719 \times \alpha \times d_j \\ & - 0.001906 \times \alpha \times N_{sod} - 0.001969 \times d_j \times N_{sod} \\ & + 0.011317 \times \alpha^2 - 0.002808 \times d_j^2 + 0.004367 \times N_{sod}^2\end{aligned}\quad (9)$$

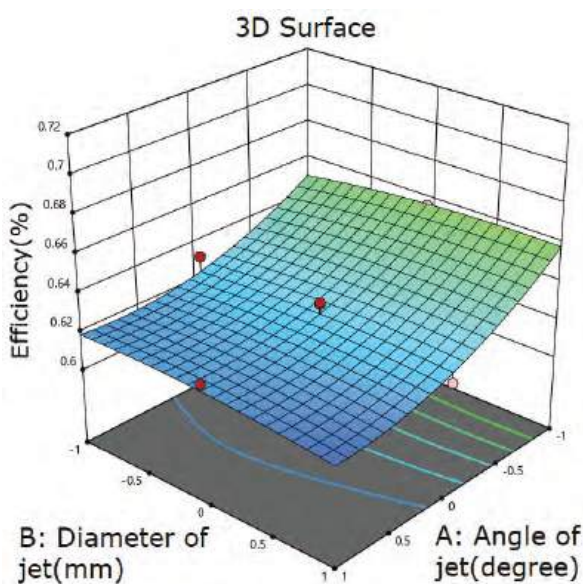
A and B represent the current study's coded values for nozzle angle (α) and diameter (d_j). AB denotes the interaction between these parameters, while A^2 and B^2 are the squared terms corresponding to the studied parameters. In the equations derived from the analysis, a positive coefficient indicates an increasing effect of the respective factor on the hydraulic efficiency (response) (η_h). In contrast, a negative coefficient signifies a decreasing impact. Table 7 represents the results of the ANOVA alongside the relevant statistical parameters for hydraulic efficiency (η_h).

The experimental data align well with the model's predictions, particularly when the model exhibits significant regression and an insignificant lack of fit. A strong F-value of 90.49 denotes that the model has a substantial statistical impact on the response variable. Its reliability is further reinforced by the p-value, which indicates only a 0.01% likelihood that such a high F-value could be attributed to random noise. The low error probability confirms that the proposed model effectively captures the experimental data. P-values below 0.05 indicate the significance of model

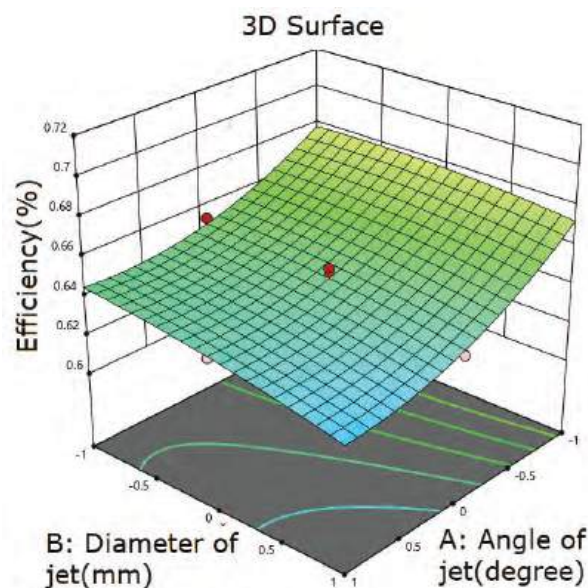
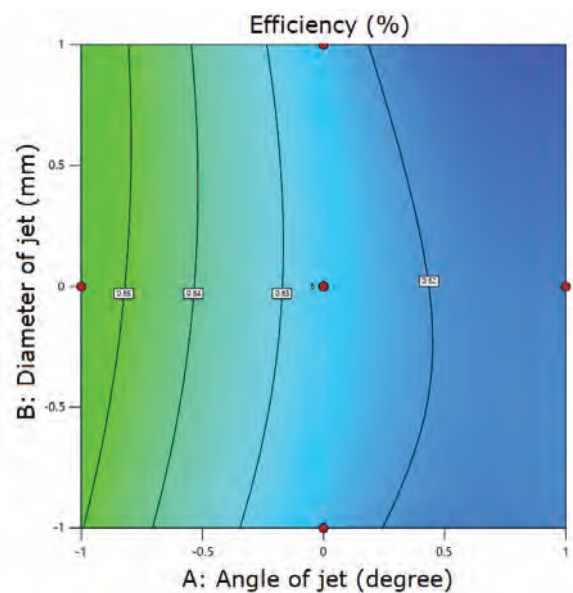
terms, while values under 0.001 indicate high importance for certain variables. Conversely, p-values exceeding 0.1 suggest that neither the model nor its components carry statistical significance. Furthermore, the model demonstrates minimal lack of fit concerning pure error, as evidenced by the lack of fit F-value of 0.9733. It is a positive outcome since the lack of fit is insignificant, indicating that the model aligns well with the predicted data.

The R^2 value, or the coefficient of determination, measures the dataset variance corresponding to the built model. In particular, R^2 is the percentage of the dependent variable's (response) variance that can be accounted for by the independent variables (predictors). An intense match between the model and the data is indicated by an R^2 value close to

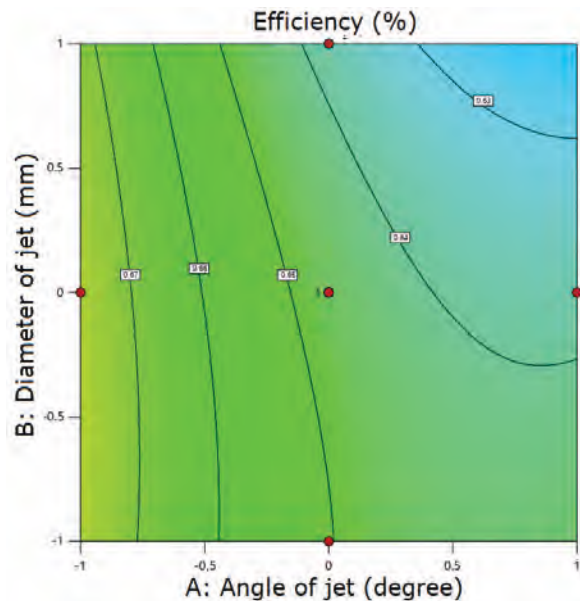
1. According to Appendix A3, R^2 values greater than 0.95 indicate the model matched the data quite well, with only 0.14% of the data being unexpected. The created model's ability to predict the runners' hydraulic performance across a variety of independent variables is demonstrated by its high adjusted R^2 value. There is less than a 0.2 discrepancy between the adjusted and the predicted R^2 having a value of 0.9532 and 0.9375, respectively. With an acceptable threshold of > 4 , the correlation's suitable precision of 44.3384 indicates that the noise ratio of the model is within an acceptable range, confirming its usefulness for examining the design space. Additionally, the comparatively low coefficient of variation (0.8090) and standard deviation (0.0053) show better model validity and accuracy.



(a)



(b)



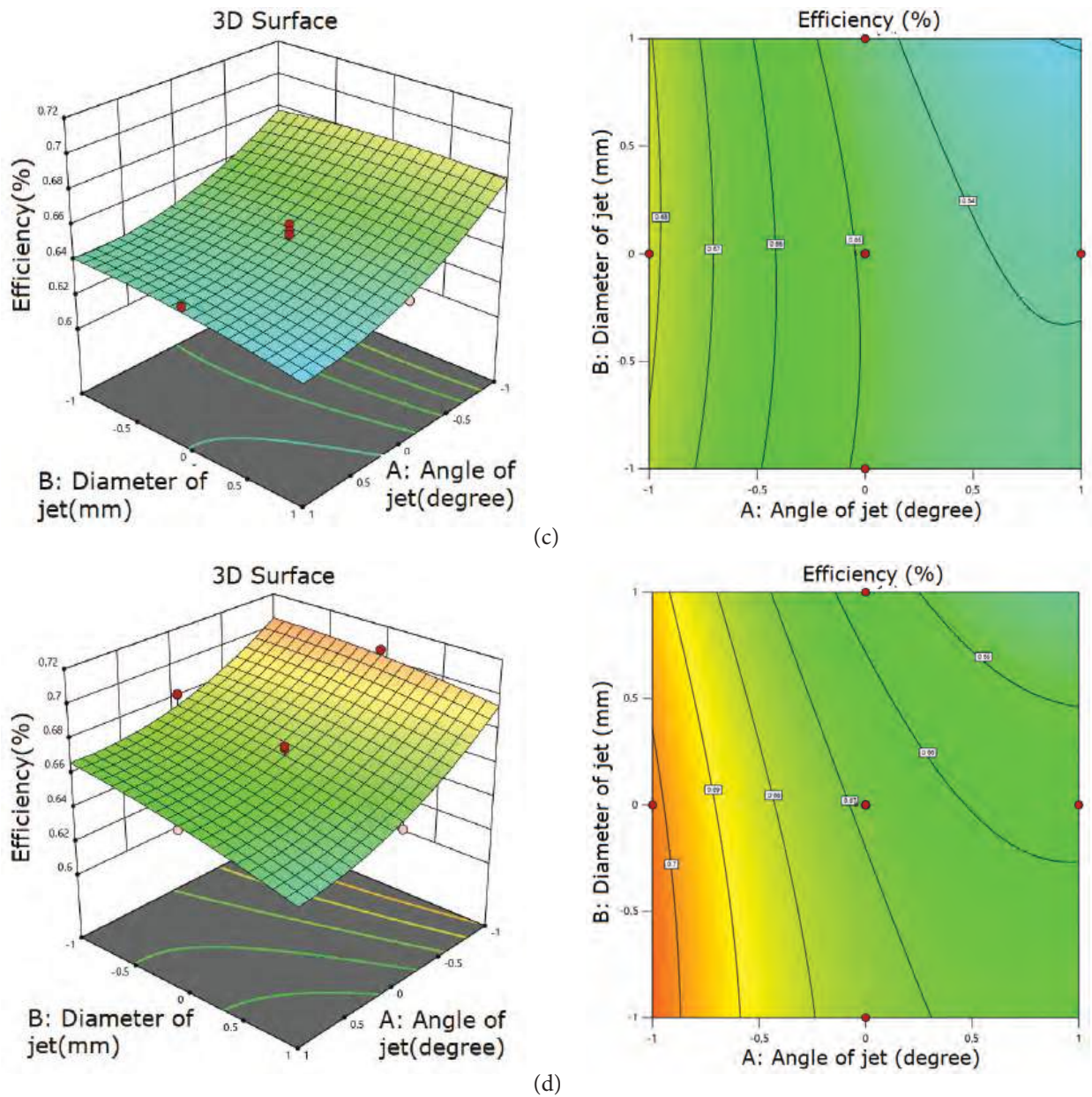


Figure 10. The response 3D surface and contour plots of hydraulic efficiency as a function of nozzle angle (α), nozzle diameter (d_j), and nozzle standoff distance (N_{sod}) in different turbine rotor types and the number of nozzles (n) of (a) PTR-1n, (d) TTR-1n, (c) PTR-2n, and (d) TTR-2n.

Table 8. Optimum values for input parameters and corresponding response

Sr. No.	Runner type	Number of nozzles (n)	Nozzle angle (α), degrees	Nozzle diameter (d_j), mm	Standoff distance (N_{sod}), mm	Response: Hydraulic efficiency (η_h)	Goal
1	Turgo Turbine Runner (TTR)	2n	95°	14	40	66.19%	Maximization

As a result, the findings of the ANOVA analysis confirm that the developed model fits the experimental data, validating its applicability for predicting the hydraulic efficiency

(η_h) of PTR and TTR with a variable number of nozzles. The evaluation of the correlational relationship between the design parameters, nozzle angle(α) and diameter(d_j) versus

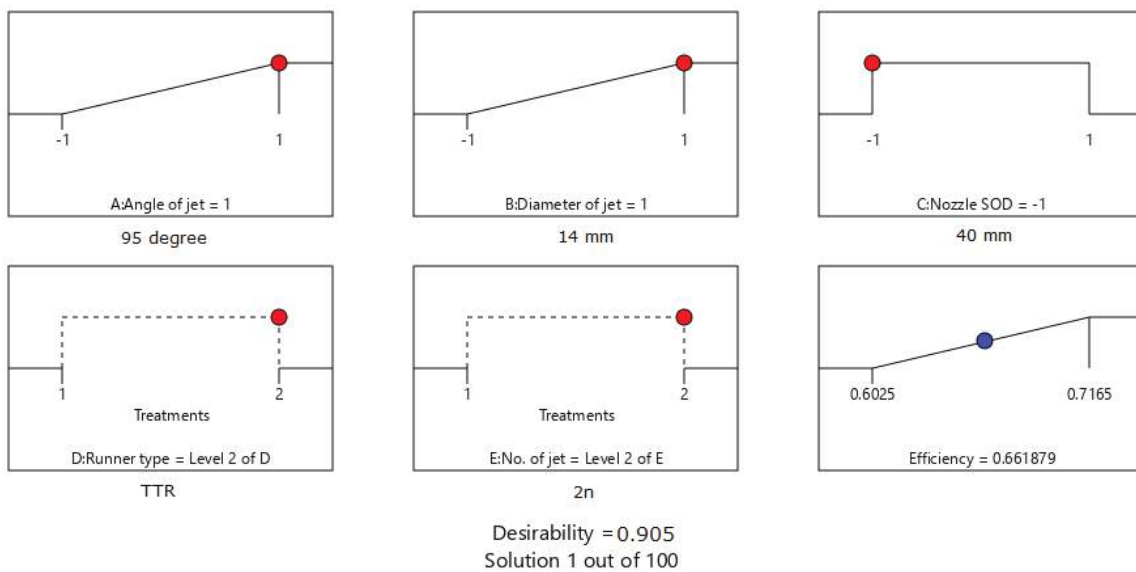


Figure 11. Desirability solution ramp for optimization of parameters for maximum hydraulic efficiency.

Table 9. Summary for comparison of present and similar past work

Sr. No.	Author/s	Parameters considered	Type of work carried out	Maximum hydraulic efficiency (η_h) _{max}
1	Alomar et al., 2022	PTR, Maximum head = 110 m, Runner diameter = 275 mm, nozzle diameter = 9.5 to 12.5 mm (4 nos. in equal steps), number of nozzles = 1	Experimental	35.5-21.6%
2	Syofii et al., 2022	PTR, Head = 3 m, Runner diameter = 275 mm, nozzle diameter = 8, 9, 10 mm, Number of nozzles = 1	Experimental	9.01-25.44%
3	Elgammi and Hamad, 2022	PTR, Head = 2.5 m, Runner pitch circle diameter = 123 mm, nozzle diameter = 9 mm, Number of nozzles = 8	Experimental	10%
4	Gallego et al., 2021	TTR, Head = 42 m, Runner diameter = 100 mm, nozzle diameter = 10, 15, 20 mm, Number of nozzles = 2	Experimental & RSM with BBD model	93.7%
5	Oyebode, 2020	PTR, Head = 4 m, Runner diameter = 260 mm, nozzle diameter = 51 mm, Number of nozzles = 1	Experimental & statistical	-----
6	Hlabanelo et al., 2020	PTR, Head = 10 m, Runner diameter = 220 mm, nozzle diameter = 21 mm, Number of nozzles = 2	Experimental and numerical	5-64%
7	Nigussie et al., 2017	PTR, Head = 47.5 m, Runner pitch circle diameter = 500 mm, nozzle diameter = 38.50 mm, Number of nozzles = 4	Numerical	78.8%
8	Warjito et al., 2017	TTR, Head = 2.7 m, Runner diameter = 120 mm, nozzle diameter = 63.9 mm, Number of nozzles = 1	Numerical	85.97%
9	Gaiser et al., 2016	TTR, Head = 35 m, Runner pitch circle diameter = 130 mm, nozzle diameter = 7.125, 12.85 and 18.59 mm, Number of nozzles = 1	Experimental & RSM with CCD model	43.7-63.8%
10	Cobb and Sharp, 2013	PTR, Runner pitch circle diameter = 100 mm, TTR, Runner pitch circle diameter = 131 & 169 mm, Head = 17-25 m & 18-28 m, nozzle diameter = 7.94 to 12.70 mm, Number of nozzles = 1	Experimental	PTR - 73% TTR - 81%-85%
11	Williamson et al., 2013	TTR, head = 0.5 – 3.5 m, runner pitch circle diameter = 150 mm, nozzle diameter = 15-35 mm (5 nos. in equal steps)	Experimental and Design of Experiments (DOE)	87%-91%
12	Present study	PTR, TTR, Head = 10 m, Runner pitch circle diameter = 100 mm, nozzle diameter = 10, 12 and 14 mm, Number of nozzles = 2	Experimental & RSM with CCD model	For optimized conditions, Table 7

Table 10. Comparison of hydraulic efficiency η_h between experimental and RSM results for optimum values of parameters

Runner type	Number of nozzles (n)	Hydraulic efficiency η_h , %		
		Experimental results	RSM results	Error(%)
Turgo turbine runner (TTR)	2n	0.6655	0.6619	0.54

the hydraulic efficiency (η_h) for PTR and TTR is illustrated in Figure 10, utilizing contour and 3D plots generated through Response Surface Methodology (RSM). Figure 10 depicts these visualizations for the hydraulic efficiency (η_h) (response) as a function of nozzle diameter (mm), nozzle angle (α), and the nozzle standoff distance (N_{sod}), with variations in turbine runner types and also with single or two nozzle arrangements. From the 3D plots with the response contour of the hydraulic performance in Figure 10 (a), it was observed that the nozzle angle(α) significantly influences the hydraulic efficiency (η_h) for the PTR, with both low and medium values of the nozzle angle(α) yielding maximized hydraulic efficiency (η_h). Positioning a nozzle angle(α) with reference to the bucket allows the generation of adequate torque and non-interference for the bucket flow to recover before reaching the exit. Consequently, following a decrease in the hydraulic efficiency (η_h) with the high standoff distance (N_{sod}) and nozzle diameter(dj), an increase is noted in the low range of standoff distance and nozzle diameter. Figure 10 (b) presents the contour and 3D plots of the hydraulic efficiency (η_h) for the TTR single-nozzle operation. In contrast to the previous scenario, the nozzle angle(α) and nozzle diameter(dj) are more dominant than the standoff distance (N_{sod}). With an increase in the nozzle angle(α), the efficiency decreases and increases with the nozzle diameter(dj).

Figure 10 (c) illustrates the contour and 3D surface plots of the hydraulic efficiency (η_h) for PTR with double nozzles tangentially arranged at the PCD of runners. The 3D surface pattern resembles a curve for the PTR turbine type; however, this configuration harvested and converted more hydraulic Energy than the TTR single-nozzle operation. The similarities between the response surfaces for the TTR and PTR indicate a comparable effect of nozzle positioning, nozzle size, and tip distance of a nozzle under single and double nozzle on the hydraulic performance of the runner. The contour plot and three-dimensional response surface illustrating the hydraulic efficiency (η_h) for the TTR with double nozzle (2n) are presented in Figure 10 (d). This visualization demonstrates that the amount of Energy extracted efficiently by the runner is superior to that of alternative turbine runner types, with a notable finding that reducing the nozzle standoff distance (N_{sod}) leads to increased hydraulic efficiency (η_h). The contour plots and 3D response surfaces shown in Figure 10(a, b, c, and d) further emphasize the significant correlation between nozzle angle(α) and nozzle diameter (dj).

Moreover, the interaction analysis between nozzle standoff distance (N_{sod}) and hydraulic efficiency (η_h) also indicates that variations in the number of nozzles(n) notably influence hydraulic efficiency (η_h). These findings suggest that alterations in the nozzle angle(α) exert a more pronounced impact on the hydraulic efficiency (η_h) of TTR than changes in nozzle diameter(dj) and nozzle standoff distance (N_{sod}), with lower hydraulic efficiency (η_h) achieved at lower nozzle diameter(dj), nozzle standoff distance (N_{sod}) and single nozzle (1n). Overall, by optimizing the nozzle angle(α), nozzle diameter(dj), nozzle standoff distance (N_{sod}), and the number of nozzles(n) for the two runners, it is feasible to enhance the hydraulic efficiency (η_h) of the PTR by up to 10% and TTR by up to 13%.

The study utilizing Response Surface Methodology (RSM) yielded several key findings regarding the influence of independent variables on the output, which is considered the hydraulic efficiency (η_h) of the turbine runner for Pico hydro application. It was determined that nozzle angle (A), nozzle diameter (B), nozzle standoff distance (C), and the number of nozzles (E) significantly affect the hydraulic efficiency (Y) of both the runners (D), with specific combinations of these variables identified as optimal for maximizing efficiency. The regression model developed from the experimental data demonstrated a robust fit and predictive capability. It was validated through ANOVA, which highlighted the statistical significance of both the types of factors and their interactions. The response surface plots provided a visual representation of these interactions, indicating how variations in one parameter can influence hydraulic efficiency in conjunction with others. Furthermore, the findings have practical implications for turbine operation and design, suggesting that strategic adjustments to nozzle angle(α), nozzle diameter(dj), nozzle standoff distance (N_{sod}), and the number of nozzles (n), alongside careful consideration of turbine runner type, can enhance performance. Lastly, the study recommends avenues for future research, including further exploring additional variables like debris and fluid density, as it is to be used for Pico hydro off-grid applications and turbine configurations to refine the understanding of their impacts on hydraulic performance.

The TTR has a half bucket compared to the PTR and a much better tendency to accommodate a higher flow rate with a half bucket width. The speed of TTR was found to be higher compared to PTR. One of the objectives was to find the optimal hydraulic efficiency as a response to the

allied design parameters. The values found in the current study for the range of parameters investigated are shown in Figure 11 for the quadratic model with red and blue balls on the graph. It has been observed that the optimal values of the numerical parameters, nozzle angle(α), nozzle diameter(d_j), and nozzle standoff distance (N_{sod}), are within the range and around the model's highest expectation.. The condition of optimality for TTR is depicted in Table 8 below. The solution is found with the objective of maximum hydraulic efficiency under given head and variable flow rate conditions. For validating the model, desirability is close to 1, and in the present study, it is found to be about 0.905, which shows good agreement between output and the desired output.

The optimum value of hydraulic efficiency achieved to maximize the hydraulic efficiency was found when the double nozzle configuration had a 40 mm nozzle standoff distance, nozzle angle, and nozzle diameter of 95° and 14, respectively.

Optimum Conditions for Maximum Hydraulic Efficiency (η_h)_{max}

The main objective of this investigation is to identify the maximum hydraulic efficiency by determining the operating and geometrical parameters that facilitate the most efficient operation of the system. The designed parameters and the required model responses for the optimization process are comprehensively detailed in Table 8. This table specifies that the response hydraulic efficiency (η_h) is integral to the optimization efforts. The input parameters

were constrained within predetermined ranges to maximize the identified response. Under these conditions, the TTR achieved the maximum hydraulic efficiency of 66.19% (as illustrated in Table 8 and Figure 11). For TTR, it corresponds to a 95° nozzle angle, 14 mm nozzle diameter, a 40 mm standoff distance, and the PTR turbine type with double nozzle (2n). The experiment, RSM-based input parameter optimization work, and the corresponding responses in hydraulic efficiency (η_h) carried out by various researchers are highlighted in Table 9 for comparison. The variation in maximum hydraulic efficiency (η_h)_{max} in the last column is a relative term, as the variation in geometrical and operating parameters of the turbine runner is considered for research.

The current research found that the hydraulic efficiency (η_h) for TTR was significantly higher than previously reported findings (Sr No.6, Table 9). It is also noted that most of the past work tabulated is done on horizontal axis turbine runners, unlike the vertical axis considered in the present study. This research shows that jet diameter, angle, and nozzle standoff distance significantly improve impulse turbine hydraulic efficiency (η_h). On the other hand, for Pico hydro applications, the RSM optimization method can enhance the vertical axis TTR's performance. When considering the PTR double nozzle configuration, the TTR configuration yielded a higher gain in hydraulic efficiency (η_h). However, there isn't a recognized way to compare the abilities of various hydroturbine runners. This study developed an RSM model for impulse turbine runners to achieve these objectives.

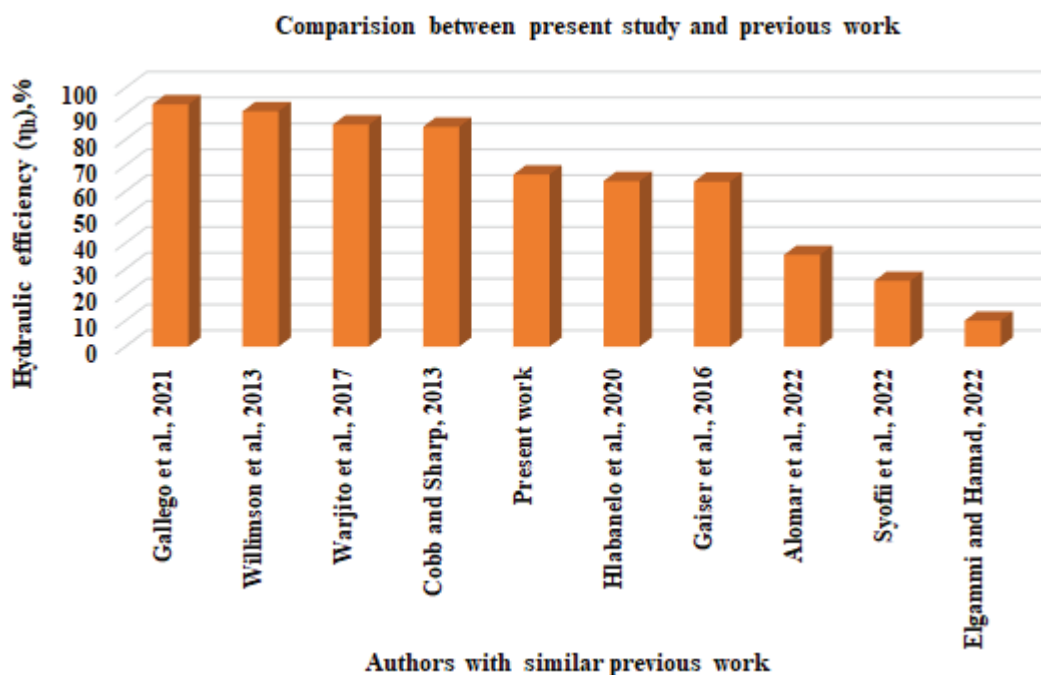
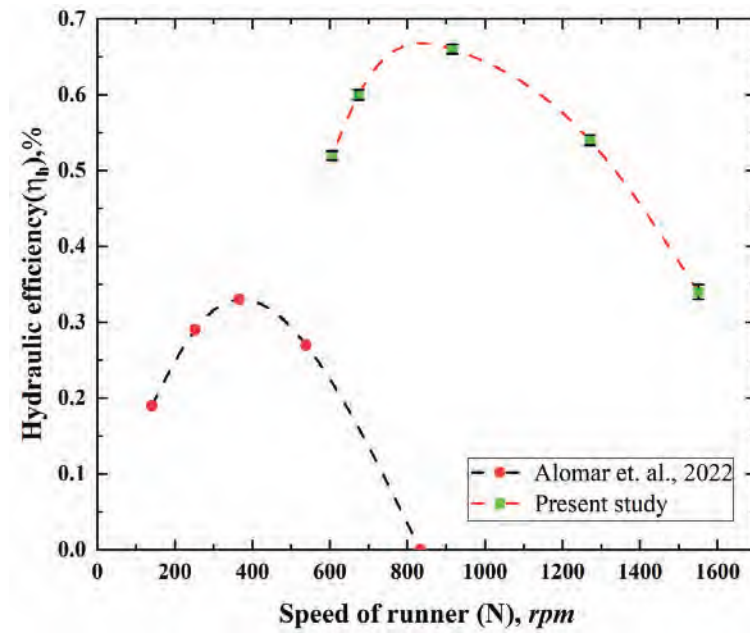


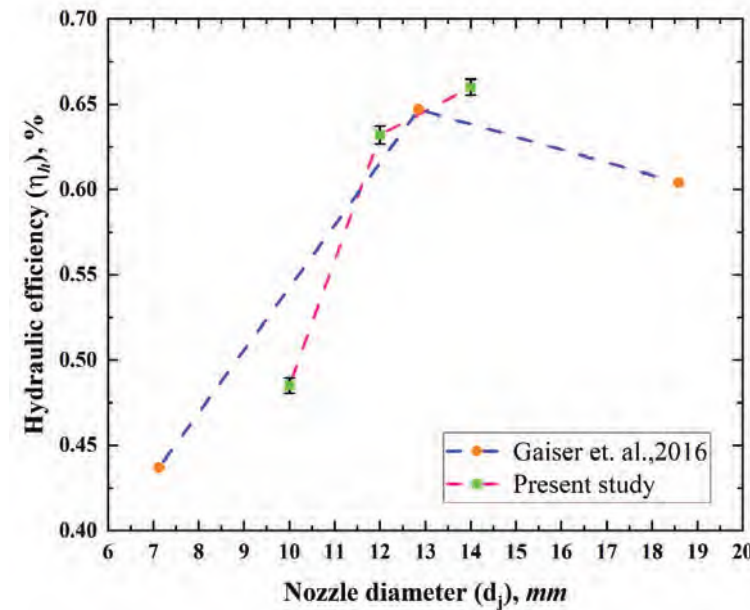
Figure 12. Comparison between similar previous work and the present study for maximum hydraulic efficiency.

The RSM model's output is compared with experimental data to validate the ideal conditions for the anticipated parameters, as Table 10 illustrates. Excellent agreement between the experimental data and the suggested RSM model is demonstrated by the results of the confirmation test, which show a slight discrepancy (0.54%) between the predicted and real hydraulic efficiency (η_h). The results of the present investigations have been compared with

similar previous work on a bar chart for maximum hydraulic efficiency. Figure 12 shows that the present study yielded hydraulic efficiency under optimized conditions under a head of 10 m compared to a lower runner PCD of 100 mm in earlier studies. The comparison revealed higher efficiency bars in the previous work due to increased head, number of nozzles, nozzle diameter, runner type, or runner diameter.



(a)



(b)

Figure 13. Turbine runner with a hydraulic performance comparison between similar previous and the present work findings (a) based on runner speed (b) based on nozzle diameter

The hydraulic performance of the PTR in the present investigation has been compared with similar previous work carried out by Alomar et al. in 2022, which had a 110 m head and 10.5 mm nozzle diameter. The trendline for the present work has been shown with error bars for the 10 mm nozzle diameter, 40 mm nozzle standoff distance, and double nozzle arrangement in Figure 13 (a). The trend lines match, and it has been found that the 65% hydraulic efficiency in the present study is higher than 33.3% of the parameters studied compared to those reported by authors in their previous work. The nozzle diameters are also checked for the findings obtained in the present study, as depicted in Figure 13 (b) in the case of TTR. The TTR in the present study is compared with the TTR for a similar trend reported by Gaiser et al. in 2016, with a head of 35 m, a runner pitch circle diameter of 130 mm, and a single nozzle. The highest efficiency observed in their study was 64.7% with a 12.85 mm nozzle.

In their work, the head, runner diameter, and flow rate are more than those investigated in the current study. However, the current curve with 100 mm PCD, 14 mm nozzle diameter, and 10 m head follows the same trendline. So, the present study has the least value of parameters, with the increased value of the hydraulic performance for the runners investigated. Thus, it suggests improved hydraulic efficiency with optimized parameters.

It has been found that the vertical axis turbines investigated in the present study are more suitable for low head conditions for the given set of design and operating parameters considered. It indicates that the present study is suitable for installing off-grid Pico hydro applications for power generation in remote areas.

CONCLUSIONS

This study aimed to assess the performance of developed impulse turbines (PTR and TTR) running at optimal efficiency levels using response surface methodology (RSM) with central composite design (CCD). The experimental test was conducted to optimize the range of operating parameters, viz. jet incidence angle (90° – 95°), nozzle diameter (10–14 mm), and number of nozzles (1 or 2) at a standoff distance (40–60 mm) under low-head operation (10 m) for the range of global parameters. The hydraulic efficiency assessment of the Pelton and Turgo turbines was compared at optimum conditions, and the following conclusions were made.

- It has been found from the literature studied and cited here that multiple research studies have been conducted in the past with horizontal axis low-head Turgo and Pelton turbines, and the vertical axis with jet deviation (α) is sidelined. Also, the effect of nozzle standoff distance (N_{sod}), which affects hydraulic efficiency (η_h) due to jet expansion, has not been identified or studied.
- The Polylactic Acid (PLA) for 3D printed buckets with aluminum runner discs was a low-cost solution compared to metallic bucket material for power generation at low head.
- With a standoff distance of more than $4d_j$, the jet deviates due to diffusion in the surrounding environment, resulting in low kinetic Energy on the buckets. As the jet diameter rises, the turbine's total speed lowers. So, the proposed model found that a minimum standoff distance of approximately 40 mm was maintained for each nozzle diameter to reduce the expansion of the jet. Also, it was observed with the Pelton turbine that the significant impact on efficiency was caused by jet misalignment on the turbine and vibrational and frictional losses, reducing the overall efficiency and power developed by the system.
- Compared to the Pelton turbine runner (PTR), the ability to handle the high flow rate of the Turgo turbine has shown increased hydraulic efficiency (η_h) with an increase in the nozzle diameter. A significant improvement in hydraulic efficiency (η_h) was achieved by increasing the overall speed ratio of the turbine under double-nozzle (2n) operation compared to single-nozzle (1n) operation. The maximum power developed from a 10 mm jet diameter, making the jet ratio 11.11 in the Pelton turbine runner (PTR) case.
- The nozzle angle (α) was crucial for testing the runners' hydraulic performance. The normal incidence with a 90° nozzle angle was found more promising compared to deviated other jet angles, 92.5° and 95° in the case of PTR, and TTR is more efficient with a 95° nozzle angle as the jet incidence becomes smoother.
- RSM with central composite design (CCD) effectively optimizes TTR and PTR performance, reducing the need for repeated experiments and facilitating quick design decisions.
- The quadratic model developed by the RSM for the evaluation of hydraulic efficiency (η_h) and the most efficient runner was found to apply to a high degree of accuracy to quantify the hydraulic performance of the runners. The P-value for all the model parameters, such as nozzle angle, nozzle diameter, and number of nozzles at a standoff distance, was below 0.0001, indicating significant effects on the model and, ultimately, the turbine performance. A thorough statistical analysis of the model was conducted to verify its accuracy and validity.
- The optimized conditions for a vertically installed TTR included a 95° nozzle angle, 14 mm nozzle diameter, 40 mm distance from the nozzle tip to the bucket, and double nozzle operation, achieving 66.19% efficiency. The present study revealed that the TTR runner was more sensitive to changes in nozzle angle than the PTR.
- The optimality conditions of the developed RSM model are compared with the results of the experimental investigation to validate the design of geometric parameters. From the validation test, it is found that the error is minimal, which shows that only 0.54 % of actual and predicted values are compared and found with a degree

of agreement. Thus, the validation process has verified the ultimate goal of the study.

- The hydraulic efficiency (η_h) achieved under optimized conditions in the present study is compared with past work, showing good efficiency improvement. The curve follows the trend line up to a 12 mm nozzle diameter, but with a further 2 mm stepped increase in nozzle diameter, the efficiency improves in the case of TTR with a 40 mm nozzle standoff distance.

HIGHLIGHTS

- A Turgo and Pelton turbine Runner with 100 mm PCD in a vertical axis configuration was designed and experimented with.
- The variation in flow, Nozzle diameter (jet diameter, d_j), number of nozzles (n), nozzle angle (jet deviation, α), and nozzle standoff distance (N_{sod}) were kept as independent variables, keeping the head constant.
- The Response Surface Methodology (RSM) with a Central Composite Design (CCD) model was used for the optimization and has shown the effect of each parameter on hydraulic performance.

ABBREVIATION

ANOVA	Analysis of variance
CFD	Computational Fluid Dynamics
RSM	Response Surface Methodology
CCD	Central Composite Design
PCD	Pitch Circle Diameter, mm
PTR	Pelton Turbine Runner
TTR	Turgo Turbine Runner
PLA	Polylactic Acid
1n	Single nozzle
2n	Double nozzle
RANS	Reynolds Averaged Navier-Stokes

Nomenclature

Q	Volumetric flow rate/Discharge, m ³ /sec
D_o	The inlet diameter of the nozzle or the inside diameter of the pipe, mm
d_j	Nozzle(jet) diameter, mm
N	Runner speed, rpm
H	Available head, m
N_s	Specific speed
g	Gravitational constant, m/s ²
P_u	Unit power
P_d	Power developed, Watt
P_h	Hydraulic power, Watt
V_{jet}	Velocity of the jet, m/s
u	Peripheral velocity of runners, m/s
T_d	Torque developed at runner shaft, N-mm.
N_{sod}	Nozzle standoff distance, mm
W	Dead weight, kg
S	Spring balance reading, kg

Greek Symbols

ρ	Density of the water, kg/m ³
η_v	Volumetric efficiency, %
α	Nozzle angle, degrees
η_m	Mechanical efficiency, %
η_h	Hydraulic efficiency, %
ϕ_s	Speed ratio/ Velocity ratio
ω	Angular speed, rad/sec
β_2	The jet angle at the bucket exit, degrees
λ_s	Speed or Velocity ratio
ψ_j	Jet ratio

Subscripts

h	Refers to hydraulic
s	Refers to speed
m	Refers to mechanical
2	Refers to exit
j	Refers to jet
sod	Refers to the standoff distance
v	Refers to volumetric
d	Refers to the developed

FUTURE SCOPE OF WORK

The work can be expanded to examine the impact of bucket spacing on performance for runners with a variable bucket number and a greater runner diameter. The bucket's structural analysis can also be studied numerically under dynamic fluid action. The effect of different sustainable materials (compatible with 3-D printing technology) on hydraulic performance can also be explored for further investigation.

ACKNOWLEDGMENT

Thanks to my beloved students, Gaurav P. Bakhru and Darshan N. Modi, for supporting the experimentation and providing help for developing test setup and 3D printed turbine runners at Advance Fluid Mechanics and Fluid Power Laboratory, Sardar Vallabhbhai National Institute of Technology (SV NIT), Surat, Gujarat, India

DISCLOSURE STATEMENT

The authors gratefully acknowledge the TEQIP committee, SV NIT, Surat for funding this study through a core research grant. The sanction order number is TEQIP/phd/2875.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

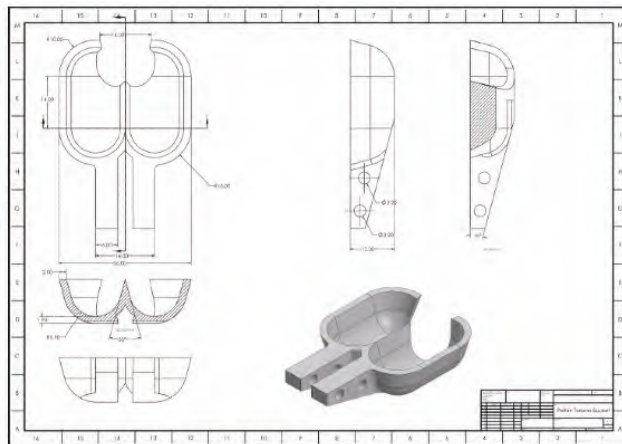
REFERENCES

- [1] Gaiser K, Erickson P, Stroeve P, Delplanque JP. An experimental investigation of design parameters for pico-hydro Turgo turbines using a response surface methodology. *Renew Energy* 2016;85:406–418. [\[CrossRef\]](#)
- [2] McCarthy MJ, Molloy NA. Review of stability of liquid jets and the influence of nozzle design. *Chem Eng J* 1974;7:1–20. [\[CrossRef\]](#)
- [3] Budiarso, Febriansyah D, Warjito, Adanta D. The effect of wheel and nozzle diameter ratio on the performance of a Turgo turbine with pico scale. *Energy Rep* 2020;:601–605. [\[CrossRef\]](#)
- [4] Alomar OR, Abd HM, Salih MM, Ali FA. Performance analysis of Pelton turbine under different operating conditions: an experimental study. *Ain Shams Eng J* 2022;13:101684. [\[CrossRef\]](#)
- [5] Kholifah N, Setyawan AC, Wijayanto DS, Widiastuti I, Saputro H. Performance of Pelton turbine for hydroelectric generation in varying design parameters. *IOP Conf Ser Mater Sci Eng* 2018;288:012108. [\[CrossRef\]](#)
- [6] Huang F, Mi J, Li D, Wang R. Impinging performance of high-pressure water jets emitting from different nozzle orifice shapes. *Geofluids* 2020;2020:8831544. [\[CrossRef\]](#)
- [7] Gupta V, Prasad V, Khare R. Effect of jet length on the performance of Pelton. *Int J Mech Eng Technol* 2016;11:11487–11494.
- [8] Jung IH, Kim YS, Shin DH, Chung JT, Shin Y. Influence of spear needle eccentricity on jet quality in micro Pelton turbine for power generation. *Energy* 2019;175:58–65. [\[CrossRef\]](#)
- [9] Zhang Z, Casey M. Experimental studies of the jet of a Pelton turbine. *Proc Inst Mech Eng A J Power Energy* 2007;221:1181–1192. [\[CrossRef\]](#)
- [10] Cobb BR, Sharp KV. Impulse (Turgo and Pelton) turbine performance characteristics and their impact on pico-hydro installations. *Renew Energy* 2013;50:959–964. [\[CrossRef\]](#)
- [11] Soe MM, War W, Swe M, Thu AM, Latt AK. Effect of jet angle on the performance of a Turgo turbine. Mandalay (Myanmar): Department of Mechanical Engineering, Mandalay Technological University; 2019.
- [12] Catanase A, Barglazan M, Hora C. Numerical simulation of a free jet in Pelton turbine. *Proc 6th Int Conf Hydraul Mach Hydrodyn* 2004;:6.
- [13] Benzon DS, Aggidis GA, Anagnostopoulos JS. Development of the Turgo impulse turbine: past and present. *Appl Energy* 2016;166:1–8. [\[CrossRef\]](#)
- [14] Williamson SJ, Stark BH, Booker JD. Performance of a low-head pico-hydro Turgo turbine. *Appl Energy* 2013;102:1114–1126. [\[CrossRef\]](#)
- [15] Williamson SJ, Stark BH, Booker JD. Low head pico hydro turbine selection using a multi-criteria analysis. *Renew Energy* 2014;61:43–50. [\[CrossRef\]](#)
- [16] Uniyal V, Kanojia N, Pandey K. Design of 5 kW pico hydro power plant using Turgo turbine. *Int J Sci Eng Res* 2016;7:363–367.
- [17] Obayes SA, Qasim MA. Effect of flow parameters on Pelton turbine performance by using different nozzles. *Int J Model Optim* 2017;7:128–133. [\[CrossRef\]](#)
- [18] Safdar I, Sultan S, Raza HA, Umer M, Ali M. Empirical analysis of turbine and generator efficiency of a pico hydro system. *Sustain Energy Technol Assess* 2020;37:100605. [\[CrossRef\]](#)
- [19] Gyanwali S, Kuikel K, Thapa A. Design and CFD analysis of pico hydro Turgo turbine. *Proc Int Symp Curr Res Hydraul. Proceedings of the International Symposium on Current Research in Hydraulic Turbines*, 2017.
- [20] Syofii I, Hidayatullah AB, Adanta D, Sari DP, Burlian F, Saputra MAA. Pico scale Turgo turbine design for remote areas application using velocity triangle approach. *J Adv Res Fluid Mech Therm Sci* 2022;97:157–167. [\[CrossRef\]](#)
- [21] Elgammi M, Hamad AA. A feasibility study of operating a low static pressure head micro Pelton turbine based on water hammer phenomenon. *Renew Energy* 2022;195:1–16. [\[CrossRef\]](#)
- [22] Ishola FA, Kilanko OO, Inegbenebor AO, Sanni TF, Adelakun AA, Adegoke DD. Design and performance analysis of a model pico size Pelton wheel turbine. *Int J Civ Eng Technol* 2019;10:727–739.
- [23] Nigussie T, Engeda A, Dribssa E. Design, modeling, and CFD analysis of a micro hydro Pelton turbine runner: for the case of selected site in Ethiopia. *Int J Rotating Mach* 2017;2017:3030217. [\[CrossRef\]](#)
- [24] Tilahun S, Paramasivam V, Tufa M, Kerebih A, Selvaraj SK. Analytical investigation of Pelton turbine for mini hydro power: for the case of selected site in Ethiopia. *Mater Today Proc* 2021;:7364–7368. [\[CrossRef\]](#)
- [25] Lin TY, Ko CY, Chen SJ, Tsai GC, Tsai HC. A novel total-flow geothermal power generator using Turgo turbine: design and field tests. *Renew Energy* 2022;186:562–572. [\[CrossRef\]](#)
- [26] Oyeboode OO. Optimization of the operational conditions for low-head Pelton wheel turbine developed for power generation. *J Eng Res Rep* 2020;:6–17. [\[CrossRef\]](#)

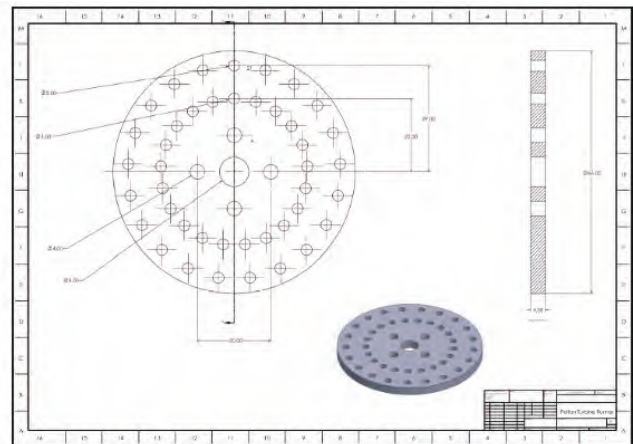
- [27] Hlabanelo JM, Sob PB, Alugongo AA. A study to improve the efficiency and performance of a Pelton wheel using potential energy at low heads. *Int J Eng Res Technol* 2020;13:2915–2926. [\[CrossRef\]](#)
- [28] Lajqi S, Bresa Q, Bresa A, Doçi I, Đurin B. Design, implementation and analysis of the overall performance of a micro hydro Turgo turbine. *J Therm Eng* 2021;7:806–822. [\[CrossRef\]](#)
- [29] Santolin A, Cavazzini G, Ardizzon G, Pavesi G. Numerical investigation of the interaction between jet and bucket in a Pelton turbine. *Proc Inst Mech Eng A J Power Energy* 2009;223:721–728. [\[CrossRef\]](#)
- [30] Xiao Y, Wang Z, Zhang J, Zeng C, Yan Z. Numerical and experimental analysis of the hydraulic performance of a prototype Pelton turbine. *Proc Inst Mech Eng A J Power Energy* 2014;228:46–55. [\[CrossRef\]](#)
- [31] Na S. Simulation of unsteady water film flow on Pelton bucket. *Energy Power Eng* 2013;5:51–55. [\[CrossRef\]](#)
- [32] Xiao YX, Zeng CJ, Zhang J, Yan ZG, Wang ZW. Numerical analysis of the bucket surface roughness effects in Pelton turbine. *IOP Conf Ser Mater Sci Eng* 2013;52:052032. [\[CrossRef\]](#)
- [33] Budiarto, Warjito, Adanta D, Puta NS, Vohra H. Cutout types analysis on pico hydro Pelton turbine. *Int J Adv Sci Eng Inf Technol* 2018;8:2024–2030. [\[CrossRef\]](#)
- [34] Warjito, Budiarto, Siswantoro AI, Adanta D, Kamal M, Dianofitra R. Simple bucket curvature for designing a low-head Turgo turbine for pico hydro application. *Int J Technol* 2017;8:1239–1247. [\[CrossRef\]](#)
- [35] Kim JW, Jo IC, Park JH, Shin Y, Chung JT. Theoretical method of selecting number of buckets for the design and verification of a Pelton turbine. *J Hydraul Res* 2017;55:695–705. [\[CrossRef\]](#)
- [36] Anagnostopoulos JS, Koukouvinis PK, Stamatelos FG, Papantonis DE. Optimal design and experimental validation of a Turgo model hydro turbine. *Proc ASME 11th Bienn Conf Eng Syst Des Anal* 2012;157–166. [\[CrossRef\]](#)
- [37] Takagi M, Watanabe Y, Ikematsu S, Hayashi T, Fujimoto T, Shimatani Y. 3D-printed Pelton turbine: how to produce effective technology linked with global knowledge. *Energy Procedia* 2014;1593–1596. [\[CrossRef\]](#)
- [38] Adanta D, Warjito, Febriansyah D, Budiarto. Feasibility analysis of a pico-scale Turgo turbine bucket using coconut shell spoons for electricity generation in remote areas in Indonesia. *J Adv Res Fluid Mech Therm Sci* 2020;69:85–97. [\[CrossRef\]](#)
- [39] Arun K, Kumar KM, Karthikeyan KMB, Mohanasutan S. Analysis on influence of bucket angle of Pelton wheel turbine for its structural integrity using aluminium alloy (A390), austenitic stainless steel (CF20), grey cast iron (325) and martensitic stainless steel (410). *Mater Today Proc* 2022;1045–1053. [\[CrossRef\]](#)
- [40] Gallego E, Rubio-Clemente A, Pineda J, Velásquez L, Chica E. Experimental analysis on the performance of a pico-hydro Turgo turbine. *J King Saud Univ Eng Sci* 2021;33:266–275. [\[CrossRef\]](#)
- [41] Velásquez L, Posada A, Chica E. Optimization of the basin and inlet channel of a gravitational water vortex hydraulic turbine using the response surface methodology. *Renew Energy* 2022;187:508–521. [\[CrossRef\]](#)
- [42] Betancour J, Romero-Menco F, Velásquez L, Rubio-Clemente A, Chica E. Design and optimization of a runner for a gravitational vortex turbine using the response surface methodology and experimental tests. *Renew Energy* 2023;210:306–320. [\[CrossRef\]](#)
- [43] Guerra J, Velásquez L, Rubio-Clemente A, Jaramillo L, Chica E. Design and optimization of a siphon turbine using the response surface methodology. *Results Eng* 2024;22:102241. [\[CrossRef\]](#)
- [44] Bouvant M, Betancour J, Velásquez L, Rubio-Clemente A, Chica E. Design optimization of an Archimedes screw turbine for hydrokinetic applications using the response surface methodology. *Renew Energy* 2021;172:941–954. [\[CrossRef\]](#)
- [45] Singh U, Gupta NK. Thermal performance analysis of heat pipe using response surface methodology. *J Therm Eng* 2023;411–423. [\[CrossRef\]](#)
- [46] Gambhir D, Sherwani AF, Arora A, Ashwni. Parametric optimization of blowdown operated double-effect vapour absorption refrigeration system. *J Therm Eng* 2022;8:78–89. [\[CrossRef\]](#)
- [47] Solanki A, Pal Y. Evaluation and optimization of single-effect vapour absorption system for the dairy industry using design of experiment approach. *J Therm Eng* 2022;8:619–631. [\[CrossRef\]](#)
- [48] Ali OM. Spark ignition engine performance analysis with low octane gasoline and methyl tert-butyl ether additive for optimum operation. *J Therm Eng* 2024;10:911–923. [\[CrossRef\]](#)
- [49] Thake J. The micro-hydro Pelton turbine manual. Rugby (UK): Practical Action Publishing; 2000. [\[CrossRef\]](#)
- [50] Ebhota WS, Inambao F. Design basics of a small hydro turbine plant for capacity building in Sub-Saharan Africa. *Afr J Sci Technol Innov Dev* 2016;8:111–120. [\[CrossRef\]](#)
- [51] Giosio DR, Henderson AD, Walker JM, Brandner PA, Sargison JE, Gautam P. Design and performance evaluation of a pump-as-turbine micro-hydro test facility with incorporated inlet flow control. *Renew Energy* 2015;78:1–6. [\[CrossRef\]](#)
- [52] Box GEP, Hunter JS, Hunter WG. Statistics for experimenters: design, innovation, and discovery. 2nd ed. Hoboken (NJ): Wiley; 2005.
- [53] Raymond MH, Douglas MC, Christine AC. Response surface methodology: process and product optimization using designed experiments. 4th ed. Hoboken (NJ): Wiley; 2016.
- [54] Gunaraj V, Murugan N. Application of response surface methodology for predicting weld bead quality in submerged arc welding of pipes. *J Mater Process Technol* 1999;88:266–275. [\[CrossRef\]](#)

APPENDICES

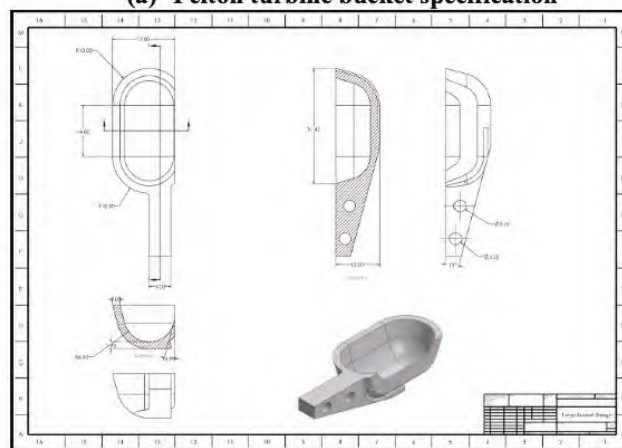
Appendix A1 Pelton Turbine Runner (PTR) and Turgo Turbine Runner (TTR) Design Specification



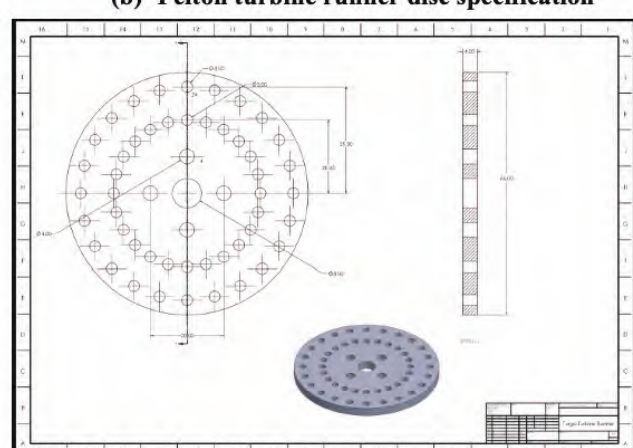
(a) Pelton turbine bucket specification



(b) Pelton turbine runner disc specification



(c) Turgo turbine bucket specification



(d) Turgo turbine runner disc specification

Appendix A2: Model summary statistics

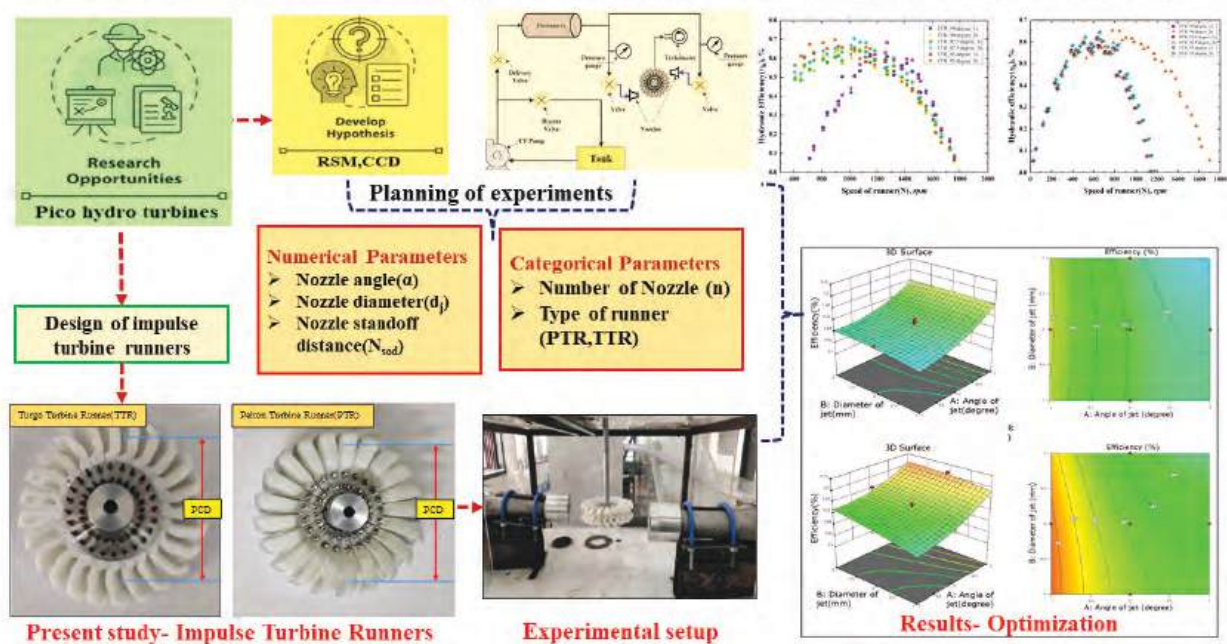
Source	Std. Dev.	R ²	Adjusted R ²	Predicted R ²	PRESS	-----
Linear	0.0094	0.8623	0.8530	0.8366	0.0077	-----
2FI	0.0088	0.8956	0.8712	0.8234	0.0084	-----
Quadratic	0.0053	0.9639	0.9532	0.9375	0.0030	Suggested
Cubic	0.0054	0.9743	0.9516	0.8710	0.0061	Aliased

Appendix A3: Fit value for experimental and predicted fit value for experimental and predicted.

Std. Dev.	0.0053	R ²	0.9639
Mean	0.6540	Adjusted R ²	0.9532
C.V. %	0.8090	Predicted R ²	0.9375
PRESS	0.0030	Adeq Precision	44.3384

Graphical abstract

Performance optimization of Low-Head Vertical Axis impulse turbine runners using Response Surface Methodology





Research Article

Advanced thermal management in aircraft lithium-ion battery packs: optimization of heat dissipation using heat spreaders and phase change materials

Srinivas MALLIMOGGALA^{1,*}, K Rama DEVI¹

¹Department of Electronics and Communication Engineering, JNTU College of Engineering Kakinada, A.P, India

ARTICLE INFO

Article history

Received: 28 August 2024

Accepted: 10 December 2024

Keywords:

Aluminum Heat Spreaders;
Battery Safety; Cooling
Strategies, COMSOL
Multiphysics; Heat Dissipation;
Lithium-Ion Batteries; Thermal
Management

ABSTRACT

Lithium-ion battery packs are essential in aviation, particularly for aircraft like the Cessna, Cirrus, and Piper models usually with 24V, and 16Ah to 35Ah capacities, depending on the specific model and avionics package installed, due to their high energy density and weight efficiency, which are critical for optimizing performance and fuel economy. This paper makes an exploration when it comes to spreading the heat from Auxiliary Power Unit (APU) battery packs with various cooling methods. This study addresses thermal management challenges, including the risk of thermal runaway, which can jeopardize safety challenges by optimizing heat dissipation in a 7s4p battery pack used in these aircraft, employing cooling techniques such as air cooling, phase change materials (PCMs), and aluminum-based heat spreaders (Al 3003-H18 and Al 6063-T83) to meet typical aviation requirements for reliability and efficiency. The simulations proved by COMSOL Multiphysics® pointed out that the aluminum variants especially the Al 3003-H18 significantly reduced peak temperatures (64.27 °C) compared to air cooling (82.36°C) at 8C, thus offered best thermal regime capability and managed the peak temperature as well as the voltage across the different rates of discharged. Pearson's correlation coefficient analysis also showed positive higher order linear regression between aluminum-based models highlighting their efficiency in dealing with the heat generation or thermal runaway. This work extends existing literature by applying aluminum heat spreaders for aviation-specific applications, offering new insights into the relationship between thermal properties and cooling strategies under high discharge conditions, thereby enhancing both safety and battery longevity in critical aviation operations.

Cite this article as: Mallimoggala S, Devi KR. Advanced thermal management in aircraft lithium-ion battery packs: optimization of heat dissipation using heat spreaders and phase change materials. J Ther Eng 2025;11(5):1420–1438.

*Corresponding author.

*E-mail address: mallimoggla@gmail.com

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkilic



Published by Yıldız Technical University Press, İstanbul, Turkey

Yıldız Technical University. This is an open access article under the CC BY-NC license (<http://creativecommons.org/licenses/by-nc/4.0/>).

INTRODUCTION

Lithium-ion (Li-ion) batteries are highly sought after in aviation because of their high energy density, low weight, and versatility in terms of powering various aircraft equipment. They have proved to be more efficient and performance in comparison to the earlier used Nickel-Cadmium (Ni-Cd) batteries. The transition began markedly with Boeing 787 that has Li-ion battery for voltage regulation, starting the engine, and for other electrical operations. These batteries have some benefits; however, they are associated with thermal hazards of thermal runaway, and as a result demand effective Battery Thermal Management System (BTMS) to track conditions and failures [1].

Battery safety is crucial in preventing thermal runaway, a dangerous reaction that can lead to fires or explosions, especially in lithium-ion batteries. Such that an efficient thermal management is essential to control heat generation and mitigate the risks of overheating, ensuring safe operation in electric vehicles and other high-demand applications. Addressing such thermal runaway conditions is vital to protect users, improve battery reliability, and support the widespread adoption of lithium-ion technology. In dynamics of aviation, the utility of the BTMS rests upon on reliability aspects that must include safety, functions, standards, and technology. To lower fire hazards, at the same time have gross batteries for electric systems, which cuts down the usage of hydraulic or pneumatic systems. Public trust and operational approval in the industry can only be won when compliance with regulations is achieved [2].

Despite the future Boeing 777 coming up with safety issues of lithium-ion battery, the FAA came up with special conditions in 2013 [3,4], while the GA aircrafts like Cessna, Cirrus and Piper among others, utilize lead-acid batteries that are sealed with 24 volts and 16 Ah with battery models like Gill 7242-16 LT and Gill 7243-16 LT. These battery packs are non-noticeable, rugged and maintenance free and such are ideal to be used in short range aircraft [5,6].

The COMSOL Multiphysics® Reference Manual [7] is a comprehensive guide for engineers and researchers, providing detailed instructions on setting up simulations, applying boundary conditions, and solving complex multi-physics problems. The COMSOL Multiphysics® Battery Design Module is a guide for designing and battery modeling.

Liu et al. [8] emphasize the need for advanced thermal management of lithium-ion batteries in extreme conditions like high temperatures, fast charging, and preventing thermal runaway in EVs. The study explores external heat transfer systems and internal smart battery innovations with embedded sensors. However, it lacks practical analysis on integrating these external and internal systems, leaving a gap in understanding their real-world compatibility and effectiveness.

Figueiras et al. [9] explores hybrid-electric propulsion systems for regional aircraft in the FutPrInt50 project, focusing on three powertrain architectures: series, parallel,

and turboelectric. Using computational and experimental data, parametric studies evaluate the effects of battery parameters and hybridization factors on performance. A thermal management system and multi-objective optimization aim to minimize mass and emissions. While the study offers performance insights, it lacks details on real-world scalability, experimental data sources, and practical feasibility for broader commercial applications.

Coutinho et al. [10] explores thermal management systems (TMS) for hybrid-electric aircraft propulsion to manage waste heat and reduce emissions. Five TMS architectures are modeled using Matlab/Simulink, comparing their cooling performance and impact on flight. Results does not show a single architecture is ideal, indicating the need for further development.

Capron et al. [11] gives a paper that employs COMSOL Multiphysics® and MATLAB® modeling to analyze the temperature distribution of battery packs. They derive two 2D models with homogenized thermal properties and other model with cell layer properties in detail to have enhanced thermal specifications. Shetty et al. [12] investigates the use of mini-channel cooling plates with a streamline configuration for thermal management of Li-ion batteries in electric vehicles. Computational analysis of five plates with varying channel numbers (3-7) at a coolant flow rate of 0.002 kg/s showed a 7.8% reduction in maximum cell temperature for the 7-channel design. Temperature uniformity also improved, suggesting this design effectively maintains battery temperatures within a safe range and could enhance future battery thermal management systems.

Zhu et al. [13] presents an improved simulation method for small-size, fast-activation thermal batteries using the COMSOL platform. A two-dimensional axisymmetric model was developed to simulate temperature changes and electrolyte melting under high-temperature conditions (50°C). The results show that additional heating flakes and improved insulation reduce heat loss, with most melting heat flowing from the negative side. The activation time of the thermal battery was calculated to be 91.2ms, with an error margin within 10% compared to experimental results, demonstrating the model's high accuracy and broader applicability for thermal battery simulations.

Zhao, Chen, and He [14] focus on the thermal management of nickel-cobalt lithium manganate (NCM) batteries for electric vehicles, using COMSOL to simulate five liquid-cooled temperature control models. A performance evaluation system based on the analytic network process (ANP) and system dynamics (SD) ranks Models 1 and 5 as the most effective. However, the study lacks real-world testing and validation, leaving a gap in understanding how these models would perform in practical conditions.

A thermal model for lithium-ion battery cells and packs is developed for COMSOL Multiphysics® by Dunning et al. [15] which modeled a battery cell, incorporating factors like electrode thickness and particle size. A 5C discharge simulation showed a 14°C temperature rise for a single cell,

validated against experimental data. When applied to a battery pack, the temperature increase was 19.6°C, with a 9% error margin. While effective for preliminary design, the study lacks real-world validation and practical insights for large-scale applications.

Sun et al. [16] assesses a battery thermal management system (BTMS) with phase change materials (PCMs) and liquid cooling. It discovers that incorporation of the PCM with liquid cooling leads to reduction of the maximum temperature and enhancement of the thermal uniformity with 10% mass fraction expanded graphite in paraffin wax PCM.

V. K. Satheesh et al. [17] proposes a converging tapered airflow duct to enhance temperature uniformity and reduce peak temperatures. Using computational fluid dynamics validated by experimental data, the tapered design improved heat dissipation and reduced peak temperature rise by 20% and temperature variation by 19%. However, the study lacks insights into practical integration and long-term durability of the proposed duct in real-world electric vehicle applications.

P. R. Gharde and S. N. Havaladar, [18] investigates a floating capsule thermal energy storage system (TESS) using beeswax and paraffin as phase change materials (PCMs) to improve heat storage capacity. The system features spherical capsules filled with beeswax within a paraffin-filled cylindrical shell, with heat transfer fluid flowing through hexagonal tubes. A mathematical model predicts a 21.5% increase in storage capacity with a 70–30% beeswax-paraffin mix. Charging and discharging times were estimated at 2.6 and 3.2 hours, respectively. However, the study lacks real-world testing, and the long-term stability of the PCMs is not discussed.

S. Gungor et al. [19] highlights the critical role of thermal management strategies for lithium-ion (Li-ion) batteries, especially as transportation and electricity generation shift towards renewable sources. And compares various thermal management strategies based on battery geometry, coolant type, and heat transfer methods, but lacks detailed analysis on post-lithium batteries and real-world scalability of these strategies in the EV industry.

B. Palmieri et al. [20] used high thermally conductive materials, particularly graphite nanoplatelet films, is being explored to enhance battery performance and safety by improving heat dissipation. This study employs a numerical model using the finite element method in COMSOL to predict heat generation during charging and discharging cycles. The findings indicate that integrating thermally conductive materials can protect neighboring cells from heat propagation, helping to maintain uniform temperature and delay thermal runaway in battery modules.

The main objective of this paper is to review and implement an enhanced thermal system in a 24V 16Ah lithium-ion battery pack used in aircraft application; the APU battery packs. Specifically, these battery packs are essential in ensuring constant power to several aircraft systems, and

the efficiency and effectiveness of the systems are determined by the battery packs' performance. Since lithium-ion batteries are define by high thermal risks such as thermal runaway where the battery produces high heat and, in some cases, cause fires, heat management is crucial.

Based on the above objective, the study incorporates an extensive Simulation technique by applying COMSOL Multiphysics® to study thermal characteristics of a 7s4p battery layout. Consideration is given to several cooling methods based on air circulation as well as PCMs and various heat spreaders made of aluminum. Thus, the goals of the study, which focus on the comparison of the thermal characteristics of these materials and cooling methods, are to determine how heat can be shed and battery temperature regulated under different discharge rates.

In addition, the present research focuses on the identification of various cooling options and their effects on the battery temperature and voltage on the strength. This involves studying the aluminum variants like Al 3003-H18 against other materials- in matters to do with sustaining lower values of maximum peak temperature and constant value of voltage.

This research study is poised to contribute its findings towards improving the thermal management systems for lithium-ion batteries in application in aviation hence improving on their safety and durability. Lastly, it is the hope of this paper to present useful information and recommendation to the engineers and researchers involved in the thermal management of battery in aviation while contributing to the general objective of enhancing the application of lithium-ion batteries in aircraft while providing operational safety and efficiency.

MODELING AND METHODOLOGY

This battery configuration and capacity of A 24V, 16Ah lithium-ion battery pack has a lot of advantages for GA aircrafts such as Cessna, Cirrus, and Piper. It has a longer cycle life, require less maintenance, increases starter power, and the Battery Management system. These batteries can interface directly with the new age avionics and electrical systems and deliver stable voltage as well as sufficient current. Besides, they are safe for environment as they do not include toxic component, namely lead.

In the thermal model of the battery pack, the primary heat source is the heat generated during the battery's discharge process. This heat generation is a result of the internal resistance and discharge rate (C-rate) of the battery cells. When modeling large battery packs, detailed geometrical and chemical information about individual battery cells may be impractical due to computational constraints, leading to the use of simpler zero-dimensional models [21]. These models, like equivalent circuits or lumped battery models, aggregate the complex interactions of ions and voltages into simplified representations, enabling efficient analysis of cell-to-cell current distribution within the pack.

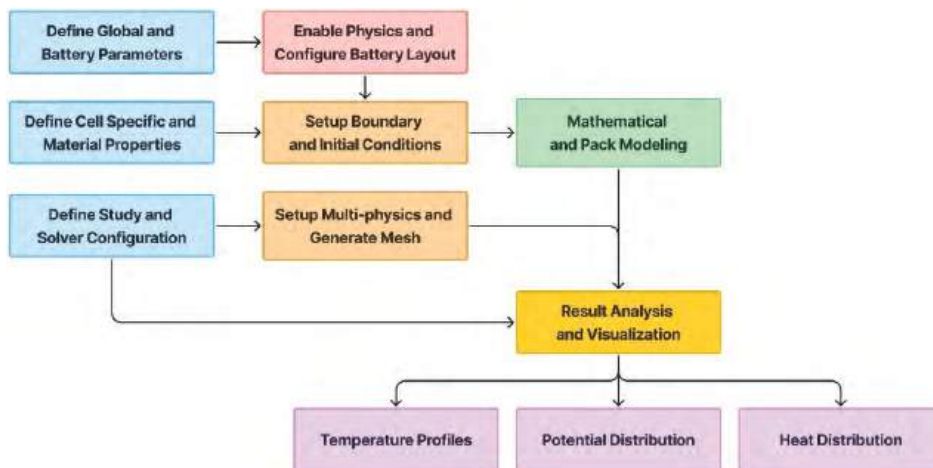


Figure 1. Flowchart for battery pack simulation process in COMSOL.

The modeling and simulation are performed by the help of multipurpose software ‘COMSOL Multiphysics®’.

Figure 1 depicts the flowchart which indicates a detailed methodological approach to the simulation of battery systems based on the global parameters and the properties of the materials used. This includes development in physical architecture layout of the battery, the imposition of constraints on the problem formulation by defining the boundary conditions, inputting of solver settings, generation of the mesh to discretize the problem, mathematical modeling of the battery pack and interpretation of results as concerns the distribution of temperature of the battery bundles, the distribution of the potential across the battery pack and the distribution of heat generation for purposes of optimization.

Geometry and Battery Parameters

Another way that the researchers present the parameters is a table 1 with a detailed lookup of parameters that has been divided into Battery Pack Metrics, Physical Dimensions, Cooling Plate Dimensions, Layout Configuration, Electrical Parameters, Activation Energies, Thermal Properties, and Initial Conditions. This data is useful for creating and prototyping battery packs with performance parameters assessed under various system operational situations and system designs.

Battery Pack Metrics, Physical Dimensions, and Cooling Plates Dimensions can be seen as fundamental components of a battery pack, that explain what the battery pack is capable of and what it cannot do. They include nominal voltage, full charge voltage, capacity, and the current at the different discharge rates. Some of the non-electrical parameters, which include battery diameter, height, and sizes of the connectors, are critical in developing geometrical models in the simulation process. Cooling Plates Dimensions contain detailed information on cooling plates’ dimensions for distinct models (B, C, D), which means that simulations

take into consideration the real space occupied and x-coordinate for evaluating their performance in heat dissipation.

Boundary and Initial Conditions

Table 1 also can be associated with the exterior and inside conditions of the battery pack, for instance, thermal properties, electrical parameters, as well as the outer/inner limits of the battery pack. The Thermal Properties part of the table contains such parameters as “thermal conductivity in-plane” “battery density and “battery heat capacity,” which define thermal interaction with the battery cells and between them. heat transfer coefficient, defines heat exchange with the environment and affects the thermal behavior of the battery pack.

Electrical Parameters assist in simulating the electrochemical behavior in battery packs making constant voltage and specific resistance, while the Activation Energies section assists in the determination of electrode materials. These conditions mimic the functioning of the battery pack when it is in the state of charging and discharging and under varying electrical loads. Geometric sizes of battery such as the diameter, the radius, and the height of battery and the cooling plate size which provide the overall spatial condition for modeling the battery pack since they determine space and mechanical interfaces of the system is provided in table 1.

“E_{OCP},” which stands for Open Circuit Potential, is the voltage of batteries when the currents are not flowing through it and it is recharging or discharging; its relation with the degree of charge is very important to monitor the behavior of batteries; therefore, E_{OCP} plays the role of a significant input for simulators that are used for assessing the battery life. Thus, “dEdT”, the Derivative of Voltage with Respect to Temperature is important to control battery thermal state and designing proper cooling in COMSOL effected environments where temperature varies frequently. Therefore, assist in designing batteries that are efficient as

Table 1. Battery pack parameters and configurations for modeling

Category	Name	Value	Description
Battery pack Metrics	Simulated time	720	Model Simulation time period (s)
	Nominal voltage	24.6	Nominal Voltage of Battery Pack (V)
	Full Charge voltage	29.4	Full Charge Voltage of Battery Pack (V)
	Nominal battery capacity	16	Nominal Battery Pack Capacity (Ah)
	0.5C discharge rate	8	Discharge current at 0.5C (A)
	1C discharge rate	16	Discharge current at 1C (A)
	2C discharge rate	32	Discharge current at 2C (A)
	4C discharge rate	64	Discharge current at 4C (A)
	6C discharge rate	96	Discharge current at 6C (A)
	8C discharge rate	128	Discharge current at 8C (A)
Physical Dimensions	Battery diameter	18	Battery diameter (mm)
	Battery radius	9	Battery radius (mm)
	Battery height	65	Battery height (mm)
	Terminal height	1	Terminal height (mm)
	Terminal radius	3	Terminal radius (mm)
	Serial connector width	2	Serial connector width (mm)
	Serial connector height	1	Serial connector height (mm)
	Parallel connector height	0.5	Parallel connector height (mm)
	Parallel connector Width	1	Parallel connector width (mm)
Cooling Plates dimensions (Width × depth × height)	Series plate dimension Model-B	9 × 99 × 65	Series Plate dimension Model B (mm)
	Parallel plate dimension for model-B	180 × 9 × 65	Parallel Plate dimension for Model B (mm)
	Plate dimension for model C	9 × 72 × 65	Plate dimension for Model C (mm)
	Plate dimension for model D	126 × 9 × 65	Plate dimension for Model D (mm)
	Total number of batteries	28	Total Number of cells in the battery pack
Layout Configuration	Number of batteries - X	7	Number of batteries in x-direction
	Number of batteries - Y	4	Number of batteries in y-direction
	Number of plates - X	6	Number of plates in x-direction
	Number of plates - Y	3	Number of plates in y-direction
Electrical Parameters	Battery cell capacity	4	Battery cell capacity (Ah)
	Current-1C	4	1C current (A)
	C-rate	0.5, 1, 2, 4, 6, 8	C rates indicating the rate of charge/discharge (C)
Activation Energies	Activation energy for eta1C	24000	Activation energy (J/mol) associated with resistive property
	Activation energy for J0	−59000	Activation energy (J/mol) relates to the current density
	Activation energy for Tau	24000	Activation energy (J/mol) with a time constant (τ)
Thermal Properties	Thermal conductivity in-plane	30	Thermal conductivity in-plane (W/m·K)
	Thermal conductivity cross-plane	1	Thermal conductivity cross-plane (W/m·K)
	Battery density	2000	Battery density (kg/m ³)
	Battery heat capacity	1400	Battery heat capacity (J/kg·K)
	Heat transfer coefficient	30	Heat transfer coefficient (W/m ² ·K)
Initial Conditions	Reference temperature	293.15	Reference temperature (K)
	Reference temperature J0	0.85	J0 at reference temperature (A/m ²)
	Reference tau time	1000	tau at reference time (s)
	Reference voltage eta_1C	0.0045	eta_1C at reference voltage (V)
	Int-ex temperature	293.15	Initial/external temperature (K)

well as safe by giving a holistic and detailed prospect of how they will behave in different electrical loads and various thermal conditions.

When it comes to the 7s4p battery pack configuration, for probe selection where cells are 1-28, cell temperature and cell potential are to be monitored. This selection includes all the cells in the battery pack making sure that thermal and electrical characterization of the array is completely captured. It is important to track these parameters if one intends to know how each cell is behaving in different operations to ensure the required performance, safety and durability of the battery system is achieved.

Layout Configuration and Material Properties

This paper presents four individual models of a 7s4p battery pack structure with different positions of the cells and cooling plates for thermal management schemata, as shown in Figure 2(a). Heat spreaders are another pair of plates with configurations of series, parallel, and grids connected to the batteries' directions. The construction parameter that has been used in the modeling is described in the following table 1.

Model A is a straight or inline pack that consist of twenty-eight individual cells labelled 1 through to 28 and for this model there are no cooling plates employed. In Model B the grid format which is compact, has more rigid rows and columns of the cells thus enhancing the uniformity of cooling. Enduring cooling plates which were positioned in PP1 to PP3 as well as SP1 to SP6, which increases the utilization and efficiency by creating a direct heat dissipation channel from cells to the cooling plates.

Model C and Model D in addition to the above model for heat distribution, it alters the mode of the cells in series and parallel for redundancy. Coupling plates SP1 to SP6 and pre-plates PP1 to PP3 are disposed between the plates like as shown in Model B. In addition, different materials are utilization to perform testing for Model B to establish the most efficient type of passive cooling system.

The characteristics of the free tetrahedral mesh used in simulations for proposed four different battery pack models (A, B, C, D) are shown in the Table 2. This data is instrumental in assessing the quality and the level of detail of the mesh required for FEM simulations which are very

important for computational modelling of physical processes in the battery packs.

This figure 2(b) illustrates the temperature-dependent behavior of key material properties in a battery pack system, along with a schematic of the battery's physical layout. The graphs show that as temperature increases, the dynamic viscosity ($\eta(T)$) and specific heat capacity ($C_p(T)$) rise, while the density ($\rho(T)$) decreases. Additionally, the thermal conductivity ($k(T)$) and speed of sound ($cs(T)$) both increase with temperature. These trends are important for understanding the thermal performance and material stability of the battery pack during operation.

Some of the types of meshes that are used include vertices, tetrahedra, triangles, edge elements, vertex elements, minimum element quality, average element quality, element volume ratio, mesh volume, space dimension, domains, boundaries, edges, and vertices. Quality of the mesh impacts simulations and the lower and average elements impact it bearing stability and accuracy. The number of nodes must be equal to the size of the modelled object, the size dimension should be set to 3. Another factor that helps create a mesh with better detail is also counting of the number of domains, boundary, edges, and vertices.

It is helpful when it comes to creating quality mesh in simulations to pick up all the necessary physical events. Such characteristics as minimum quality and average quality cause shifts in the range of results' stability and reliability. The implementation of the well-constructed mesh allows one to predict and analyze the battery pack's physical behaviors more efficiently. This table offers a specific description of mesh characteristics in each model, so that the researchers can judge the mesh for the capability of the analysis of the complex behavior and its adequacy in different configurations of the model.

Figure 3, also gives a clear and comprehensive comparison of four Battery pack models (A, B, C, D) that shows the design aspects which could use for further examination in simulation for purpose of optimization, the figure offers a broad perception of the structural and systems and features of the Battery pack models for the purpose of analysis.

For each model of the housing, different designs are incorporated to safeguard the inner most parts of the battery packs, including the mechanical and external aspects. Mesh design visualizations are essential for the simulation

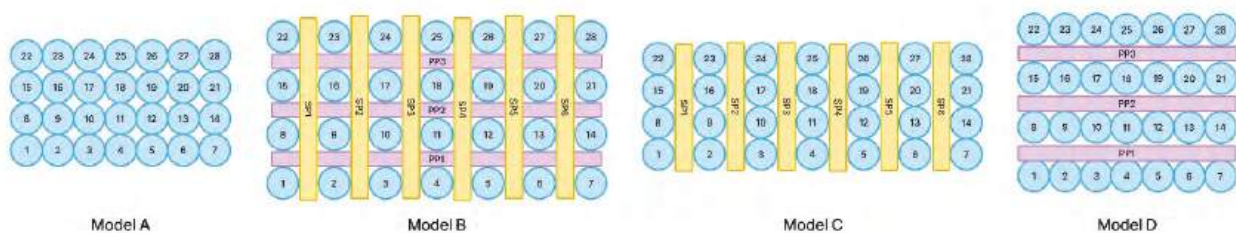


Figure 2(a). Schematic Layouts of Battery Pack Configurations: Models A, B, C, and D.

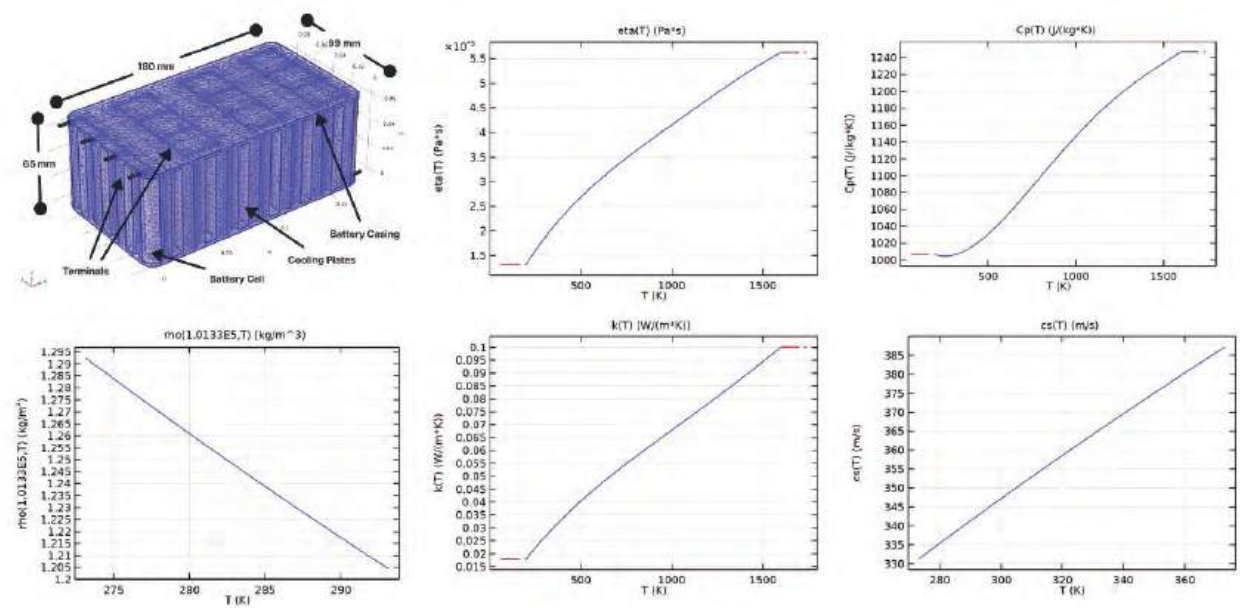


Figure 3(b). Thermal and Material Properties of Battery Pack Components as a Function of Temperature.

Table 2. Comparison of mesh characteristics and quality metrics across models A, B, C, and D

Type	Parameter	Model A	Model B	Model C	Model D
Free Tetrahedral mesh	Mesh vertices	52554	54316	44698	69895
	Tetrahedra	291782	304479	249897	392250
	Triangles	68149	75108	62833	90140
	Edge elements	8276	12021	9038	10289
	Vertex elements	1386	1704	1482	1452
	Number of elements	291782	304479	249897	392250
	Minimum element quality	0.04949	0.01485	0.0203	0.01798
	Average element quality	0.5787	0.5617	0.5548	0.5757
	Element volume ratio	5.75E-04	3.40E-04	3.55E-04	5.34E-04
	Mesh volume (m ³)	6.339E-4	0.001242	9.06E-4	8.72E-4
Geometry	Space dimension	3	3	3	3
	Number of domains	223	224	229	226
	Number of boundaries	1417	1750	1501	1477
	Number of edges	2068	3328	2800	2749
	Number of vertices	1386	1704	1482	1452
Battery Volumetrics	Battery Cell Volume (m ³)	1.65×10 ⁻⁵	1.65×10 ⁻⁵	1.65×10 ⁻⁵	1.65×10 ⁻⁵
	Battery Pack Volume (m ³)	4.62×10 ⁻⁵	4.62×10 ⁻⁵	4.62×10 ⁻⁵	4.62×10 ⁻⁵
	Total Volume of Cooling Plates (m ³) -		6.63×10 ⁻⁴	4.21x10 ⁻⁵	7.37x10 ⁻⁵

to be accurate, because they show computational grids that are used in finite element analysis.

As for terminals, they impact the electrical efficiency and performance; concerning active battery configurations, they define space, connectivity, and thermal properties. The cooling design is shown for each model, only for Model A, no integrated cooling system is present. Among the

presented models, B, C, and D have cooling systems, which proves that they are designed for critical applications.

Table 3 depicts a list of materials and their thermal characteristics that exist in a 7s4p battery design. It lists the materials that should be avoided when constructing the battery terminals and Active battery as well as different cooling methods. Battery terminals and active battery components are said to be

Table 3. Material properties for plates used in battery pack models

Condition	Material	Heat capacity at constant pressure (J/kg·K)	Thermal conductivity (W/m·K)	Density (kg/m ³)
Battery terminals	Aluminum	900	238	2700
Active battery	lithium- ion	1400	30	2000
Cooling model A	7s4p Straight configuration without cooling			
Cooling model B1	Air	1005	0.026	1.225
Cooling model B2	Paraffin solid	1850	0.4	861
Cooling model B3	Paraffin liquid	2384	0.15	778
Cooling model B4	Aluminum	900	238	2700
Cooling model B5	Aluminum 6063-T83	900	201	2700
Cooling model B6	Aluminum 3003-H18	893	155	2730
Cooling model C	Aluminum in Series	900	238	2700
Cooling model D	Aluminum in Parallel	900	238	2700

made of aluminum or lithium-ion and their heat and electrical conductivity, density, and heat capacity is given.

This table 3 also presents four cooling models: A Model A without any cooling plates. Model B utilization includes air, PCM (paraffin) as well as heat spreaders (aluminum and its alloys). And Model C and D, it employs the use of aluminum in series and parallel circuits separately. Thus, the specific objectives are the following: Assess the thermal conductivity and apparent heat capacity of air, paraffin, aluminum and aluminum alloys (6063-T83, 3003-H18) in various battery pack arrangements.

Multiphysics and Mathematical Model

Some basic equations that are frequently used while modeling the electrochemical as well as the thermal aspects of a battery cell includes Cell Voltage Equation represented by (1), The definition of 1C current represented by equation (2), applied current as represented by equation (3), Volume integral of voltage as represented by equation (4), Normalization condition as represented by equation (5). These equations define several coefficients related to electrical characteristics of battery cells, voltage, current and internal processes that are important in the battery cell simulation and creation of control algorithms in the battery management systems [20].

$$E_{cell} = E_{OCV}(SOC, T) + \eta_{IR} + \eta_{act} \quad (1)$$

$$I_{1C} = \frac{Q_{cell,0}}{3600(s)} \quad (2)$$

$$I_{cell} = I_{app} \quad (3)$$

$$V_{cell} = \int_{\Omega_{cell}} d_{vol} \Omega \quad (4)$$

$$d_{vol} = 1 \quad (5)$$

The thermal, electrochemical working in a battery cell, including the thermal aspect, heat production and the relation between both aspects. The Heat Equation with Internal Heat Generation: heat conduction in solids (6) and liquids (7) are important tool for describing thermal processes in battery and evaluating temperature fields. They include heat conduction, convective heat transfer, and internal heat generation and takes account of it (8).

$$\rho c_p \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T) + Q_h \quad (6)$$

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p u \cdot \nabla T = \nabla \cdot (k \nabla T) + Q_h \quad (7)$$

$$Q_h = \left(\eta_{IR} + \eta_{act} + T \frac{\partial E_{OCV}(SOC|_{X=1,T})}{\partial T} \right) I_{cell} + Q_{mix} \quad (8)$$

These equations provided describe various aspects of thermal dynamics within a battery system. The Mixing Heat Equation (9) calculates the heat generated from mixing processes within the battery. The Thermal Open-Circuit Voltage Equation (10) defines how the thermal effects alter the open-circuit voltage. The Thermal Conduction Equation (11) illustrates the relationship between thermal conductivity and heat transfer, where the heat flux is proportional to the temperature gradient within the material and the difference between the surface temperature and ambient temperature. Together, these equations provide a comprehensive framework for understanding thermal management in battery design [7].

$$Q_{mix} = \frac{3Q_{cell}}{\tau} \int_0^1 \left(\frac{\partial E_{OCV,therm}}{\partial SOC} \frac{\partial SOC}{\partial x} \frac{\partial SOC}{\partial x} x^2 dx \right) \quad (9)$$

$$E_{OCV,therm} = E_{OCV,ref}(SOC) - T_{ref} \frac{\partial E_{OCV}(SOC)}{\partial T} \quad (10)$$

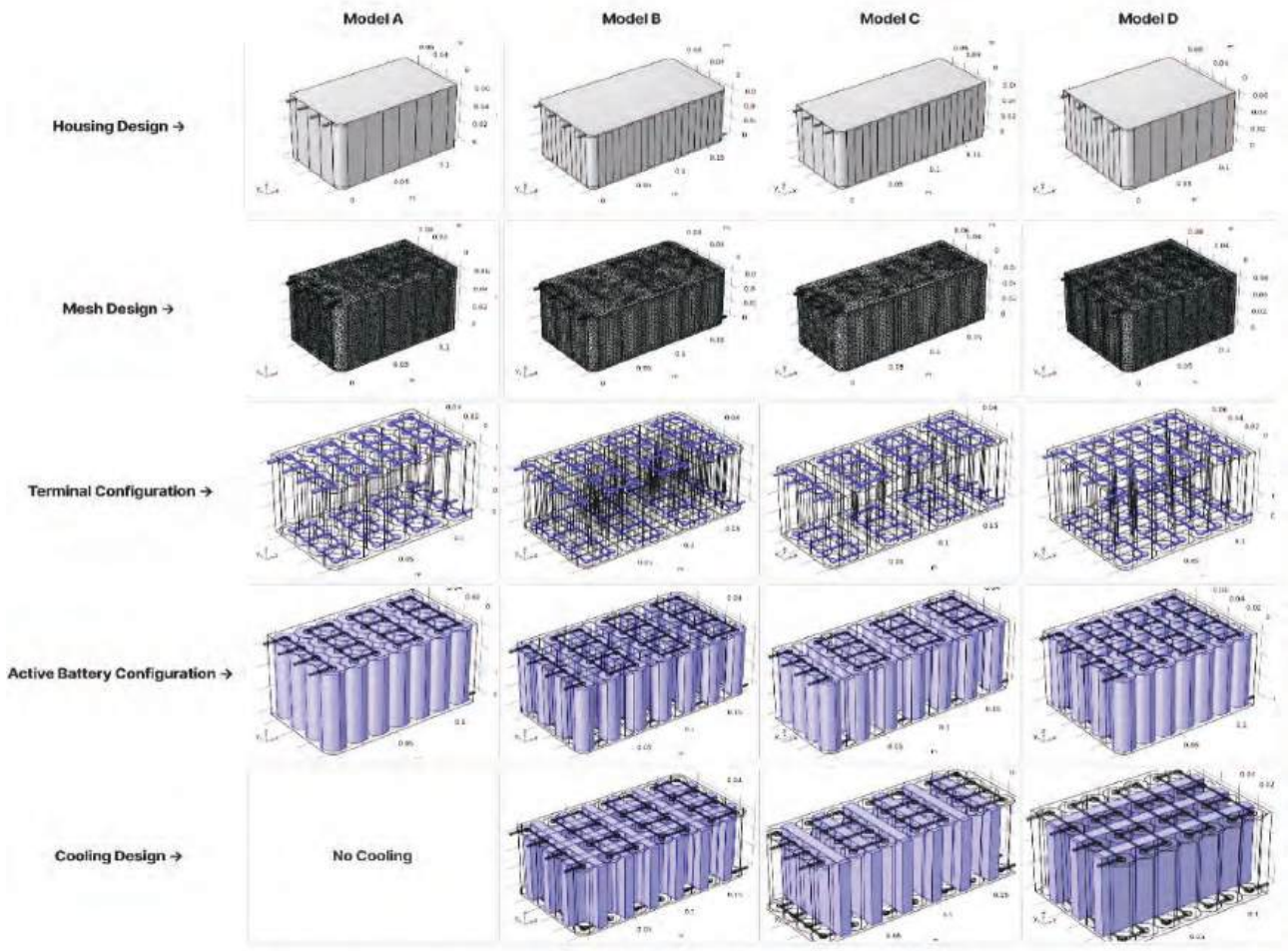


Figure 4. Comparison of Battery Pack Configurations and Cooling Designs.

$$-k_s \frac{\partial T}{\partial n} = h_{nat}(T_s - T_a) \quad (11)$$

These equations are critical for simulations of the battery thermal management and safety and can be used to optimize the design of the thermal systems, reduction of the risk of thermal runaway and setting up thermal safe limits. They also forecast temperatures and their effects on batteries with respect to their usage in different applications and life span [7,22]. Equations for heat transfer through boundaries: (12) equations for heat convection: (13) and for heat accumulation and specific heats (14).

$$q = -k \nabla T \quad (12)$$

$$q_{conv} = h(T_s - T_a) \quad (13)$$

$$\rho c_p \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T) + Q_h \quad (14)$$

To determine the thermal behavior of the battery pack during operation, theoretical calculations were performed based on the relevant input parameters provided in Table 1.

These calculations consider the battery pack metrics, physical dimensions, cooling plate dimensions, layout configuration, electrical parameters, activation energies, thermal properties, and initial conditions. The heat generated within the battery pack due to internal resistance during discharge was calculated using Joule's law, as shown in Equation (15), Total Heat Capacity of the System (16), Temperature Rise of the System (17)

$$Q_b = N_{cell} \times I^2 \times R \times t \quad (15)$$

$$C_{total} = (m_{battery} \times c_{battery}) + (m_{aluminum} \times c_{aluminum}) \quad (16)$$

$$\Delta T = \frac{Q_b}{C_{total}} \quad (17)$$

$$T_T = T_0 + \Delta T \quad (18)$$

Where: N_{cell} is 28 (number of cells), I is the discharge current for each C-rate (as presented in Table 1), R is the internal resistance of the cells (0.005 Ω), t is the discharge

time (720 s), TT is total temperature, T0 is the initial temperature. For example, let us consider the M-B6 at a 0.5C discharge rate, the current is $I = 2$ A, C_{total} is the total heat capacity of the system, $m_{battery}$ and $m_{aluminum}$ is the mass of the battery and aluminum respectively, similarly $c_{battery}$ and $c_{aluminum}$ for specific heat capacity of battery and aluminum respectively. Therefore, the heat generated by the battery pack is:

$$Q_b(0.5c) = 28 \times (2)2 \times 0.005 \times 720 = 403.2J.$$

$$\Delta T = \frac{403.2}{2909.93} \approx 0.14^\circ C$$

$$T_r = 20 + 0.14 = 20.14$$

For the 1C discharge rate ($I = 4$ A):

$$Q_b(1c) = 28 \times (4)2 \times 0.005 \times 720 = 1612.8J.$$

$$\Delta T = \frac{1612.8}{2909.93} \approx 0.55^\circ C$$

$$T_r = 20 + 0.55 = 20.55$$

Similarly, the other values can be calculated in the same process and get 22.22°C, 28.86°C, 39.95°C, 55.47°C for 2C, 4C, 6C, 8C respectively.

RESULTS AND DISCUSSION

The testing was conducted at an ambient temperature of 20°C with various discharge C-rates, of at 0.5C, 1C, 2C, 4C, 6C, and 8C, rate currents to perform a comprehensive analysis on the thermal conditions of the battery pack arrangements. The experimental results were meticulously analyzed to assess several critical aspects of Temperature

Profiles and Voltage Profiles as presented in figures 4 and 5, Peak Temperatures Comparison presented in figure 6 thermal distribution also known as Heat Profile presented in figure 7 and at last the Correlation Matrix of Different Battery Pack Configurations presented in figure 8.

The distribution of thermal and electrical performance of different configurations is given in figures 4, 5 & 6 and table 4. Critical factors of battery performance have been analyzed for this paper and sensitivity analysis has been done to emphasize the best cooling strategies. From the following the models are represented as M-A (Model A), M-B1 (Model B1 (Air)), M-B2 (Model B2 (Paraffin Solid)), M-B3 (Model B3 (Paraffin Liquid)), M-B4 (Model B4 (Aluminum)), M-B5 (Model B5 (Al 6063-T83)), M-B6 (Model B6 (Al 3003-H18)), M-C (Model C (Aluminum in series)), M-D (Model D (Aluminum in parallel)).

Temperature Management and Temperature Profiles

Discharge rate has a great influence on the temperature regulation which is regarded as the key factor. Higher discharge rates cause the efficiency to be lower meaning that for all the configurations there is a greater temperature rise as seen in figure 4. The rate at which the technique cools the product is also an essential aspect. Comparing between the cooling methods with aluminum, this investigation reveals that M-B4, M-B5, and M-B6 exhibit higher thermal control in the cooling system than the methods using M-B1, M-B2 and M-B3 air and PCM (paraffin).

From observing the figures 4, 5(a-c) and table 4 we can determine the cooling efficiency of each model. At 0.5C discharge rate for M-A, M-B1, M-B2, M-B3, M-B4, M-B5, M-B6, M-C, and M-D, the peak temperature raises from 20°C (initial temperature) to 21.64°C, 21.34°C, 20.99°C,

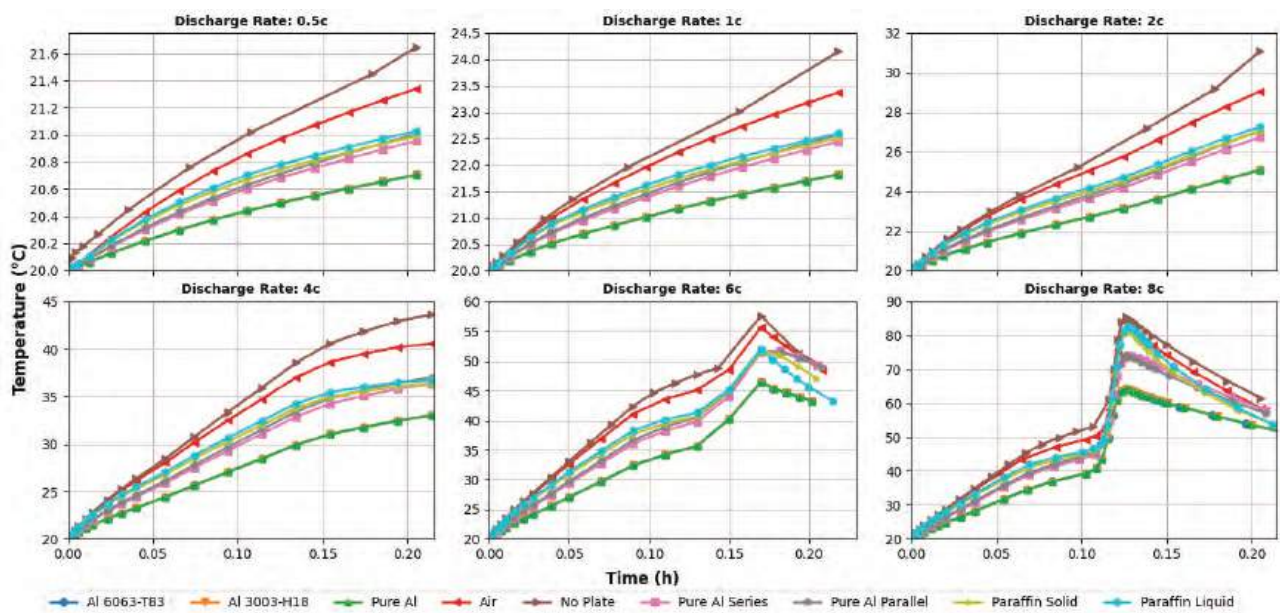


Figure 5. Temperature Profiles of Battery Pack Configurations Under Various Discharge Rates.

21.02°C, 20.7°C, 20.7°C, 20.7°C, 20.95°C, and 21°C respectively. Similarly, at 1C discharge rate, the peak temperature rises to 24.15°C, 23.37°C, 22.51°C, 22.59°C, 21.82°C, 21.82°C, 21.82°C, 22.44°C, and 22.57°C respectively. At 2C discharge rate, the peak temperature rises further to 31.08°C, 29.04°C, 27.02°C, 27.25°C, 25.07°C, 25.07°C, 25.06°C, 26.73°C, and 27.05°C. At 4C discharge rate, the peak temperatures increase significantly, reaching 43.64°C, 40.57°C, 36.35°C, 36.72°C, 33.02°C, 33.02°C, 33.01°C, 36.34°C, and 37.01°C respectively.

At 6C discharge rate, the peak temperatures rise to 57.6°C, 55.66°C, 51.62°C, 52.63°C, 46.45°C, 46.46°C, 46.45°C, 51.73°C, and 51.49°C at 0.2 hours during the 6C discharge, the temperatures stabilize at 49.11°C, 48.51°C, 47.12°C, 43.32°C, 43.33°C, 43.22°C, 43.3°C, 49.38°C, and 48.93°C respectively. At 8C discharge rate, the peak temperatures reach 85.36°C, 82.36°C, 81.37°C, 82.63°C, 64.23°C, 64.27°C, 64.31°C, 74.08°C, and 74.32°C. At time

0.2 hours during the 8C discharge, the temperatures stabilize at 61.55°C, 58.55°C, 54.14°C, 53.89°C, 51.38°C, 51.68°C, 51.37°C, 58.05°C, and 57.14°C respectively. From this, we observe that M-B4, M-B5, and M-B6 consistently exhibit the lowest peak temperatures across all discharge rates, indicating superior cooling efficiency.

Discharge rate has a strong significance, and voltage drop is more noticeable at higher discharge rates impacting all the tested settings. The impact of the cooling method is equally relevant since a better way of cooling will contribute to stabilizing of voltage due to minimized thermal decay.

From observing the voltage data across various models at different discharge rates, we can analyze the performance of each configuration. At a 0.5C discharge rate, the voltages for Models M-A, M-B1, M-B2, M-B3, M-B4, M-B5, M-B6, M-C, and M-D are recorded as 4.03V, 4.04V, 4.04V, 4.04V, 4.04V, 4.04V, 4.04V, 4.04V, and 4.04V, respectively. At 1C discharge, the voltages remain relatively stable, with values

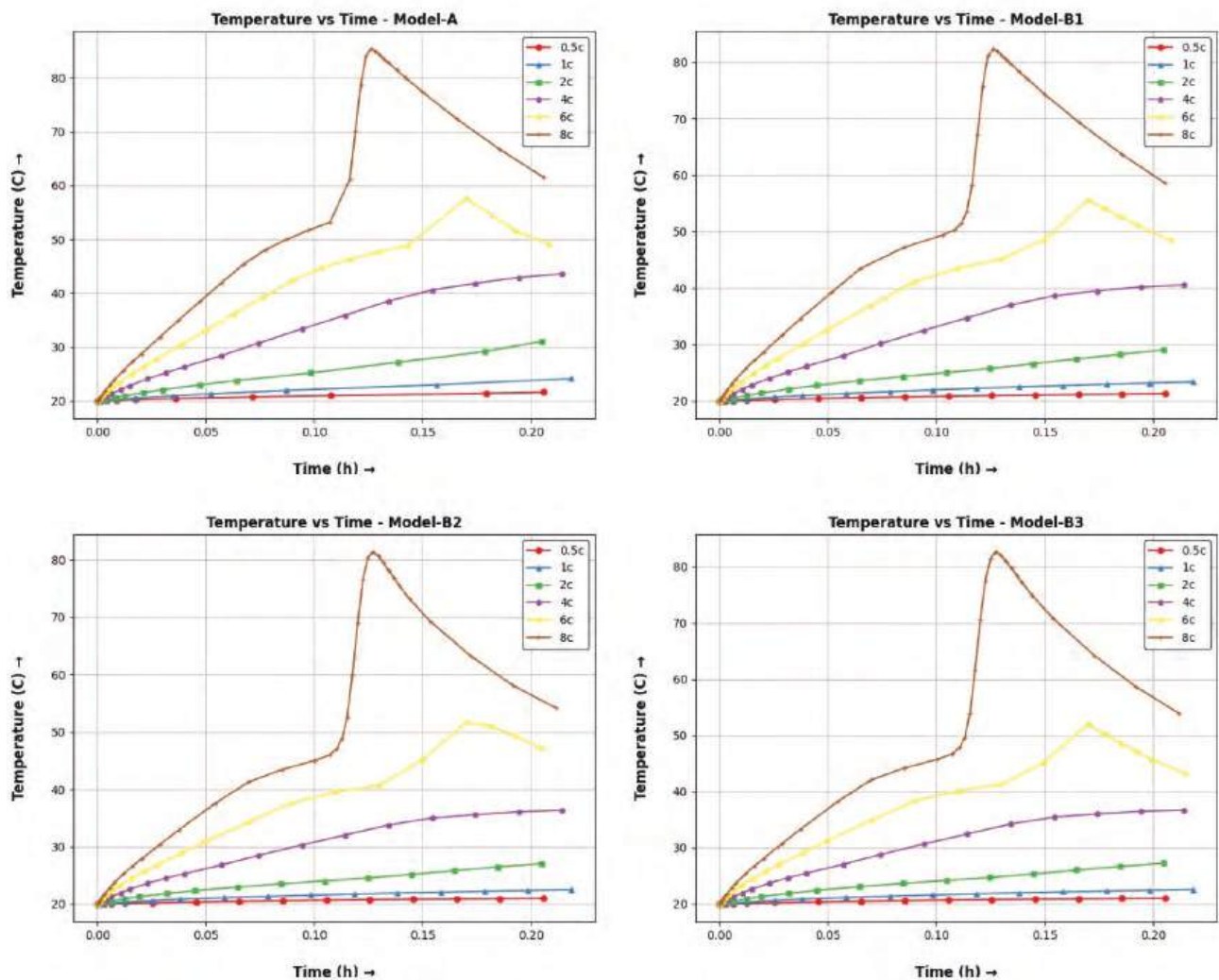


Figure 6(a). Peak Temperature Profiles of Model (M-A, M-B1, M-B2, and M-B3) Configuration Under Various Discharge Rates.

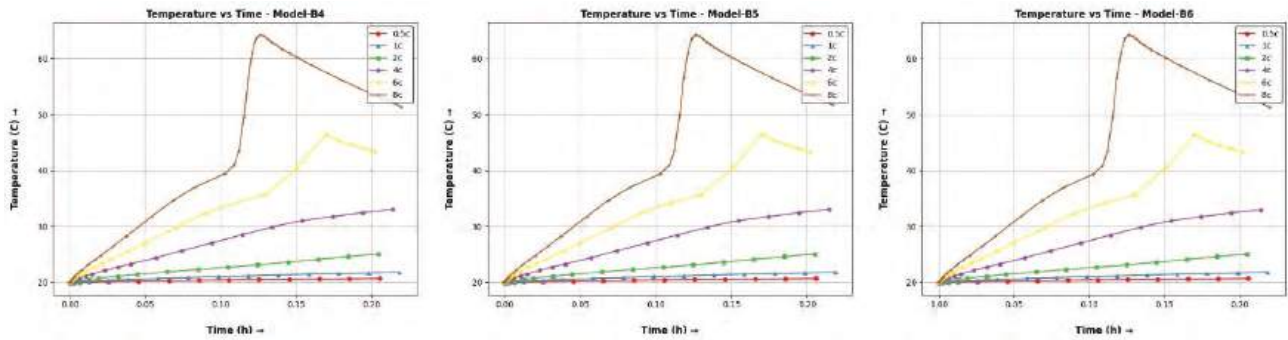


Figure 7(b). Peak Temperature Profiles of each Model (M-B4, M-B5, and M-B6) Configuration Under Various Discharge Rates.

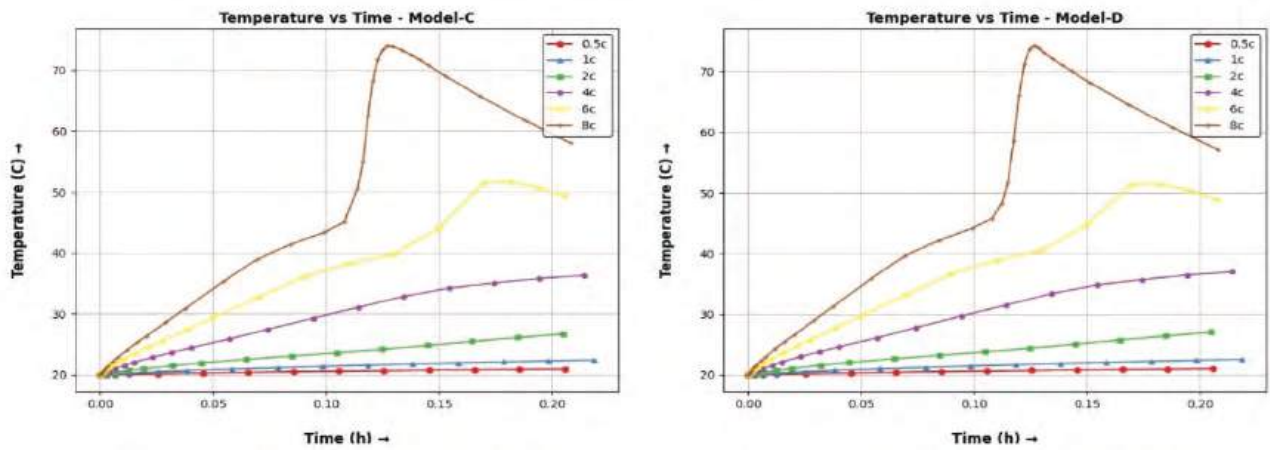


Figure 8(c). Peak Temperature Profiles of each Model (M-C and M-D) Configuration Under Various Discharge Rates.

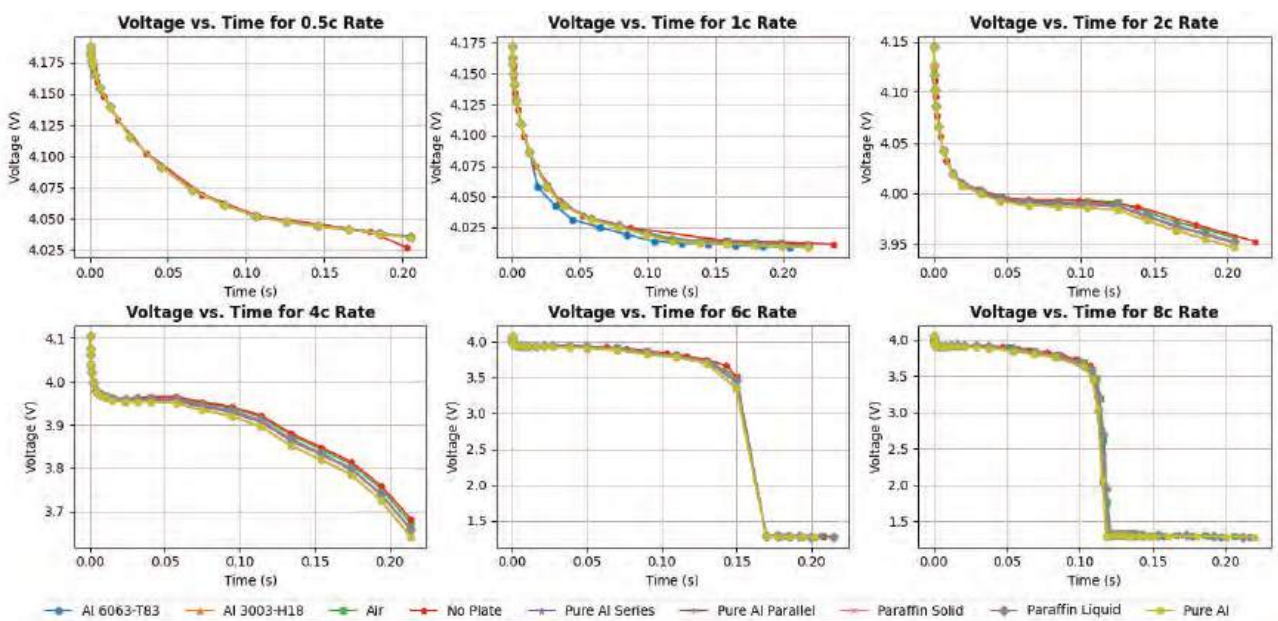


Figure 9. Voltage profiles per cell of Battery Pack Configurations Under Various Discharge Rates.

of 4.01V across most models, except for Model B4 through B6, which maintain the same voltage. When discharged at 2C, there is a slight drop in voltage, particularly for Models B1, B2, and B3, which exhibit voltages of 3.96V. The trend continues at 4C, where the voltages decrease further, with Model B4 reaching 3.64V, indicating a more significant impact of discharge rates on these models. At higher rates of 6C and 8C, voltages drop substantially, particularly for Model B6, which stabilizes at 1.27V at 6C and maintains 1.28V at 8C. This behavior suggests that Models B4, B5, and B6 consistently exhibit a higher capacity to retain voltage across varying discharge rates, indicating better performance and efficiency in voltage retention during operation.

Therefore, considering the thermal and electrical results, it is possible to state that M-B6 (Al 3003-H18) has the best results in temperature dissipation and voltage regulation at all the discharge current rates. A similar success is achieved by M-D (Aluminum in Parallel); it is optimal about the higher charge/discharge rates thus fitting well high-end functionalities.

The bar chart in figure 7 gives the direct comparison of peak temperatures at a given configuration at different discharge rates. This supports the observation made in Figure 4 and 5(a-c) with the help of Table 4 indicating that all techniques based on aluminum gives a lower peak temperature.

Thermal Distribution (Heat Profile)

The thermal distribution shown in Figure 7 is in the different battery pack configuration models M-A, M-B(1-6), M-C, and M-D at different discharge rates of 0.5C, 1C, 2C, 4C, 6C, and 8C. Each row represents a different cooling technique: The researched cooling types include no plate (M-A), air cooling (M-B1), paraffin solid and paraffin liquid cooling (M-B2 and M-B3), pure aluminum cooling (M-B4), aluminum variants (M-B5 and M-B6), and aluminum configurations in series and parallel (M-C and M-D).

The thermal maps reveal that M-A tends to exhibit a high degree of temperature variations, especially at a higher rate of discharge, which proves that the thermal regulation of this model is inadequate. The cooling techniques that use

Table 4. Comparison of Cooling Techniques for Battery Cells at Various Discharge Rates

Configuration	Parameter	0.5C	1C	2C	4C	6C	8C
Model A	Voltage at 0.2h (V)	4.03	4.01	3.96	3.68	1.29	1.29
	Temperature at 0.2h (°C)	21.64	24.15	31.08	43.64	49.11	61.55
	Peak Temperature Recorded (°C)	21.64	24.15	31.08	43.64	57.6	85.36
Model B1 (Air)	Voltage at 0.2h (V)	4.04	4.01	3.96	3.67	1.29	1.29
	Temperature at 0.2h (°C)	21.34	23.37	29.04	40.57	48.51	58.55
	Peak Temperature Recorded (°C)	21.34	23.37	29.04	40.57	55.66	82.36
Model B2 (Paraffin Solid)	Voltage at 0.2h (V)	4.04	4.01	3.95	3.66	1.28	1.29
	Temperature at 0.2h (°C)	20.99	22.51	27.02	36.35	47.12	54.14
	Peak Temperature Recorded (°C)	20.99	22.51	27.02	36.35	51.62	81.37
Model B3 (Paraffin Liquid)	Voltage at 0.2h (V)	4.04	4.01	3.95	3.66	1.28	1.29
	Temperature at 0.2h (°C)	21.02	22.59	27.25	36.72	43.32	53.89
	Peak Temperature Recorded (°C)	21.02	22.59	27.25	36.73	52.63	82.63
Model B4 (Aluminum)	Voltage at 0.2h (V)	4.04	4.01	3.95	3.64	1.27	1.28
	Temperature at 0.2h (°C)	20.7	21.82	25.07	33.02	43.33	51.38
	Peak Temperature Recorded (°C)	20.7	21.82	25.07	33.02	46.45	64.23
Model B5 (Al 6063-T83)	Voltage at 0.2h (V)	4.04	4.01	3.95	3.64	1.27	1.28
	Temperature at 0.2h (°C)	20.7	21.82	25.07	33.02	43.32	51.68
	Peak Temperature Recorded (°C)	20.7	21.82	25.07	33.02	46.46	64.27
Model B6 (Al 3003-H18)	Voltage at 0.2h (V)	4.04	4.01	3.95	3.64	1.27	1.28
	Temperature at 0.2h (°C)	20.7	21.82	25.06	33.01	43.3	51.37
	Peak Temperature Recorded (°C)	20.7	21.81	25.06	33.01	46.45	64.31
Model C (Aluminum in series)	Voltage at 0.2h (V)	4.04	4.01	3.95	3.66	1.29	1.29
	Temperature at 0.2h (°C)	20.95	22.44	26.73	36.34	49.38	58.05
	Peak Temperature Recorded (°C)	20.95	22.44	26.73	36.34	51.73	74.08
Model D (Aluminum in parallel)	Voltage at 0.2h (V)	4.04	4.01	3.95	3.66	1.29	1.29
	Temperature at 0.2h (°C)	21	22.57	27.05	37.01	48.93	57.14
	Peak Temperature Recorded (°C)	21	22.57	27.05	37.01	51.49	74.32

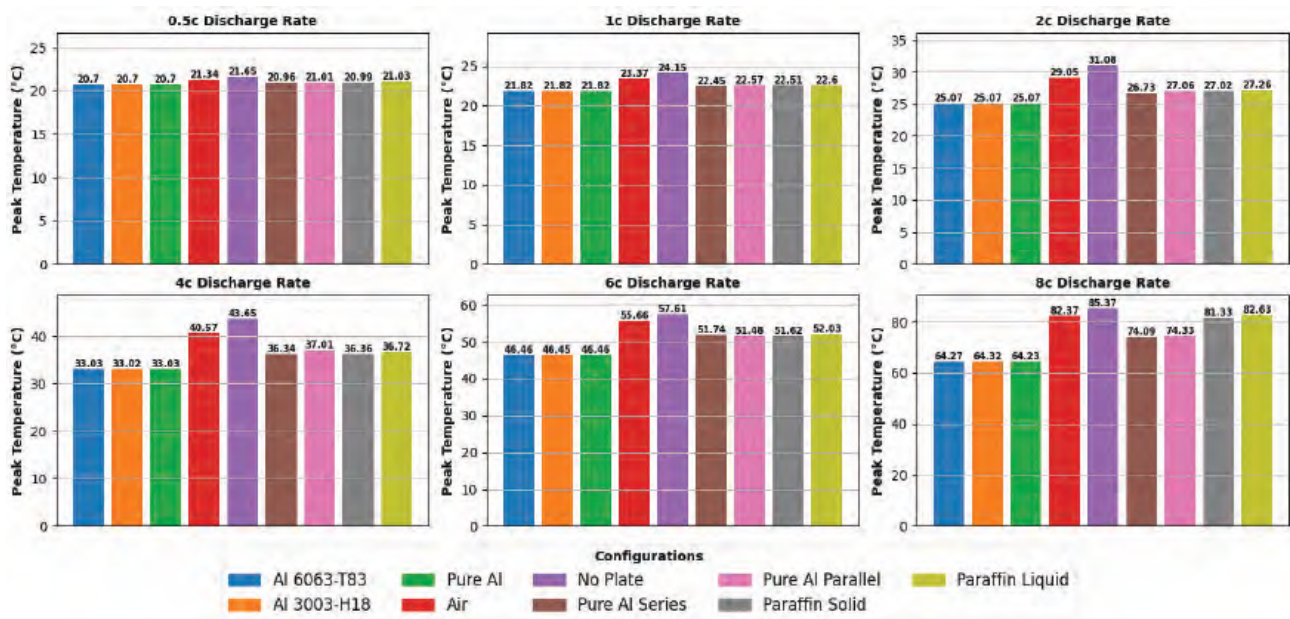


Figure 10. Comparison of Peak Temperatures for Battery Pack Configurations Across Different Discharge Rates.

air and paraffin (M-B1, M-B2 and M-B3) are also moderate effective at reducing the temperature, yet at high discharge rates; hot zones are still observed.

In the case of pure aluminum as well as the various aluminum derivatives (M-B4, M-B5, and M-B6), the heat dissipation is more evenly distributed and heat issues are not apparent even when operating at high discharge rates. The aluminum in the series and parallel (M-C and M-D) has the best thermal dissipation as it has the least isothermal changes across all rates of discharge. This figure for the most part conveys the information that aluminum-based cooling methods serve the best in heat removal and thermal regulation of battery packs.

Correlation Analysis for Various Discharge Rates

Correlation analysis proves to be an invaluable tool for both predictive accuracy and real-time application in

industries like aerospace, where thermal management is critical. It offers valuable insights for material selection and performance prediction, helping to optimize thermal management techniques. The strength of the correlations between the models and materials across various discharge rates (0.5C to 8C) provides insights into how closely related the performance of different thermal management systems are under similar stress conditions.

The correlation matrices shown in Figure 8(a) to 8(c), depicts the correlation values of various measurements for different discharge rates (0.5C, 1C, 2C, 4C, 6C, 8C). Each matrix is a 2 by 2 matrix, in which cells are the correlation coefficients between the two M-A and M-B (1-6), M-C and M-D. The numbers and color gradient alongside them represent the level of correlation, with numbers close to 1. While the closer to 1 means a positive direction, values

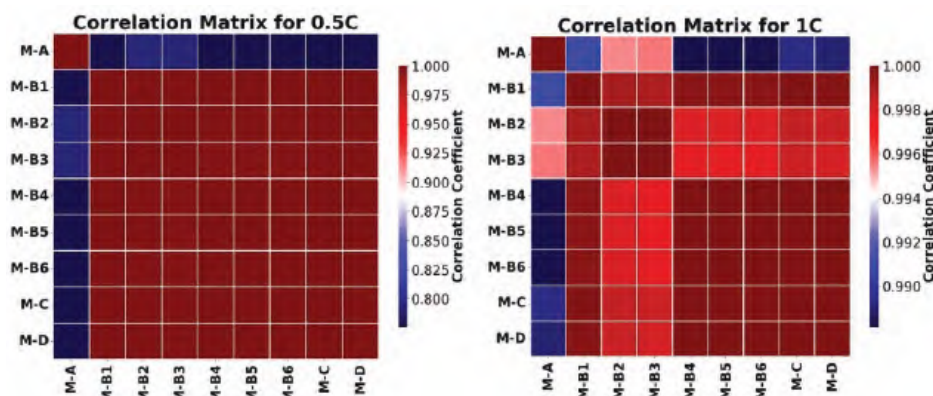


Figure 11(a). Correlation matrix of different models at 0.5C and 1C discharge rate.

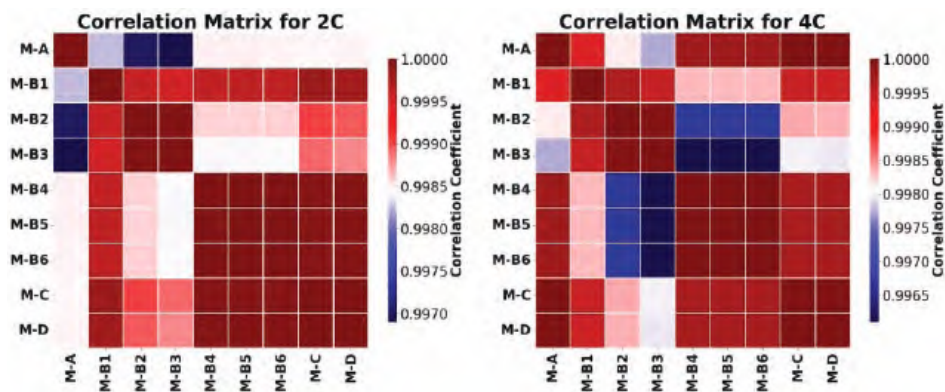


Figure 12(b). Correlation matrix of different models at 2C and 4C discharge rate.

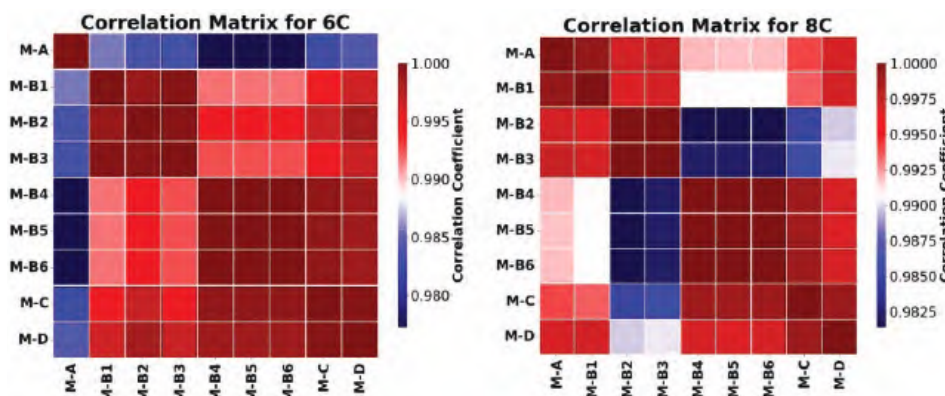


Figure 13(c). Correlation matrix of different models at 6C and 8C discharge rate.

closer to 0 mean negative direction and hence, 0 supports a strong positive direction. The correlation between these is -1 implying a negative correlation where values often give 0 it implies that there is no correlation.

Red represents strong positive correlations (closer to +1): Shows that one measurement has variation in direct proportion with the other. Blue represents strong negative correlations (closer to -1): Expresses that if the first variable is large, the second one is least and vice versa. No Correlation (close to 0): Used when there is no correlation between the measurements. At the 0.

In smaller aircraft like Piper, Cessna, and Cirrus, efficient thermal management is crucial for optimizing battery systems and overall performance. The correlation analysis reveals that aluminum and its alloys (M-B4, M-B5, M-B6, M-C, M-D) exhibit high consistency in thermal conductivity across all discharge rates, making them ideal for lightweight aircraft where minimizing heat buildup is essential. Paraffin-based Phase Change Materials (M-B2, M-B3) also show strong reliability, offering excellent thermal storage capabilities. For these aircraft, combining highly conductive materials like aluminum with PCMs could significantly improve cooling efficiency, ensuring stable performance, and extending the lifespan of

energy systems in compact, high-demand environments. These insights align with current advancements in lightweight, efficient thermal management technologies used in modern aviation, potentially enhancing safety and energy efficiency in smaller aircraft systems.

Discharge rate of 0.5C, the results of the correlation matrix reveal that the models have different levels of positive correlation. It is also observed that there is moderate positive relationship between Model A (M-A) and other models wherein the weakest association was detected and the lowest with Model M-B2 at 0.7929. The models of aluminum and its alloys (M-B4, M-B5, M-B6, M-C, M-D) are very closely related with each other approaches the maximum value 1.0. This can be interpreted to mean that the thermal and electrical characteristics of these models are quite close when it comes to this given discharge rate.

Concerning the 1C discharge rates, the correlation coefficients trend is characterized by the increase of the figures, which shows the growing dependence of the models. These respective correlations range from being very strongly positively related as M-A with M-B1 at 0.9908. The models based on aluminum (M-B4, M-B5, M-B6, M-C, M-D) again and again depict close correlation, in some cases even arriving at

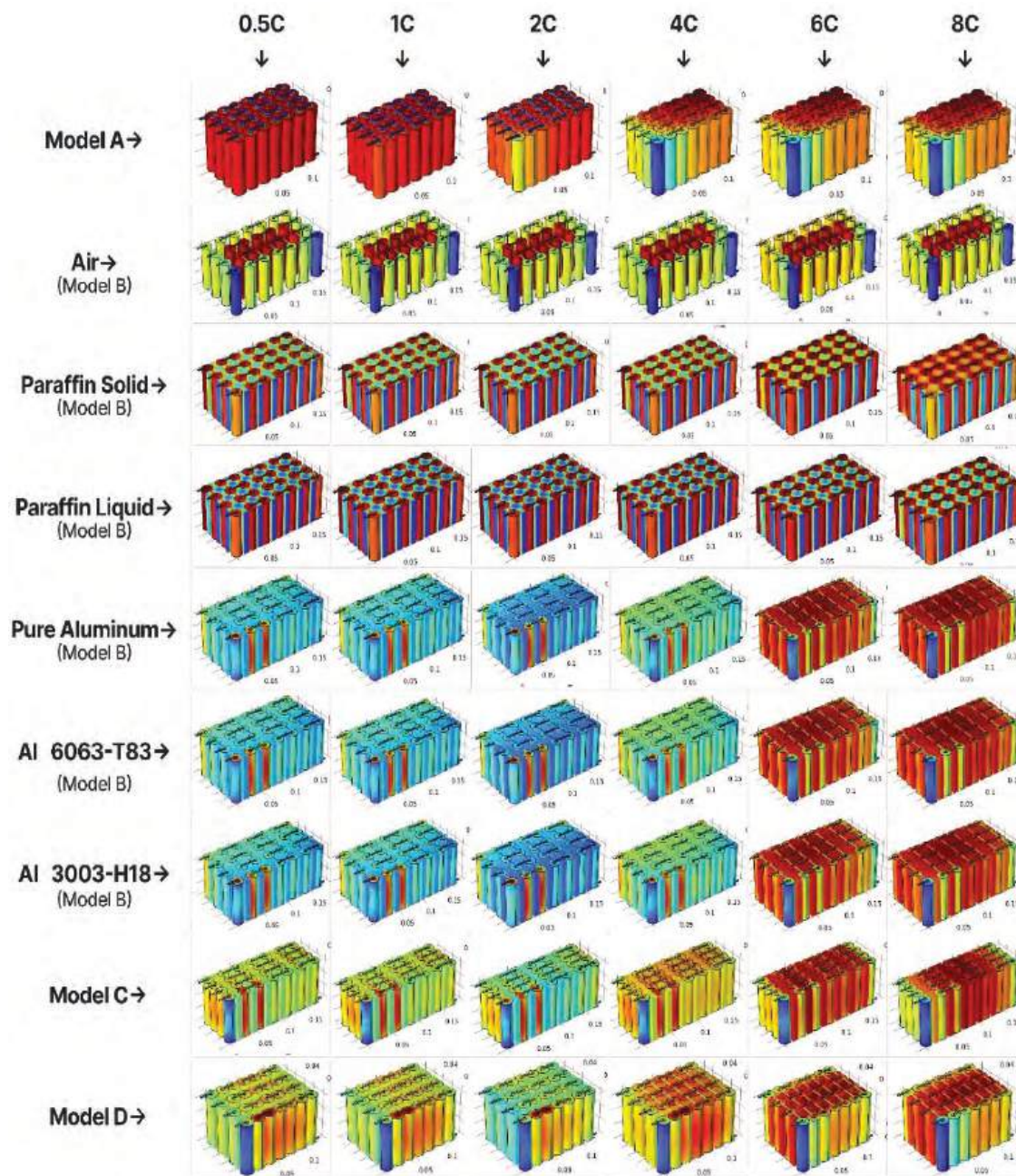


Figure 14. Thermal distribution in battery pack models A, B, C, and D across various discharge rates

the level of 0.999. Such a high correlation between the two models implies that the models' response to the thermal stresses is similar at this higher discharge rate.

At a 2C discharge rate, the correlation coefficients between the models are also considered to be high. This indicates that, Model A (M-A) has positive relationship with other models particularly Model B1 (M-B1) at 0.9982. All the models with paraffin and aluminum (M-B2, M-B3, M-B4, M-B5, M-B6, M-C, M-D) have the similar and high correlation coefficients, which gives evidence that the efficiency of the thermal management stratagem planning with the increased discharge condition is also satisfactory.

The same is also true when the discharge rate has been raised to 4C; the correlation matrix shows that M-A correlates with other models like M-B1 with an indication of 0.9993. The aluminum and its alloy models (M-B4, M-B5, M-B6, M-C, M-D) almost have 1:1 correlation coefficient, which depicts that these model's behavior is quite predictable and almost similar under this higher discharge stress.

Further, at the 6C discharge rate the association between different models reveal insignificant decline particularly between the M-A model and the others and the lowest value was recorded with M-B1 model as 0.9857. Nevertheless, considering their results under this discharge

rate, it is evident that the correlations in the thermal and electrical properties of the aluminum-based models (M-B4, M-B5, M-B6, M-C and M-D) are still very high and therefore; a strong positive relationship is still evident.

The correlation matrix shows very high correlation for most of the models for the level contributing to the highest discharge rate of 8C. M-A has a correlation coefficient of 0.9993 with M-B1. The same situation can be observed with aluminum and alloy models M-B4, M-B5, M-B6, M-B, M-D, with near-perfect coefficients which remained stable and prove the efficiency of the corresponding thermal management schemes at the maximum tested discharge rate as well.

M-A, Correlations range from 0.7759 to 0.9993 were reported. A striking feature of M-B1 is demonstrated to have very higher correlation coefficients with the other models that exceeds 0.990. M-B2 correlates almost perfectly with another aluminum-based model which shows that its performance is almost similar. M-B3 resembles M-B2 in terms of very high correlation. This study found out that M-B4 has a high correlation with other aluminum models in most models established constantly high correlations.

Frequently, the M-B5 correlation coefficients approach 1.0, indicating predictable performance. M-B6 is generally very high affinity with other models especially the aluminum-based models. Both M-C have a high correlation with other aluminum models particularly when the discharge rates are high. The values for M-D lay near one, meaning good performance and similarity between the model and other models.

CONCLUSION

Among all the cooling techniques, aluminum-based heat spreaders demonstrated superior thermal performance, with Al 3003-H18 showing the most effective results. At an 8C discharge rate, this model achieved a maximum temperature of 57.22°C, outperforming air cooling (82.62°C) and standard aluminum (64.23°C). Also, Al heat spreaders enhanced voltage regulation depending on the discharge current. For instance, that voltage drop for Al 3003-H18 was small, that retained 4.04V, 4.01V, 3.95V, 3.64V, and 1.27V at discharge rates from 0.5C to 8C, which pointed to the decreased thermal induced degradation. The positive correlation values of above all aluminum-based models briefed that they are showing consistent results and are very reliable when tested under different thermal conditions which confirm their use for aviation.

Air cooling, being easy and cheap, proved to have a low efficiency and further declined at high discharge rates (82.36°C) at 8C. This makes it less suitable for high-performance operations such as aerodynamics, thermals and other critical aviation uses. Paraffin solid and liquid (Phase Change Materials) provided moderate enhancements over passive air cooling at the same time. However, their peak temperatures at 8C (54.14°C and 53.89°C) were higher than the aluminum-based methods; thus, it can be deduced that

the proposed techniques are more suitable for moderate thermal management but not at very high stresses.

Aluminum, especially Al 3003-H18, proved to be the best material for effective heat dissipation and maintaining uniform temperature distribution. The quantitative results, maintaining steady-state temperatures at 0.2 hours, 20.7°C, 21.82°C, 25.06°C, 33.01°C, 43.3°C, and 51.37°C at discharge rates from 0.5C to 8C, highlight aluminum's superior thermal conductivity and stability. The quantitative results, such as, coupled with minimal voltage drops, highlight aluminum's high thermal conductivity and stability. This makes it an ideal choice for high-performance and safety-critical applications in aviation.

Thus, assessment of this study's findings offers considerable value for the realization and monetization of effective thermal management systems in airborne applications. The obtained results indicate that the use of the aluminum-based cooling has a higher cooling effectiveness and therefore integrating the right kind of cooling into battery management systems will expand the safety, efficiency, and service time of lithium-ion batteries. As it could be observed from the aircraft manufacturers and operators' point of view, the described cooling techniques may lead to the improvement and durability of battery systems meeting the current load levels and operating conditions of modern aviation.

NOMENCLATURE

General Parameters

SOC	State of Charge
T	Temperature
t	Time (s)
x	Spatial coordinate (m)
r	Radial coordinate in spherical (m)
ρ	Density (kg/m ³)
c_p	Specific heat capacity at constant pressure (J/kg·K)
k	Thermal conductivity (W/m·K)
μ	Velocity vector (m/s)
τ	Characteristic time (s)

Electrical and Electrochemical Parameters

E_{cell}	Cell voltage (V)
$E_{OCV(SOC,T)}$	Open Circuit Voltage (V)
η_{IR}	Overpotential due to internal resistance (V)
η_{act}	Activation overpotential (V)
I_{cell}	Current through the cell (A)
I_{1C}	Current corresponding to a 1C rate (A)
$Q_{cell,0}$	Nominal capacity of the battery cell (Ah)

Thermal Parameters

Q_h	Total heat generation within the cell (W/m ³)
Q_{mix}	Heat generation due to mixing or diffusion (W/m ³)
$E_{OCV,therm}$	Thermal component of the open circuit voltage (V)

$E_{OCV,ref}$	Reference OCV at a reference temperature (V)
T_{ref}	Reference temperature (C)

Other Symbols

∇	Nabla operator (vector differential operator)
$\cdot$	Dot product
Ω	Domain of integration
d_{vol}	Differential volume element
k_s	Shell's thermal conductivity (W/m·K)
h_{nat}	Natural convective heat transfer coefficient of air (W/m ² ·K)
T_s	Temperature of the shell (K)
T_a	Ambient temperature (K)
n	Direction normal (wall-normal direction)

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Lithium-ion aircraft batteries as a smoke/fire risk. SKYbrary Aviation Safety. Jul 24, 2024. Available at: <https://skybrary.aero/articles/lithium-ion-aircraft-batteries-smokefire-risk> Accessed on Sep 04, 2025
- [2] Lithium-ion batteries for aerospace applications. Nanografi Nano Technology. Available at: <https://nanografi.com/blog/lithiumion-batteries-for-aerospace-applications/> Accessed on Sep 04, 2025
- [3] Special conditions: Boeing Model 777-200, -300, and -300ER series airplanes; rechargeable lithium-ion batteries and battery systems. Fed Regist. Dec 19, 2013. Available at: <https://www.federal-register.gov/documents/2013/12/19/2013-30232/>
- [4] Lithium-ion batteries. KADEx Aero Supply. Dec 23, 2021. Available at: <https://kadexaero.com/product-details/advanced-lithium-ion-main-batteries/> Accessed on Sep 04, 2025
- [5] Gill 7242-16 LT series 24-volt/16Ah super capacity sealed lead acid aircraft battery. SkyGeek. Available at: <https://skygeek.com/gill-7242-16-aircraft-battery.html> Accessed on Sep 04, 2025
- [6] Gill 7243-16 LT sealed lead acid battery. Aircraft Spruce. Available at: <https://www.aircraftspruce.com/catalog/elpages/gill7243battery11-12210.php> Accessed on Sep 04, 2025
- [7] COMSOL Inc. COMSOL Multiphysics reference manual, version 5.6. Burlington: COMSOL Inc.; 2020. Available at: file:///C:/Users/mmrun/Downloads/COMSOL_ReferenceManual.pdf Accessed on Sep 04, 2025
- [8] Liu S, Zhang G, Wang CY. Challenges and innovations of lithium-ion battery thermal management under extreme conditions: a review. ASME J Heat Mass Transfer 2023;145:082301. [CrossRef]
- [9] Figueiras I, Coutinho M, Afonso F, Suleman A. On the study of thermal-propulsive systems for regional aircraft. Aerospace 2023;10:113. [CrossRef]
- [10] Coutinho M, Afonso F, Souza A, Bento D, Gandolfi R, Barbosa FR, et al. A study on thermal management systems for hybrid–electric aircraft. Aerospace 2023;10:745. [CrossRef]
- [11] Capron O, Samba A, Omar N, Gualous H, Bossche P, Mierlo J. Large and high-power cylindrical batteries: analysis of the battery packs temperature distributions using COMSOL Multiphysics and MATLAB simulation softwares. 2014. Available from: <https://www.semanticscholar.org/paper/Large-and-High-Power-Cylindrical-Batteries-Analysis-Capron-Samba/7a05ff0820328120ae9cb6500aac7fc6ec6a966d> Accessed on Sep 04, 2025
- [12] Shetty DD, Venugopal V, PR, Zuber M, Badruddin IA, Kini C. Computational flow analysis of different streamline cooling plates for thermal management of lithium-ion battery. Cogent Eng 2022;9:2048996. [CrossRef]
- [13] Zhu Y, Li K, Kang E, Quan T, Sun T, Luo J, et al. Simulation investigation on thermal characteristics of thermal battery activation process based on COMSOL. Crystals 2023;13:641. [CrossRef]
- [14] Zhao Y, Chen J, He W. Design and performance evaluation of liquid-cooled heat dissipation structure for lithium battery module. Processes 2023;11:1769. [CrossRef]
- [15] Dunning J, Mackin T, Rozsnyo R. Heat generation modeling of a lithium battery: from the cell, to the pack on COMSOL Multiphysics. 2015. Available from: <https://>

- www.semanticscholar.org/paper/Heat-Generation-Modeling-of-a-Lithium-Battery%3A-from-Dunning-Mackin/fc0ff5f40f7fc71b69fc906dec8ea2295b76e99a Accessed on Sep 04, 2025.
- [16] Sun J, Dan D, Wei M, Cai S, Zhao Y, Wright E. Pack-level modeling and thermal analysis of a battery thermal management system with phase change materials and liquid cooling. *Energies* 2023;16:5815. [\[CrossRef\]](#)
- [17] Satheesh VK, et al. Enhancement in air-cooling of lithium-ion battery packs using tapered airflow duct. *J Therm Eng* 2024;10:375–385. [\[CrossRef\]](#)
- [18] Gharde PR, Havaladar SN. Numerical investigation of an amalgamation of two-phase change materials thermal energy storage system. *J Therm Eng* 2024;10:263–272. [\[CrossRef\]](#)
- [19] Gungor S, Gocmen S, Cetkin E. A review on battery thermal management strategies in lithium-ion and post-lithium batteries for electric vehicles. *J Therm Eng* 2023;9:1078–1099. [\[CrossRef\]](#)
- [20] Palmieri B, Cilento F, Siviello C, Bertocchi F, Giordano M, Martone A. Mitigation of heat propagation in a battery pack by interstitial graphite nanoplatelet layer: coupled electrochemical-heat transfer model. *J Compos Sci* 2022;6:296. [\[CrossRef\]](#)
- [21] COMSOL Inc. Introduction to battery design module. COMSOL Multiphysics Documentation. 2021. Available at: <https://doc.comsol.com/6.1/doc/com.comsol.help.battery/IntroductionToBatteryDesignModule.pdf> Accessed on Sep 04, 2025.



Research Article

Comparative performance analysis of the Andasol-1 power plant in M'sila (an Algerian province)

Khaled BOUCHAREB^{1,2}, Nabila IHADDADENE^{1,3,*}, Razika IHADDADENE^{1,3},
Khellaf BELKHIRI¹

¹Department of Mechanical Engineering, M'Sila University, M'Sila, 28000, Algeria

²Laboratory of Materials and Mechanics of Structure L.M.M.S, University of M'Sila, M'Sila, 28000, Algeria

³Laboratory of Water, Environment and Renewable Energies Laboratory, Med Boudiaf University, M'Sila, 28000, Algeria

ARTICLE INFO

Article history

Received: 07 September 2024

Accepted: 10 December 2024

Keywords:

Andasol-1 Power Plant; Climatic Conditions; Parabolic Trough Concentrated Solar Power; M'Sila; Performance Assessment; SAM Software

ABSTRACT

Within the field of environmentally friendly electricity generation, parabolic trough concentrated solar power (CSP-PT) technology is acknowledged as one of the most efficient and practicable options. The current research employs the System Advisor Model (SAM) software to assess the viability of establishing a CSP facility utilizing parabolic trough technology, akin to the Andasol-1 power plant in southern Spain, within the M'Sila region of northern Algeria. It also statistically identifies the meteorological factors that most significantly influence Andasol-1's performance in M'Sila and models its operation based on them. Furthermore, it compares Andasol-1's performance across a range of locations reported in the literature. When the Andasol-1 power plant with a capacity of 50 MW is installed in M'Sila, it generates electricity year-round from 11:00 a.m. to 5:00 p.m., with an average output ranging from 25 MW_e to 52 MW_e. It can reach 17 hours of production, or even more, from March to September due to the favorable weather conditions and energy storage system (TES). The electricity generation increases with Direct Normal Irradiance (DNI) and ambient temperature, while it decreases with relative humidity. These variables collectively explain 98.2% of the electricity production variance in M'Sila ($R^2 = 0.982$), underscoring the significance of the linear regression model proposed. When Andasol-1 is erected in Kuwait, Tataouine (Tunisia), or at its true site (Spain), it produces less electricity annually than M'Sila, Tajoura (Libya), and Ma'an (Jordan). Furthermore, M'Sila's climate is the second most favorable for Andasol-1, after Jordan's, since it can generate 177.22 GWh of power annually there, exceeding its actual location. Climate change allows for the installation of solar power plants in northern regions as well.

Cite this article as: Bouchareb K, Ihaddadene N, Ihaddadene R, Belkhiri K. Comparative performance analysis of the Andasol-1 power plant in M'sila (an Algerian province). J Ther Eng 2025;11(5):1439–1454.

*Corresponding author.

*E-mail address: nabila.ihaddadene@univ-msila.dz

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkiic



INTRODUCTION

In the environmentally friendly electricity generation area, concentrated solar power (CSP) technologies are recognized as one of the best and most practical alternatives [1]. Indeed, CSP technologies offer the advantage of incorporating a thermal energy storage system (TES), which may store excess heat energy generated during daytime overproduction for later usage when the sun is not shining [2], support grid stability, and secure dispatchability for on-demand electrical power, unlike other renewable energy technologies, such as wind power and photovoltaics [3].

The four CSP technologies that are currently on the market are the solar parabolic dish (SPD), linear Fresnel reflector (LFR), solar power tower, and parabolic trough collector (PTC) [4]. This latter technology is one of the most advanced and extensively used today. Indeed, when compared to other solar energy conversion technologies, the parabolic trough represents 76.6% of the overall market [5]. In addition, compared to solar towers and linear Fresnel technologies, parabolic trough technology is expected to maintain its dominance in the market [6,7,8,9]. Additionally, it has the ability to generate electricity either independently or in combination with conventional energy technologies at the lowest Levelized Cost of Electricity (LCOE) [10]. Moreover, parabolic trough collectors (PTCs) have also demonstrated their effectiveness in the field of water desalination [11,12].

Solar radiation is concentrated on an absorber tube in a parabolic trough collector (PTC) by large, curved mirrors in the shape of a U. Parabolic trough collectors are designed to track the sun along one axis, from east to west when they are aligned on the north-south axis, and from north to south when they are aligned on the east-west axis. Solar radiation heats a heat transfer fluid (HTF) running through the absorber tube, typically synthetic oil or water/steam. The thermal energy gathered in this way is transferred to a standard steam turbine power cycle to produce electricity.

With over 62 active projects, PTC is a well-established technology. Numerous theoretical studies have been published in the literature on the feasibility of integrating CSP based on parabolic collectors in various parts of the world. Following is a summary of some works on this subject.

Ismail A. et al. [13] investigated the feasibility of installing concentrated solar power (CSP) plants in Libya's capital (Tripoli) through an energetic, exergetic, and economic thermodynamic analysis. They found that installing the Andasol parabolic trough CSP plant in Tripoli provided a very positive return on investment, higher performance, and a decreased levelized cost of electricity (LCOE). Attai Y.A. et al. [14] compared a concentrated solar power plant to be installed in Aswan, South Egypt, to Palma del Rio II, a reference, to determine the relative benefit of constructing such a power plant in Aswan. The results obtained by the system advisor model (SAM) software show that Aswan produces electrical energy at a lower cost than Spain, and its

annual CSP energy output is higher as well. Sultan A. J. et al. [15] evaluated the techno-economic performance of a parabolic trough solar power plant in Kuwait (desert climatic conditions and limited water supplies) and compared it to that of the reference plant in southern Spain (Andasol-1) for two design configurations employing wet and dry cooling options. The outcomes demonstrate that Kuwait outperformed Spain in terms of overall performance. Due to the DNI influence, ambient temperature, wind speed, and solar field heat loss/dumped energy, the Kuwait case performed better. Jilani N. A. et al. [16] conducted a techno-economic analysis of two CSP technologies—the parabolic trough collector (PTC) and the solar power tower (SPT)—in preparation for possible installation in Malaysia using two reference projects, the Andasol-1, and PS-10 systems in Spain, as references. The annual electricity generation and the initial cost are two critical aspects influencing the construction, installation, and deployment of PTC or SPT technology. Shagdar E. et al. [17] performed a numerical analysis both at nominal and part-load conditions of the 50 MW CSP's performance for four different reference days (i.e., March 22, June 22, September 22, and December 22), which correspond to the four seasons of the Mongolian climate. The findings indicate that solar radiation and solar incidence angle have a significant impact on the CSP plant's operating performance. The reference days outside of the winter constitute the best time with the highest operating performance for the 50 MW CSP plant. Additionally, the CSP plant's operating performance at part-load levels was noticeably lower than at nominal loads. Based on a reference plant design from Andasol-1 in southern Spain, Liqreina et al. [18] evaluated the competitiveness of dry cooling of steam power blocks of concentrating solar power (CSP) plants in comparison to wet cooling at the Ma'an site (desert zone) in southern Jordan. Although the cost of dry cooling is higher than that of wet cooling, dry cooled CSP plants are still a possibility in hot, arid areas with excellent solar resource, like Ma'an, and can even achieve lower costs than comparable wet-cooled plants situated in areas with less solar resource. Trabelsi et al. [19] investigated the influence of using molten salt as heat transfer fluid (HTF) in conjunction with dry cooling in a parabolic trough CSP plant at Tataouine, Tunisia, to significantly reduce the levelized cost of electricity (LCOE) and water use. In parabolic trough power plants, the results obtained indicate that molten salt is more economically advantageous than synthetic oil.

The feasibility of constructing solar power facilities in Algeria, like in many other countries, has been theoretically investigated, particularly in the country's desert region. Indeed, Achour et al. [20] investigated the performance of an Integrated Solar Combined Cycle (ISCC) plant under the southern Algerian climate through a developed thermodynamic model to assess solar radiation intensity as well as the overall performance of the hybrid solar power plant. It has been found that the amount of electricity

produced increases with several operation parameters, such as the time of day, the mass flow rate of the heat transfer fluid, and the solar incidence angle on the collector surface. Benhaji et al. [21] performed a theoretical investigation using the System Advisor Model software (SAM) to evaluate and model a 100 MW parabolic trough-based solar thermal power plant with two different cooling systems: dry and evaporative condensers and thermal storage at Tamanrasset, southern Algeria. When comparing the power plant's electricity output to Tamanrasset's actual load demand, it was found that the plant will be able to supply 78% and 60% of the town's electrical demands throughout the winter and summer, respectively. Additionally, it was demonstrated that, compared to a normal electric generation system, the solar power plant may reduce carbon dioxide gas emissions by 2 million tons annually. With the aid of the System Advisor Model software, Ikhlef K. et al. [22] conducted a techno-economic evaluation of a 155 MW solar power plant (a 25 MW parabolic trough connected with a 130 MW gas turbine) known as the Solar Power Plant One (SPPI) located in Hassi R'mel (Algeria's southern region). They found that in the southern Algeria environment, wet cooling was more cost-effective than dry cooling and that Therminol VP1 was the optimal heat transfer fluid. Additionally, the best period for storage system optimization was four hours. Furthermore, integrating the backup system with the power plant increased annual energy output by 11.2% compared to the only solar field. Benabdellah H. M. et al. [23] carried out a techno-economic analysis of an Integrated Solar Combined Cycle (ISCC) power plant based on a Parabolic Trough Concentrator (PTC) system in Hassi R'mel (an Algerian Sahara region) with a new thermal storage system. They found considerable improvements in the efficiency of the power plant and the conversion of solar energy (up to 14%). The improved ISCC-PTC power plant at this location will enable savings of around 30 million dollars in natural gas consumption and 13 million dollars in emission taxes.

The current climate change (earth warming and water shortage) felt by earth's inhabitants (particularly since 2021) works in favor of increasing solar energy potential. Therefore, the exploitation of solar energy is not limited to desert areas; it can also be advantageous in northern regions. M'Sila, located in northern Algeria at the Sahara's gates, is a real example of the increase in its solar energy potential since 2008, according to the study done by Younes et al. [24]. Indeed, M'Sila, with global horizontal irradiation (GHI) of 1.79 MWh/m²/year and direct normal irradiation (DNI) of 2117.93 kWh/m²/year, is a suitable location for building concentrating solar power facilities [24]. This region deserves to be studied to see whether building a solar thermal power plant similar to that erected in Spain (Andasol-1) under its environment is feasible given its geographic location, which qualifies it to become an economic and development pole. Moreover, M'Sila's electricity usage has increased over the past ten years due to improved

lifestyles [24]. Furthermore, government officials have paid close attention to encouraging investment in this area. For all these reasons, this study was carried out.

The originality of the current research lies in identifying the meteorological variables that have the most pronounced impact on the performance of Andasol-1 in M'Sila through stepwise linear regression and formulating a model for its electricity power generation based on these factors. Furthermore, the study compares Andasol-1's performance across various locations reported in the literature, rather than limiting the comparison solely to the reference location, Spain.

MATERIALS AND METHODS

The present theoretical investigation was conducted at M'Sila in northern Algeria to assess the performance of the reference power plant (Andasol-1), which has been operational since 2010 in southern Spain. The System Advisor Model (SAM) software, a commonly used tool to examine CSP plants, was utilized to perform this study. Meteorological data for the Andasol-1 site in Spain was supplied by the National Renewable Energy Laboratory via the SAM software library. Conversely, weather information for M'Sila was obtained from the METEONORM7 database. This meteorological data was input into the SAM software to simulate the performance of the CSP plant that will be erected by carrying out the following steps:

- Create a model base by simulating the reference solar power plant (Andasol-1), which has a capacity of 50 MW_e, at its actual site and comparing the findings with the real outputs.
- Simulate the reference plant under M'Sila's climatic conditions.
- Analyse the performance of the Andasol-1 solar power plant in M'Sila, with particular emphasis on the hourly annual energy distribution across the principal components of the solar installation, namely the solar field, the storage system, and the power block.
- Study of the effect of M'Sila's weather conditions on the Andasol-1's power output.
- Compare the performance of the Andasol-1 solar power plant with the different research projects carried out on the same plant in different countries, including its actual location in terms of net electricity generation, operating hours, SF efficiency, total plant efficiency, and water usage.

Technical Specifications Of The Reference Power Plant (Andasol-1)

The Andasol-1 plant, built by Solar Millennium, is the first parabolic trough plant in the world featuring a thermal storage technology [25]. It is close to Granada in southern Spain and had a 50 MWe maximum capacity when it first started operating in 2010 [26]. This plant covers 477 acres in total and primarily consists of a solar field, a thermal energy

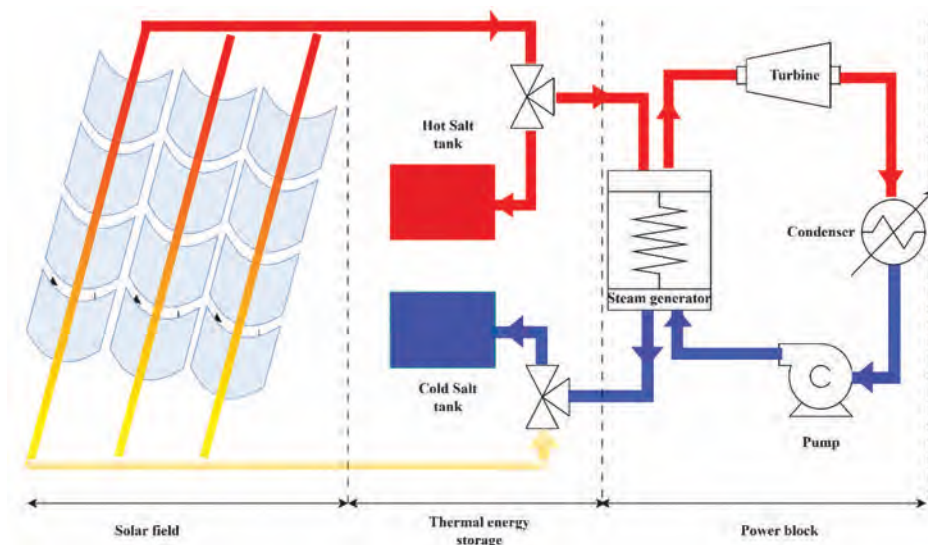


Figure 1. Schematic diagram of the Andasol-1 power plant.

storage system (TES), and a conventional power block. The solar collectors are arranged from north to south and rotate around this axis to follow the sun's daily course from east to west [27]. The main technical data for the reference plant is presented in Table 1 and was collected from published studies in the literature [15,25,26,28]. As depicted in Figure 1, the reference power plant primarily consists of a solar field, a thermal energy storage system (TES), and a conventional power block.

Description of the Selected Site

As previously mentioned, the M'Sila location was chosen for this study to evaluate the Andasol-1 solar power plant's performance in the M'Sila environment.

Given its distinctive geographic location, the province of M'Sila is regarded as one of the most prominent provinces in Algeria's highlands. M'Sila, a town in northern Algeria with 1,387,158 residents and an area of 18,175 km², lies about 250 kilometers south of Algiers, the country's capital [29].

According to geographic coordinates, it is located at 4° 32'43" longitude, 35° 42'07" latitude, and 441 m altitude. More than half of M'Sila's lands have a slope of less than 3%. In that province, the industry sector comes in first with a share of 57.86%, followed by the construction materials sector in second place with a share of 19.54%, and the service sector in third place with a share of 16.34% [30]. An 880-MW power plant is also present there [29].

Meteorological Data for the Selected Site

Concentrated solar power plants' performance and efficiency are significantly influenced by the local weather where they are installed. For M'Sila (the chosen site), the SAM software needs meteorological data such as direct normal irradiation (DNI) and ambient temperature to

Table 1. Technical data of the reference plant (Andasol-1) [15,25,26,28]

Description	Value
Solar field	
Solar field aperture area	510,120 m ²
Number of loops	156
HTF type	Dowtherm A
Design loop inlet temperature	293°C
Design loop outlet temperature	393°C
Collector	
Collector type EuroTrough	ET150
Length of one single collector	148.5 m
Aperture width of a collector	5.77 m
Focal length	1.71 m
Row spacing	17.3 m
Mirror reflectivity	93.5%
Optical efficiency	75%
Total collectors	624
Reflective aperture area	817.5 m ²
Storage system	
Full load storage hours	7.5 h
Storage fluid	Molten salt
Thermal storage capacity	1085.53 MWh _{th}
Cold tank temperature	292°C
Hot tank temperature	386°C
Power Block	
Nominal capacity	50 MW _e
Turbine inlet pressure	100 bar
Rated cycle conversion efficiency	38%
Cooling system type	Wet

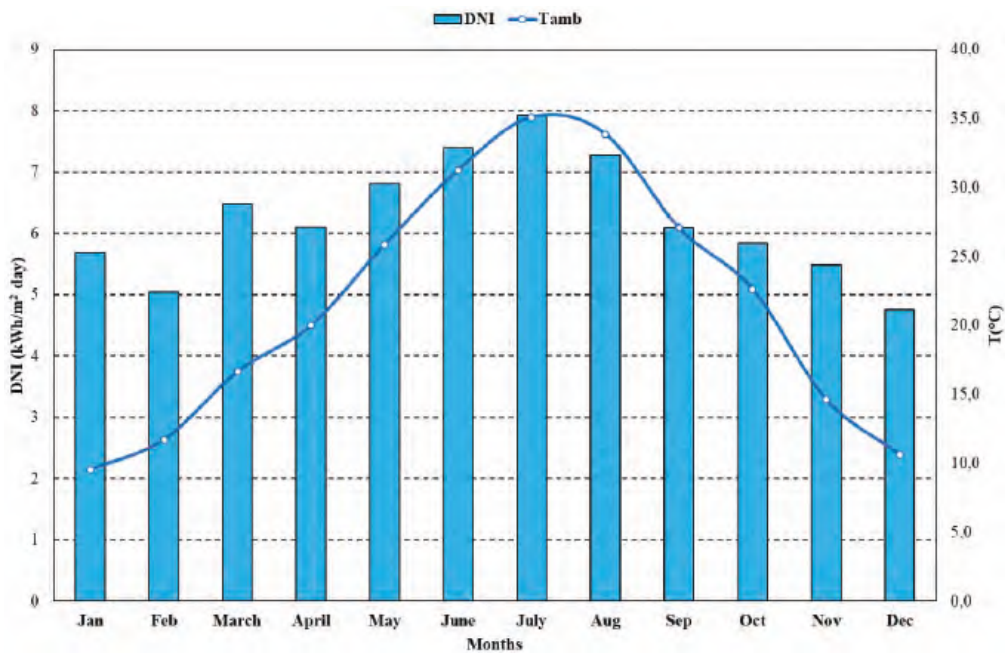


Figure 2. Monthly average daily direct normal irradiation and temperature in M'Sila.

perform the current investigation. Since it represents the average of all the years for which measurements are available, the METEONORM7 database generates meteorological data for one year, regarded as a typical year [21]. A Typical Meteorological Year file (TMY3) was downloaded from this software database, which includes hourly measurements of several meteorological parameters.

Figure 2 shows the monthly average daily direct normal irradiation and the monthly average daily temperature in M'Sila. It can be observed that, aside from February and December, the monthly average daily direct normal irradiation (DNI) value for the M'Sila location exceeds 5.5 kWh/m²/day all year long. Moreover, the smallest DNI value of 4.75 kWh/m²/day was recorded in December, while the maximum DNI value of 7.93 kWh/m²/day was registered in July. Further, it is estimated that M'Sila receives 2282.54 kWh/m²/year of direct normal irradiation (DNI) annually. The majority of the current CSP plants are erected in sites where the DNI value does not surpass 2153.5 kWh/m²/year [31]; hence, M'Sila is qualified to have CSP plants installed because of its significant DNI. It should be noted that M'Sila is today classified as an arid region [24] with low precipitation, especially over the past two years, as a result of the climate change it has undergone.

One of the fundamental meteorological parameters that exerts a significant impact on the heat dissipation of various CSP plant configurations is the ambient temperature. In M'Sila, a predominant range of hourly temperatures fluctuates between 10 and 32 °C. The issue concerning the potential freezing of the heat transfer fluid Therminol VP-1 emerges during the winter season in M'Sila due to the

possibility of temperatures dropping below 12 °C (Figure 2).

Furthermore, January recorded the lowest average monthly temperature at 9.5 °C, while July experienced the highest average monthly temperature at 35 °C (Figure 2).

The winds in M'Sila vary from calm winds to moderate breezes. In that region, the wind speed can thus change from 0 m/s to 4.8 m/s. Furthermore, only April sees wind speeds greater than 4 m/s; for the remainder of the year, the wind blows at less than 4 m/s.

CSP-PT PLANT'S ENERGY ANALYSIS

Assuming steady-state operating conditions, the energy analysis based on thermodynamic laws is carried out as follows for each component of a CSP-PT power plant:

The incident annual radiative solar power ($P_{\text{sun,incident}}$) on the solar field's net aperture area (A_{ap}) is given as [15]:

$$P_{\text{sun,incident}} = \sum_{i=1}^{365} \sum_{j=0}^{23} \text{DNI}_{ij} A_{\text{ap}} \quad (1)$$

The following formula provides the net aperture area of the solar field [15]:

$$A_{\text{ap}} = W_{\text{ap}} L M N = \sum_{i=1}^{N_{\text{SCA}}} A_{\text{collector},i} \quad (2)$$

Where W_{ap} is the width aperture of the collector shown in Figure 3, L is the length of the collector, M is the number of collector rows in series (see Figure 1), N is the number

of parallel lines (see again Figure 1), and N_{SCA} is the entire number of collectors in the solar field.

DNI_{ij} is the direct normal irradiance incident on the net aperture area of the solar field during the hour j of the day i .

The radiation incident on the solar field's net aperture area is greater than that incident on the absorber tube due to optical losses, which minimize the PTC's heat uptake from direct incident irradiation. These two quantities are connected by the optical efficiency ($\eta_{optical}$) of the collector, given as follows:

$$\eta_{optical} = \frac{P_{absorber}}{P_{sun, incident}} \quad (3)$$

Where $P_{absorber}$ is the annual power received on the absorber tube.

The heat flux absorbed by the heat transfer fluid (HTF) is represented by the thermal power output from the parabolic trough collector (Q_u), which also accounts for the heat power incident on the absorber tube and that lost from it into the environment [15,26], as depicted in Figure 3. It is given by:

$$Q_u = R_1 R_2 A_{ap} N \left[F_R (\tau\alpha)_n IAM I_{beam} - \frac{F_R U_L}{ConcRat} (T_{in} - T_{amb}) \right] \quad (4)$$

In which,

$$R_1 = \frac{M \dot{m} Cp_{fluid}}{A_{ap}} \left(\frac{1 - e^{(-F' U_L A_{ap}) / (M \dot{m} Cp_{fluid})}}{R_{test}} \right) \quad (5)$$

$$R_{test} = g_{test} Cp_{fluid} (1 - e^{(-F' U_L / (g_{test} Cp_{fluid}))}) \quad (6)$$

$$F' U_L = \begin{cases} F_R U_L & \text{if } \frac{F_R U_L}{g_{test} Cp_{fluid} ConcRat} \geq 1 \\ g_{test} Cp_{fluid} \left(1 - e^{(-\frac{F_R U_L}{g_{test} Cp_{fluid} ConcRat})} \right) & \text{if } \frac{F_R U_L}{g_{test} Cp_{fluid} ConcRat} < 1 \end{cases} \quad (7)$$

$$ConcRat = \frac{A_{ap}}{A_{absorber}} \quad (8)$$

$$R_2 = \frac{1 - \left(1 - ((R_1 A_{ap} F_R U_L) / (\dot{m} Cp_{fluid} M ConcRat)) \right)^M}{M ((R_1 A_{ap} F_R U_L) / (\dot{m} Cp_{fluid} M ConcRat))} \quad (9)$$

Where $F_R(\tau\alpha)_n$ represents the efficiency when the solar radiation is absorbed by the absorber tube and removed by the HTF, $F'U_L$ is the modified loss coefficient, IAM is the incidence angle modifier, I_{beam} is the amount of the beam solar radiation incident on the collector surface, T_{in} is the temperature of the HTF entering the collector array, T_{amb} is the ambient temperature, ConcRat is the concentration ratio between the aperture area and the absorber area, and is the mass flow rate of the HTF.

From the above, the absorber thermal efficiency ($\eta_{thermal}$) can be defined as the ratio of power acquired by the HTF to that incident on the absorber tube. The formula below provides it:

$$\eta_{thermal} = \frac{P_{HTF}}{P_{absorber}} = 1 - \frac{P_{th, loss}}{P_{sun, incident} \eta_{optical}} \quad (10)$$

Similarly, the solar field efficiency (η_{SF}) shown below can be identified as the ratio of the power acquired by the HTF to the incident power on the net aperture area of the solar field:

$$\eta_{SF} = \frac{P_{HTF}}{P_{sun, incident}} \quad (11)$$

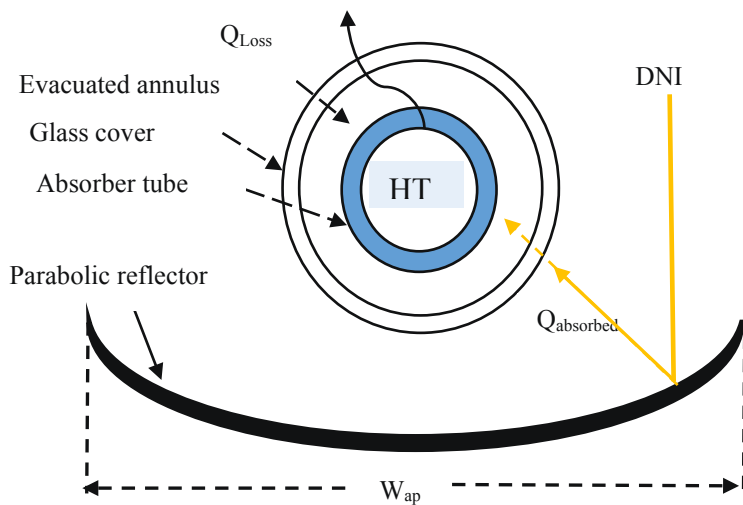


Figure 3. Parabolic trough collector's heat exchanges.

It should be noted that this annual efficiency can also be evaluated on an hourly, daily, and monthly basis.

The amount of DNI that is prevented from reaching the collector by mirror defocusing is known as the dumped energy or total defocused thermal energy resulting from the solar field (SF). The discarded power (Q_{dump}) is provided as [15,26]:

$$Q_{dump} = A_{ap} DNI \eta_{optical} \eta_{def} = \dot{m} C p_{fluid} (T_{out} - T_{max}) \quad (12)$$

Where T_{max} is the highest temperature at which the fluid may leave the collector, η_{def} is the ratio of defocused SCA units, and T_{out} is the temperature of the fluid at the collector outlet, given as [15,26]:

$$T_{out} = \begin{cases} T_{in} + \frac{Q_u}{\dot{m} C p_{fluid}} & \text{si } \dot{m} > 0 \\ T_{amb} + F_R (\tau \alpha)_n IAM I_{beam} \frac{ConcRat}{F_R U_L} & \text{si } \dot{m} = 0 \end{cases} \quad (13)$$

The dumped energy can be simulated by SAM software on an hourly basis.

Considering the heat exchanger's thermal losses, the net power block efficiency ($\eta_{net,PB}$) shows how well thermal input is converted to electric output. It's presented as:

$$\eta_{net,PB} = \frac{P_{elc,turb} - P_{ele,pompe} - P_{aux,cond}}{(P_{HTF} - P_{loss,piping})} = \frac{P_{ele,net,PB}}{(P_{HTF} - P_{loss,piping})} \quad (14)$$

Where $P_{ele,turb}$ is the electric power output of the steam turbine in MW; $P_{ele,pump}$ is the consumption rate of the condensate and feed water pumps; $P_{aux,cond}$ is the consumption rate of the condenser auxiliary; $P_{loss,piping}$ is the thermal losses rate in the piping; and $P_{ele,net,PB}$ is the power block's net power output in MW.

The ratio of the electric power generated by the solar power plant to the solar power incident on the solar field is the overall solar to electric efficiency of the plant

($\eta_{overall}$), which is determined along with the other efficiencies as follows:

$$\eta_{overall} = \eta_{optical} \eta_{thermal} \eta_{piping} \eta_{net,PB} \eta_{aux,SF} = \frac{P_{ele,a}}{P_{sun,incident}} \quad (15)$$

$P_{ele,a}$ is the steam turbine's annual electric power output measured in MW; $\eta_{aux,SF}$ is the SF auxiliary's efficiency, expressing the effect of the SF tracking consumptions and circulating pumps on the net PB output; and η_{piping} is the piping efficiency, assessing the effect of thermal losses in the piping, including those that occur at night, on the heat transfer fluid (HTF) used to transfer thermal power.

The thermal energy storage capacity of the TES system, measured in hours of thermal energy delivered at the thermal input level of the power block design, is known as the full load hours of the TES. It is calculated as follows [15,32]:

$$Q_m^{TES} = N_h^{TES} \frac{P_{elec,PB}}{\eta_{PB}} \quad (16)$$

Where Q_m^{TES} is the thermal energy storage capacity of the TES system, N_h^{TES} is the number of full load hours of storage, and $P_{elec,PB}$ is the power block's power output.

RESULTS AND DISCUSSION

Results Validation

This section's aim is to compare the SAM simulation results for the Andasol-1 reference plant, located in southern Spain, with the real and simulated outputs published by Herrmann et al. [28], Bataineh et al. [33], and Sultan et al. [15] in order to show the accuracy of the simulation process. There are reasonable deviations, as demonstrated by Table 2, which displays our simulation results along with real and simulated data for the Andasol-1 power plant published in the literature. The percentage discrepancies for the performance of the Spanish reference power plant between the

Table 2. Outcomes on the operation of the Andasol-1 power plant under its location's climate conditions, both real and simulated

Description	Unit	Published data (Herrmann et al. [28])	Published data (Bataineh et al. [33])	Published data	Our simulation (Sultan et al [15])
Annual DNI	kWh/m ²	2202	2052	2033.3	2034.7
Annual solar irradiation on the SF	MWh/a	1,105,430	—	1,037,586.6	1,037,932
Annual thermal power from the SF	MWh/a	510,030	—	479,609.8	473,996.5
Annual net electricity output	MWh _e /a	157,206	153,560.0	151,569.8	151.282.5
Annual solar field efficiency	%	46.1	44.2	46.2	45.7
Annual overall efficiency	%	14.7	13.1	14.6	14.6
Number of full load hours of steam Turbine	h	3144	3098	3003	3056
Total water consumption	m ³ /a	612,000.00	—	508,949.0	584,541.0

Table 3. Percentage differences between the reported results related to the reference power plant's performance under its location's climate

Description	(% difference)					
	Bataineh et al. [33]. compared to	Sultan et al. [15]. compared to		Simulated data from this work compared to		
	Herrmann et al. [28]	Herrmann et al. [28],	Bataineh et al. [33]	Herrmann et al. [28],	Bataineh et al. [33],	Sultan et al. [15]
Annual DNI	-7.31	-8.30	-0.92	-8.22	-0.85	0.07
Annual Solar irradiation on the solar field	—	-6.54	—	-6.5	—	0.03
Thermal power from the solar field	—	-6.34	—	-7.6	—	-1.18
Annual net electricity output	-2.37	-3.72	-1.31	-3.92	-1.51	-0.19
Annual overall efficiency	-12.21	-0.68	10.27	-0.69	10.27	0.0
Annual solar field efficiency	-4.29	0.22	4.33	-0.88	3.28	-1.1
Number of full load hours of steam Turbine	-1.48	-4.70	-3.16	-2.8	-1.35	1.76
Total water consumption	—	-20.25	—	-4.7	—	12.93

simulated results taken into consideration (mentioned in Table 2) and the actual data are displayed in Table 3. Our simulation, as can be seen, approaches that of Sultan et al. [15], which, on the whole, is not too far from the real data reported by Herrmann et al. [28], particularly in terms of yearly overall plant efficiency, yearly net electricity output, thermal power from the solar field, and annual solar field efficiency.

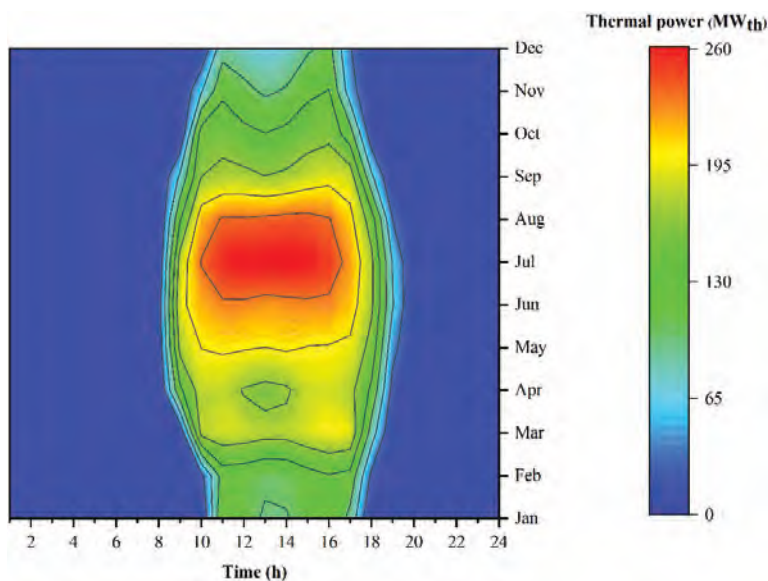
Indeed, for all investigated parameters, the difference between our simulation findings and the actual operating results of the Andasol-1 central is less than 9%. Since SAM simulation software can accurately predict the actual data of the reference plant, it will be utilized in the following

sections to assess its performance in the M'Sila climate and compare it to the Andasol-1 reference plant in Spain and, on the other hand, to the simulated works of the same plant in different parts of the world.

Andasol-1 Power Plant Simulation Under M'sila Climate Conditions

The following part discusses how power varies throughout the various components of the simulated Andasol-1 power plant when it is operating in M'Sila.

The hourly annual distribution of the thermal power generated by the solar field is shown in Figure 4. As can

**Figure 4.** Hourly annual distribution of the thermal power generated by the solar field of Andasol-1 under the climatic conditions of M'Sila.

be observed, the hourly solar field's thermal power production peak occurs during the summer months (from June to August), with values ranging from 230 to 267 MW_{th} /hour. However, this peak is at its lowest during the winter months. Indeed, it doesn't go above 150 MW_{th} /hour. Furthermore, the spring and autumn seasons record values between 135 MW_{th} /hour and 204 MW_{th} /hour for the hourly solar field's thermal power output peak. Moreover, the Andasol-1 solar field, when installed in M'Sila, generates thermal power from 11:00 a.m. to 5:00 p.m. throughout the year, ranging between 77.70 MW_{th} and 256 MW_{th} on average. In May, June, and July, thermal energy production can last up to 12 hours, starting at 8 a.m. and ending at 7 p.m.

The thermal power production of the Andasol-1 solar field situated in M'Sila varies over the course of the year. Specifically, this solar power plant demonstrates its highest thermal power output during the summer months and its lowest during the winter season, with average capacities of 2292 MW_{th} and 728 MW_{th} , respectively. Moreover, the average thermal power production during spring surpasses that of autumn, with mean values standing at 1746 MW_{th} and 1178 MW_{th} , respectively.

Figure 5 shows the power available in the thermal storage system (TES) of the proposed solar power plant in M'Sila over the year on an hourly basis. It is evident that, given its direct correlation with DNI level, the TES system is inefficient in the winter and crucial in the summer, with average values of 57 and 703 MW_{th} , respectively. Moreover, the maximum hourly thermal power in the thermal storage system, which ranges from 86 to 123 MW_{th} /hour, is evidently noticed to occur in the summer (June to August).

Nevertheless, in the winter months, this peak is at its lowest since it doesn't exceed 25 MW_{th} /hour. In addition, the hourly storage system's thermal power peak is recorded at 21 MW_{th} /hour to 70 MW_{th} /hour in the spring and autumn. Further, when erected in M'Sila, the Andasol-1 power plant's storage system stores thermal power year-round from 11:00 a.m. to 4:00 p.m., ranging between 3 MW_{th} and 102 MW_{th} on average. The TES system accumulates thermal power from 9:00 a.m. to 5:00 p.m. from April through September, or even until 6:00 p.m. in May and June.

The hourly annual distribution of electrical power generated by the proposed solar power plant in M'Sila, which is the final electrical power ready to be delivered to the grid, is depicted in Figure 6. As seen, the Andasol-1 power plant, when located in M'Sila, produces the most electricity during the summer and the least during the winter, with average values of 745 and 235 MW_e , respectively. Moreover, the average electrical power output in the spring is higher than that in the fall, averaging 562 and 389 MW_e , respectively. It is clear that the summer months of June through August are when the hourly power production peak, which averages 52 MW_e /hour, occurs. Still, this peak is at its lowest during the winter because it never rises above 40 MW_e /hour. Furthermore, in the spring and fall, the hourly electricity production peak ranges from 41 MW_e /hour to 48 MW_e /hour. Further, when the Andasol-1 power plant is installed in M'Sila, it generates electricity year-round from 11:00 a.m. to 5:00 p.m., with an average output ranging from 25 MW_e to 52 MW_e . It can reach 17 hours of production, or even more, from March to September due to the favorable weather conditions and energy storage system (TES), as seen above.

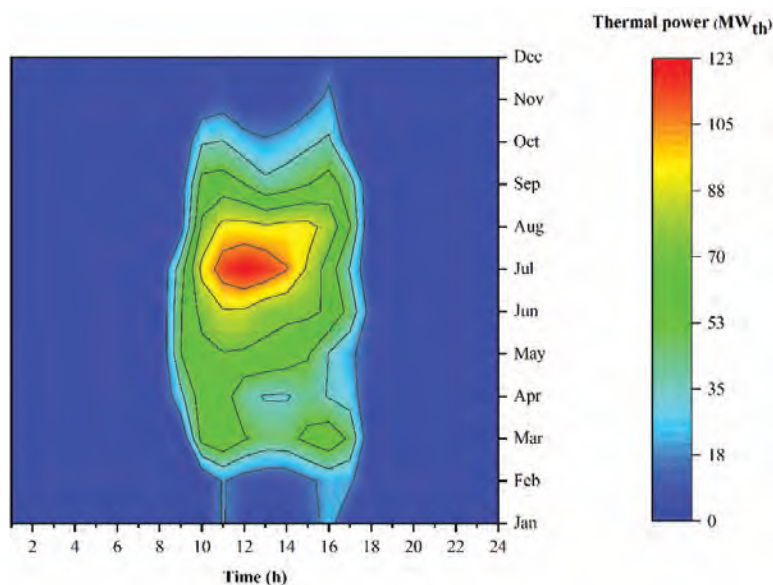


Figure 5. Hourly annual distribution of the thermal power accumulated by the TES of Andasol-1 under the climatic conditions of M'Sila.

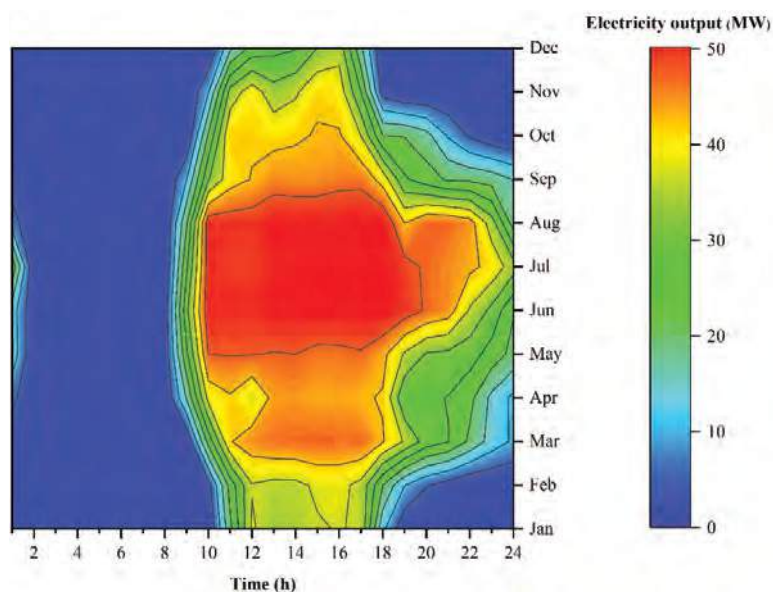


Figure 6. Hourly annual distribution of the electricity output by the Andasol-1 under the climatic conditions of M'Sila.

Figure 7 displays the proposed CSP-PT power plant's monthly net electricity production in conjunction with the meteorological conditions in M'Sila. As seen, the Andasol-1 power plant in M'Sila generates 24.15 GWh of electrical energy at its maximum in July and 4.81 GWh at its minimum in December. These values correspond to M'Sila's monthly maximum and minimum recorded DNI, respectively. Moreover, as the ambient temperature rises, so

does the net electrical output, and vice versa. Further, relative humidity and net electricity production are inversely related. The lowest relative humidity practically coincides with the maximum productivity, and vice versa. However, the net electricity output is not impacted by the wind since it blows at less than 4 m/s most of the time in M'Sila, except for April (the windiest month), where the production records a slight decrease.

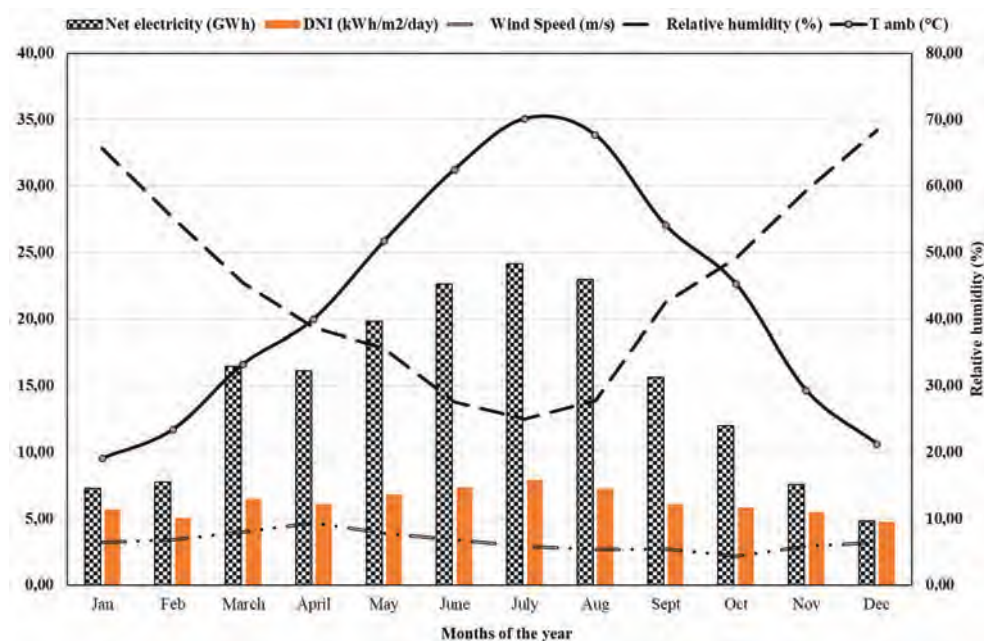


Figure 7. Andasol-1 power plant's monthly net electricity production in conjunction with the meteorological conditions in M'Sila.

Our results are in agreement with those reported in the literature [15,26,34] regarding the effect of DNI and ambient temperature on the functioning of Andasol-1 (net electricity production) in different regions of the world. Furthermore, in contrast to the existing qualitative studies cited above, we suggest using stepwise linear regression—a quantitative study—to identify the meteorological factors that have the most significant effect on M'Sila's clean electricity output. The findings of stepwise linear regression on the impact of weather conditions on Andasol-1's power output, when sited in M'Sila, are displayed in Table 4.

As seen, DNI, temperature, and relative humidity are significant factors affecting electricity production in M'Sila since their p-values are less than 0.05 (significant level). Furthermore, this statistical analysis reaffirms that, when situated in M'Sila, wind speed has no discernible impact on the power plant's production.

When combined, the variables DNI, temperature, and relative humidity account for 98.2% of the variance in electricity production in M'Sila ($R^2=0.982$ in Table 4). In this regard, we propose the following model, which describes the power generation in M'Sila based on the most influential hourly weather conditions:

$$P = 287.03 + 0.009 \times DNI - 5.482 \times T - 3.342 \times \phi \quad (17)$$

Where,

P is the net power output in MW, DNI is the direct normal irradiance in W/m^2 , T is the dry temperature in $^{\circ}C$, and ϕ is the relative humidity in %.

Additionally, Table 4 (ANOVA) indicates that the suggested model is significant with $F(3,20) = 361.583$ and P (significance F) < 0.05.

The PCA (Principal Component Analysis) approach is another statistical study performed to evaluate the contribution of the months of the year in the production of power in M'Sila. The outcomes indicate that 86.42% of the variance in the hourly annual production is accounted for by the seven months of July, June, August, May, April, September, and March; the remaining months, December, January, November, February, and October, account for 11.26% of the variance. Stated differently, the generation of electricity throughout the summer, spring, and the first month of autumn accounts for 78% of the total yearly output, surpassing 15 GWh/month. Production stays below 12 GWh per month in the winter and the remaining autumn months.

Andasol-1 Power Plant Simulation Under Different Climate Conditions

The simulated net electricity generated by Andasol-1 is shown in Figure 8 on a monthly and annual basis in different regions, each with its own environmental parameters. Andasol-1, which generates 189.05 GWh of power yearly in Jordan's Ma'an, a cold desert region, has demonstrated its ability to function efficiently in these conditions. The second climatic condition favorable to Andasol-1 is that of M'Sila (arid climate, according to our recent work [24]), where it can produce 177.22 GWh of electricity annually. Furthermore, the subtropical desert steppe climate of Tajoura, Libya, enables Andasol-1 to achieve nearly the same amount of electricity annually as M'Sila. When Andasol-1 is erected in Kuwait, Tataouine, Tunisia, or at its true site (Spain), it produces less electricity annually than M'Sila, Tajoura, and Ma'an. A difference of more than 11 GWh can be noticed. Once more, the site's environment has a significant impact on the CSP power plant's net output production.

Table 4. Effect of meteorological conditions on Andasol-1's power output using stepwise linear regression

Regression Statistics					
Multiple R					0.991
R Square					0.982
Adjusted R Square					0.979
Standard Error					2.632
Observations					24
ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	3	7512.461	2504.154	361.583	0.000
Residual	20	138.511	6.926		
Total	23	7650.971			
		<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>
Intercept		287.0336794	95.20558914	3.014882656	0.00684
Hourly Data: Resource Beam normal irradiance (W/m^2)		0.008920514	0.003467863	2.572337228	0.01818
Hourly Data: Resource Dry bulb temperature ($^{\circ}C$)		-5.481630697	2.510739957	-2.183272976	0.04110
Hourly Data: Relative humidity (%)		-3.341774233	0.901347004	-3.707533523	0.00139

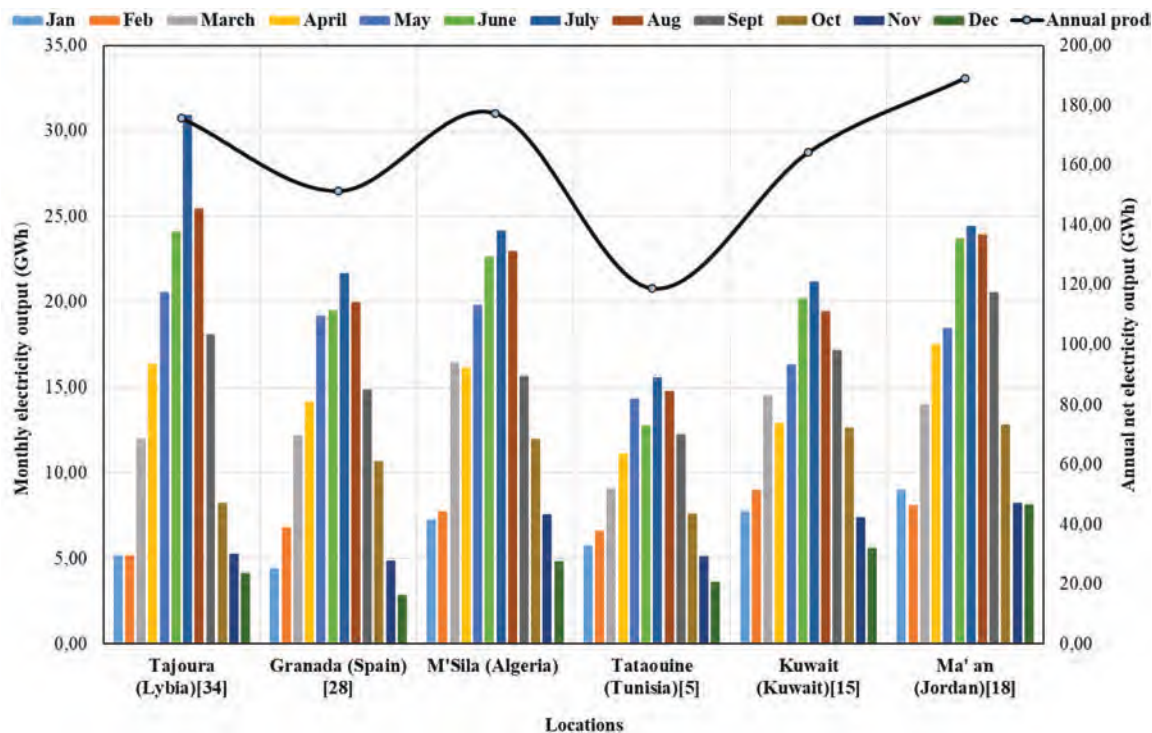


Figure 8. Andasol-1 power plant's monthly and annual net electricity production in different zones in the world.

Regardless of where in the northern hemisphere it is located, Andasol-1 produces the most power in July and the least in December; nevertheless, the extreme values reached differ from place to place (Figure 8). As it appears, Andasol-1, situated in Tajoura, Libya, can produce up to 30.93 GWh of electricity in July, compared to 15.56 GWh in Tataouine, Tunisia, over the same period. Moreover, when located in Ma'an, Jordan, it may produce up to 8.14 GWh of electricity in December, compared to 2.87 GWh at its actual location in Spain.

Andasol-1 generates more electricity at all of the locations under consideration during the summer, spring, and first month of fall. More than 74% of the yearly production is produced during this time. During the winter and the remaining autumn months, the net electrical energy produced remains below 13 GWh/month, except in Tataouine, where it falls below 8 GWh/month. Further, Andasol-1, constructed in M'Sila, generates more electricity monthly than it does in its current location all year round.

As seen in Figures 8 and 9, of all the regions examined, Ma'an (Jordan) has the highest annual DNI (2693 kWh/m²/

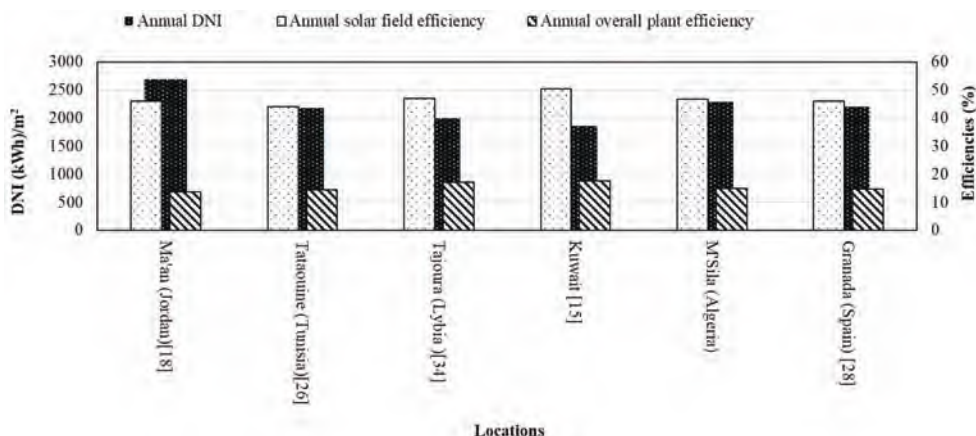


Figure 9. DNI of the zones studied and efficiencies of the Andasol-1 installed there.

m²) [18], producing, thus, the maximum amount of clean electricity. Conversely, Kuwait, with the lowest annual DNI (1857.1 kWh/m²) [15], produces more electricity than southern Spain because of its highest ambient temperature and lowest wind speed, which result in the highest solar field efficiency of 50.3% (lowest heat loss from the solar field). Furthermore, Tataouine in Tunisia has the lowest solar field yield of the sites under investigation, at 44.02% [5]. Kuwait has the greatest annual overall plant efficiency, with an annual efficiency of 17.5%, while Ma'an in Jordan has the lowest efficiency, at 13.8%. Southern Spain and M'Sila exhibit nearly identical overall efficiency.

The highest number of operating hours at full load (4399 h) of Andasol-1 is seen at Ma'an, whose DNI is the highest [18]. M'Sila, Kuwait, and Southern Spain (the actual site) follow with 3580 h, 3306 h [15], and 3144 h [28] of full load running, respectively. Moreover, Andasol-1 offers the lowest running hours at full load (2548 h) in Tataouine, Tunisia [5].

Again, Andasol-1, erected in Ma'an, consumes the highest amount of water, 751617.7 m³/year [18], instead of 592799 m³/year (the least amount) when installed in Kuwait [15]. Andasol-1 in M'Sila uses more water than Spain, Tataouine, and Kuwait, with values of 674301 m³/year, 612000 m³/year [28], 649735 m³/year [26], and 592799 m³/year [15], respectively.

Under the climatic circumstances of M'Sila, the discarded power (Q_{dump}) is estimated to be 8046.86 MW/year. It is recorded, as in southern Spain, during the period from March to September, with July having the highest value.

From the above, it is clear that the location affects the solar thermal power plant's performance significantly. Moreover, it is better to discuss the combined impact of the site's meteorological variables (DNI, temperature, relative humidity, and wind speed) rather than each one separately.

CONCLUSION

The increasing global demand for energy and the negative environmental impacts associated with traditional energy sources have made the transition to renewable energy sources imperative. In this context, the present theoretical study aims to assess the feasibility of implementing Andasol-1, an operational solar thermal power plant located in Spain, within M'Sila, a desert region in Algeria, utilizing SAM software. Meteorological data for M'Sila were obtained from the METEONORM7 database and served as input for the SAM software.

The following results are to be retained:

- When installed in M'Sila, the Andasol-1 solar power plant is projected to generate thermal power averaging between 77.70 MWth and 256 MWth from 11:00 a.m. to 5:00 p.m. throughout the year. It will store thermal power from 11:00 a.m. to 4:00 p.m., with an average range of 3 MWth to 102 MWth annually. The facility will produce electricity from 11:00 a.m. to 5:00 p.m.,

yielding an average output of 25 MWe to 52 MWe year-round.

- The meteorological factors that most significantly influence clean electricity generation in M'Sila are direct normal irradiance (DNI), temperature, and relative humidity, which collectively account for 98.2% of the variance in electricity production ($R^2 = 0.982$). The climate of M'Sila ranks as the second most favorable for Andasol-1 after Jordan's climate since it can produce an annual total of 177.22 GWh of electricity—surpassing its current operational site.
- Furthermore, by deploying concentrated solar power (CSP) units utilizing parabolic trough technology, M'Sila can alleviate pressure on its thermal power plant during summer months while simultaneously reducing greenhouse gas emissions.

NOMENCLATURE

Acronyms

CSP	Concentrated solar power
CSP-PT	Parabolic trough concentrated solar power
GHI	Global horizontal irradiation
HTF	Heat transfer fluid
IEA	The International Energy Agency
ISCC	Integrated Solar Combined Cycle
LCOE	Levelized cost of electricity
LFR	Linear Fresnel reflector
NREL	National Renewable Energy Laboratory ()
PCA	Principal Component Analysis
PTC	Parabolic trough collector
PV	Photovoltaic
Q_{inc}	Incident solar energy on the solar field per unit time
Q_{field}	Thermal energy produced by the solar field per unit time
Q_{inp}	Thermal energy that enters the power block per unit time
Q_{st}	Change in the amount of thermal energy in the storage system per unit time
SAM	System Advisor Model
SF	Solar field
SCA	Solar Collector Assembly
SNL	Sandia National Laboratories
SPPI	Solar Power Plant One
SPD	Solar parabolic dish
SPT	Solar power tower
TES	Thermal energy storage
TMY3	Typical Meteorological Year file
W_{net}	Net electrical power produced by the solar power plant

Symbols

A_{ap}	Solar field's net aperture area (m ²)
ConcRat	Concentration ratio between the aperture area and the absorber area (–)

DNI	Direct normal irradiation (W/m ²)
F'	Collector efficiency factor (-)
FR	Collector heat removal factor (-)
$F_R(\tau\alpha)_n$	The efficiency when the solar radiation is absorbed by the absorber tube and removed by the fluid (-)
IAM	Incidence angle modifier (-)
I_{beam}	Amount of beam solar radiation incident on the collector surface (W/m ²)
L	Collector length (m)
M	Number of rows of collectors in series (-)
$\dot{m}$	Mass flow rate of fluid (kg/h)
N	Number of parallel lines (-)
N_{SCA}	Total number of collectors (-)
N_h^{TES}	Number of full load hours of storage (-)
P	Net power output in MW
$P_{\text{sun,incident}}$	Radiative solar power incident on net aperture area (MW)
P_{absorber}	Annual power received on the absorber tube (MW)
$P_{\text{aux,cond}}$	Condenser auxiliaries consumption rate (MW)
$P_{\text{ele,turb}}$	Steam turbine electric power output (MWe)
$P_{\text{ele,pump}}$	Condensate and feedwater pump consumption rate (MW)
$P_{\text{loss,piping}}$	Piping thermal losses rate (MW)
$P_{\text{ele,net,PB}}$	Net power output from the power block (MW)
$P_{\text{elec, PB}}$	Power block's power output (MW)
$P_{\text{th,SF}}$	Thermal power delivered by the solar field at the design point (MW)
$P_{\text{th,in,PB}}$	Power block's thermal power need at nominal conditions (MW)
$P_{\text{ele,a}}$	Annual net electricity output (MW)
Phi	Relative humidity (%)
Q_u	Thermal power output from the solar field (MW)
Q_m^{TES}	Thermal energy storage capacity (MWh)
Q_{dump}	Dumped thermal power from the solar field (MW)
T	Dry temperature in (°C)
T_{amb}	Ambient temperature (°C)
T_{in}	Temperature of the HTF entering the collector array (°C)
T_{max}	Highest temperature at which the fluid may leave the collector (°C)
T_{out}	Temperature of the fluid at the collector outlet (°C)
U_L	Loss coefficient (W/m ² °C)
W_{ap}	Width aperture of the collector (m)
Greek symbols	
h_{thermal}	Thermal efficiency (%)
h_{SF}	Solar field efficiency (%)
h_{optical}	Optical efficiency (%)
$\eta_{\text{net,PB}}$	Net power block efficiency (%)
η_{overall}	Annual overall plant efficiency (%)
$\eta_{\text{aux,SF}}$	Efficiency of solar field auxiliary (%)
η_{piping}	Piping efficiency (%)

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Khan J, Arsalan MH. Solar power technologies for sustainable electricity generation - A review. *Renew Sustain Energy Rev* 2016;55:414–425. [\[CrossRef\]](#)
- [2] Awan AB, Zubair M, Chandra Mouli KVV. Design, optimization and performance comparison of solar tower and photovoltaic power plants. *Energy* 2020;199:117450. [\[CrossRef\]](#)
- [3] Dinter F, Gonzalez DM. Operability, reliability and economic benefits of CSP with thermal energy storage: First year of operation of ANDASOL 3. *Energy Procedia* 2014;49:2472–2481. [\[CrossRef\]](#)
- [4] Islam MT, Huda N, Abdullah AB, Saidur R. A comprehensive review of state-of-the-art concentrating solar power (CSP) technologies: Current status and research trends. *Renew Sustain Energy Rev* 2018;91:987–1018. [\[CrossRef\]](#)
- [5] Balghouthi M, Trabelsi SE, Ben Amara M, Ali ABH, Guizani A. Potential of concentrating solar power (CSP) technology in Tunisia and the possibility of interconnection with Europe. *Renew Sustain Energy Rev* 2016;56:1227–1248. [\[CrossRef\]](#)
- [6] Pavlović TM, Radonjić IS, Milosavljević DD, Pantić LS. A review of concentrating solar power plants in the world and their potential use in Serbia. *Renew Sustain Energy Rev* 2012;16:3891–3902. [\[CrossRef\]](#)
- [7] Suresh NS, Thirumalai NC, Rao BS, Ramaswamy MA. Methodology for sizing the solar field for parabolic trough technology with thermal storage and hybridization. *Sol Energy* 2014;110:247–259. [\[CrossRef\]](#)

- [8] Ouagued M, Khellaf A, Loukarfi L. Estimation of the temperature, heat gain and heat loss by solar parabolic trough collector under Algerian climate using different thermal oils. *Energy Convers Manag* 2013;75:191–201. [CrossRef]
- [9] Yilmaz İH, Mwesigye A. Modeling, simulation and performance analysis of parabolic trough solar collectors: A comprehensive review. *Appl Energy* 2018;225:135–174. [CrossRef]
- [10] Baharoon DA, Rahman HA, Omar WZW, Fadhl SO. Historical development of concentrating solar power technologies to generate clean electricity efficiently – A review. *Renew Sustain Energy Rev* 2015;41:996–1027. [CrossRef]
- [11] Kedar S, Bewoor A, Murali G, More GV, Roy A. Thermal analysis of sea water hybrid solar desalination system - an experimental approach. *Int J Heat Technol* 2024;42:1349–1358. [CrossRef]
- [12] Kedar S, Murali G, Bewoor AK. Mathematical modelling and analysis of hybrid solar desalination system using evacuated tube collector (ETC) and compound parabolic concentrator (CPC). *Math Model Eng Probl* 2021;8:45–51. [CrossRef]
- [13] Ehtiwesh IAS, Neto Da Silva F, Sousa ACM. Deployment of parabolic trough concentrated solar power plants in North Africa – a case study for Libya. *Int J Green Energy* 2019;16:72–85. [CrossRef]
- [14] Attai YA, Megallaa KF, Aziz SS. Comparison for performance of concentrated parabolic trough power plant in Egypt & Spain. *Int J Emerg Technol Adv Eng* 2015;5:327–333.
- [15] Sultan AJ, Hughes KJ, Ingham DB, Ma L, Pourkashanian M. Techno-economic competitiveness of 50 MW concentrating solar power plants for electricity generation under Kuwait climatic conditions. *Renew Sustain Energy Rev* 2020;134:110342. [CrossRef]
- [16] Jailani NA, Ahmad A, Norazahar N. A techno-economic analysis of parabolic trough collector (PTC) and solar power tower (SPT) as solar energy in Malaysia. *J Energy Saf Technol* 2021;4:13–28. [CrossRef]
- [17] Shagdar E, Lougou BG, Sereeter B, Shuai Y, Mustafa A, Ganbold E, Han D. Performance analysis of the 50 MW concentrating solar power plant under various operation conditions. *Energies* 2022;15:1–24. [CrossRef]
- [18] Liqreina A, Qoaider L. Dry cooling of concentrating solar power (CSP) plants, an economic competitive option for the desert regions of the MENA region. *Sol Energy* 2014;103:417–424. [CrossRef]
- [19] Trabelsi SE, Qoaider L, Guizani A. Investigation of using molten salt as heat transfer fluid for dry cooled solar parabolic trough power plants under desert conditions. *Energy Convers Manag* 2018;156:253–263. [CrossRef]
- [20] Achour L, Bouharkat M, Behar O. Performance assessment of an integrated solar combined cycle in the southern of Algeria. *Energy Reports* 2018;4:207–217. [CrossRef]
- [21] Benhadji Serradj DE, Sebitosi AB, Fadlallah SO. Design and performance analysis of a parabolic trough power plant under the climatological conditions of Tamanrasset, Algeria. *Int J Environ Sci Technol (Tehran)* 2022;19:3359–3376. [CrossRef]
- [22] Ikhlef K, Larbi S. Techno-economic optimization for implantation of parabolic trough power plant: Case study of Algeria. *J Renew Sustain Energy* 2020;12:063704. [CrossRef]
- [23] Benabdellah HM, Ghenaïet A. Energy, exergy, and economic analysis of an integrated solar combined cycle power plant. *Eng Reports* 2021;3:1–25. [CrossRef]
- [24] Kherbiche Y, Ihaddadene N, Ihaddadene R, Hadji F, Mohamed J, Beghidja AH. Solar energy potential evaluation. Case of study: M'Sila, an Algerian province. *Int J Sustain Dev Plan* 2021;16:1501–1508. [CrossRef]
- [25] National Renewable Energy Laboratory. System Advisor Model (SAM) Case Study. pp. 1–10. Available at: https://sam.nrel.gov/images/web_page_files/sam_case_csp_physical_trough_andasol-1_2013-1-15.pdf Accessed on Sep 05, 2025.
- [26] Trabelsi SE, Chargui R, Qoaider L, Liqreina A, Guizani A. Techno-economic performance of concentrating solar power plants under the climatic conditions of the southern region of Tunisia. *Energy Convers Manag* 2016;119:203–214. [CrossRef]
- [27] Andasol 1 project. Available at: <https://solarpaces.nrel.gov/project/andasol-1> Accessed on January 20, 2023.
- [28] Herrmann U, Geyer M. The Andasol Project. Workshop on thermal storage for trough power systems 2002; February 20–22.
- [29] Monograph on the Province of M'Sila, National Agency for Intermediation and Land Regulation (ANIREF), Ministry of Industry. Available at: <https://www.aniref.dz/DocumentsPDF/monographies/MONOGRAPHIE%20WILAYA%20MSILA.pdf> Accessed on April 6, 2023.
- [30] <http://dim-msila.dz/?p=267> Accessed on April 6, 2023.
- [31] Abbas M, Merzouk NK, Belgroun Z, Aburidah H. Parametric study of the installation of a solar power tower plant under Saharan climate of Algeria: Case study of Tamanrasset. In: *Proceedings of the First International Conference on Nano-electronics, Communications and Renewable Energy (ICNCRE 2013)* 2013:357–362.
- [32] Blair N, Dobos A, Freeman J, Neises T, Wagne M, Ferguson T, Gilman P, Janzou S. System Advisor Model, SAM 2014.1.14, General Description. NREL Report No. TP-6A20-61019; National Renewable Energy Laboratory (NREL): Golden, CO, USA, 2014; p. 19. [CrossRef]

- [33] Bataineh KM, Gharaibeh A. Optimization analyses of parabolic trough (CSP) plants for the desert regions of the Middle East and North Africa (MENA). *Jordan J Mech Ind Eng* 2018;12:33–43.
- [34] Belgasim B, Aldali Y, Abdunnabi MJR, Hashem G, Hossin K. The potential of concentrating solar power (CSP) for electricity generation in Libya. *Renew Sustain Energy Rev* 2018;90:1–15. [\[CrossRef\]](#)



Research Article

Effect of bristled shark scale structures on the aerodynamic characteristics of leading-edge protuberanced wing section

S. SMRITHIKA¹, S. ARUNVINTHAN^{1,*}

¹Department of Aerospace Engineering, SASTRA Deemed University, Thanjavur, Tamil Nadu, 613401, India

ARTICLE INFO

Article history

Received: 28 May 2017

Accepted: 08 August 2017

Keywords:

Aerodynamic Force Coefficients;
Biomimetics; Flow Control
Leading-Edge Protuberances,
Time-Series Data, Wind Tunnel
Testing, Surface Pressure
Distribution

ABSTRACT

The present paper focuses its attention on assessing the influence of bristled shark scale structures over biologically inspired leading-edge protuberanced (LEP) airfoil section. It is worth noting that utilization of bristled shark scale structures as an effective flow control method remains untouched to date and this paper aims to study the same. NACA 63(4)-021 airfoil has been utilized as the baseline model in this study as it is closely reminiscent of the flippers of the Humpback whales. The test models include a baseline LEP model and two modified models M1 and M2 fitted with a single strip of shark scale structures at 0.6C and consecutive strips placed between 0.6 and 0.8C respectively. All the sets of experiments were conducted in the low-speed subsonic wind tunnel facility. The leading edge protuberanced wing utilized in the present study features an amplitude of 0.12C and wavelength of 0.5C based on the foundation developed by the previous researchers. The bristled shark scale structures inspired by the short-fin mako as well as the test model were 3D printed using PLA material at a resolution of $100\mu/m$. The test models were experimentally evaluated for a wide range of angles of attack ranging from $0^\circ \leq \alpha \leq 70^\circ$ in increments of 5° at $Re=1.71 \times 10^5$. Surface pressure measurements were obtained over the test models with the help of MPS4264 Scanivalve pressure scanner which are pneumatically connected to the pressure tapings. Aerodynamic forces and force coefficients were then estimated using pressure integration technique from the surface pressure measurements. Results reveal that the bristled shark scale tends to improve the aerodynamic characteristics in terms of lift increment and delay in flow separation. In other words, the modified models are effective as flow control means over the leading-edge protuberanced airfoil section. M1 and M2 improve the lift coefficient by 44% and 18.6% respectively when compared against the LEP baseline model. The prevailing spanwise gradient in the LEP baseline model is reduced around 85% in the modified model M1.

Cite this article as: Smrithika S, Arunvinthan S. Effect of bristled shark scale structures on the aerodynamic characteristics of leading-edge protuberanced wing section. J Ther Eng 2025;11(5):1455–1467.

*Corresponding author.

*E-mail address: smrithika613@gmail.com, sarunvinthan@gmail.com

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkilic



INTRODUCTION

One classical problem associated with the airfoils is the flow separation. The flow separation is a phenomenon involving separation of the boundary layer from the airfoil surface. Since, airfoils are widely utilized in various man-made applications like wind turbine blades, missiles, etc. [1-7], it is essential to control the flow separation over the airfoils. This necessitates flow control methods which can effectively delay the flow separation. Flow control methodologies are very much crucial in tailoring the aerodynamic characteristics of airfoil. These techniques manipulate the airflow over the airfoil which enhances lift, reduces drag, delays flow separation and hence improves the overall efficiency. Generally, flow control methods are classified into active, passive and hybrid. In the recent years, the use of biomimetics in flow control techniques piqued interest among the researchers. One such fascinating biomimetic study based on the shark scale denticular structure is presented in this research paper. Studies suggest that the shark scale denticular morphology could act like a vortex generator [8]. Originally, the shark skin has dermal denticles (scales) that impede body separation by providing local flow separation control, thus helping the shark to swim faster in water with least drag which inspired the aerodynamic designers and wind engineers. Arunvinthan et al. [8] replicated shark scale structures inspired by short fin mako as vortex generators and reported that the Shark scale-based vortex generators have improved the lift coefficient of an airfoil by 3.8% and delays the stall by controlling the flow separation thus reducing the drag. Vortex generators are small aerodynamic devices that are designed to improve the flow characteristics by delaying boundary layer separation and enhancing mixing between different layers of airflow.

Experiments carried out by researchers [9] confirmed the improvements in lift-to-drag ratio of denticle-inspired surfaces when compared with best-reported traditional low-profile vortex generators. Lang et al. [10] claimed that these shark scale structures behave in a similar fashion to streamwise riblets thereby reducing turbulent skin friction drag by 9.9%. Later, it was discovered that the shark's bristling denticles function as a vortex generator, a passive flow control device. According to the previous research [11], the results revealed that cavities developed in between the denticles, producing embedded vortices that are in opposition to wake creation. Lang et al. evaluated a simplified 3D shark scale based embedded cavity model with a flat base featuring 90° angle. Research results revealed that such square cavity formed by the bristled shark skin could effectively create an interconnecting web of vorticity thus, rendering hydrodynamic benefit. This clearly shows that the bristled shark scale structure could potentially benefit as a flow control device. To gain more insight into the underlying flow physics, subsequently, Santos et al. [12] experimentally investigated the effectiveness of bristled

shark skin as an effective means for boundary layer flow control. Results revealed that the angled shark scale structure tends to create a rotational flow near the wall thus injecting momentum into the boundary layer thereby converts the adverse pressure gradient into a favourable one, thus prolonging the flow attachment over the surface. Subsequently several researchers like Natarajan et al. [13] focused their attention on analyzing the hydrodynamic characteristics of such shark scale-based vortex generators and reported that the shark scale-based structures not only help in reattaching flow but also helps reducing the overall drag. These studies confirm the hydrodynamic benefits of the shark scale structure as a viable passive flow control technique. However, the difficulty and the challenge lies in replicating them to real-world scenarios. Researchers adopted several fabrication methods [14] such as Bio-replicated forming method, Direct manufacturing method, Indirect manufacturing method etc. to mimic the shark scale structures. Bio-replicated forming methods involve micro-embossing [15], vacuum casting [16], elastomeric stamping, etc. The Micro-embossing technique involves pressing a patterned mould into the substrate under controlled heat and pressure. The vacuum casting method utilizes a mould kept in a vacuum chamber. The resin is filled under the vacuum condition which eliminates the air bubbles. Even though these methods are simple to process they are limited to biological resources. Hence these methods cannot be used for wind tunnel testing. Direct manufacturing method for surface microstructures like surface machining (Walsh & Lindemann, 1984) [17] or surface-scratching are relatively easy to process but the processing efficiency is poor. Additionally, to achieve the similar sizing as the real-world short fin Mako played a significant challenge with the Direct Manufacturing methods as it is complicated, time-consuming, and costly. Indirect method can be classified into photolithography [18], laser etching [19], 3D-printing [20]. Photolithography technique utilizes light to transfer the pattern onto a light-sensitive layer on the substrate. After which material is added or removed based on the pattern. Direct manufacturing methods have high efficiency in the processing but the processing involved is complex. Considering all these issues and challenges with the various manufacturing methods, it has been decided to 3D print shark scale structures to create an economically feasible way while producing similar accuracy.

Recently, researchers have started experimenting with shark scale structures as a viable alternative for passive flow control means because of its intrinsic working nature. However, it should be noted that most of the research were aimed at investigating the hydrodynamic characteristics and hence the aerodynamic influence remains still lack. Considering the Reynolds number similarity, it is obvious that the shark-scale structures if implemented on airfoil will tend to behave in a similar fashion rendering aerodynamic benefit. Therefore, in the present study, the

authors planned to investigate the aerodynamic characteristics of such bristled shark scale structure. In the recent years, novel biomimetic Leading-Edge Protuberanced (LEP) wing section spurred research interest among many researchers because of its aerodynamic robustness. Therefore, it has been decided to test the influence of bristled shark scale structure on the aerodynamic characteristics of LEP wing. Frank E. Fish [21] initiated the study on the LEP wing inspired from the Humpback whales. Fish morphologically investigated the flippers of the humpback whales and reported that the biologically inspired LEP based wind turbine blades possess 25% more airflow than the conventional blade. Subsequently, Fish and Watts [22] acquired a patent for this technology. Miklosovic et al. [23] experimentally evaluated flippers with and without LEP and reported that LEP model offers stall delay benefit. Several researchers studied on the effectiveness of LEP in terms of amplitude [24], [25], wavelength [26], [27] and incidence angle [28] and reported that modifying the wavelength of LEP has negligible effects on the performance, while change in amplitude and incidence angle poses significant effects on the performance. Hansen et al. [29,30] investigated the working mechanism of the LEP and reported that the LEP generate vortices which change in direction of rotation with the change in the leading-edge geometry cancelling out each other resulting in a shorter wake creating no additional drag penalty. [31] proposed a different theory and suggested the local variation in the

chord along the peak and the trough section of the LEP along with 3d spanwise flow results in non-uniform separation characteristics. Therefore, it provides extended flow reattachment even at greater angles thus providing aerodynamic robustness.

The present paper bridges both biomimetic studies of shark scale structures and humpback whale flippers. The primary objective of this study is to assess the aerodynamic performance characteristics of biologically inspired bristled shark scale structures on the novel LEP airfoil section. Therefore, in this present study, the influence of bristled shark scale structure as an effective means of flow control technique on the aerodynamic characteristics on the LEP wing was experimentally evaluated at $Re=1.71 \times 10^5$. While most of the previous literatures refers to the investigation of shark scale structures on hydrodynamics, the present paper aims at exploring the influence of such shark scale structures on the LEP airfoil as an aerodynamic flow control technique. To the extent of authors knowledge, this will be the first of its kind in the world exploring the effect of bristled shark scale structures on the aerodynamic characteristics of bio-inspired LEP airfoil section. Surface pressure distribution were examined to gain more insight into the underlying flow physics. It is believed that the present study will act as a pilot study in utilizing shark scale structures as a passive flow control device for airfoils and its real-world applications.

Table 1. Comparison of literature with the experimental findings

Author	Experimental / Numerical	Findings from the literature review	Relation with the Experimental findings
Arunvinthan et al. [8]	Experiment	Shark scale structures without bristling angle improve the lift coefficient of conventional airfoil by 3.8%.	The bristled shark scale structures improve the lift coefficient of LEP airfoil maximum by 44%.
Domel et al. [9]	Experiment	Shark denticles significantly improves the aerodynamics of conventional airfoils by extended flow attachment region thus enhancing lift with maximum of 323%.	The bristled shark scales enhance the aerodynamics of LEP airfoils by prolonged attachment of the flow (similar to conventional airfoils tested by Domel et al.).
Lang et al. [10,11]	Experiment	Bristled shark skin geometry effectively controls the flow separation and can be utilized as separation control mechanism.	Bristled shark scales extend the flow separation over the peak and trough section of the LEP airfoils which is evident from the CP vs x/C plots.
Santos et al. [12]	Experiment	Bristled shark skin control flow separation by eliminating adverse pressure gradient	Adverse pressure gradient formed at the peak section of LEP airfoil has been delayed by the incorporation of bristled shark scales. Therefore, bristled shark scale structures also delay adverse pressure gradient in LEP airfoils.
Natarajan et al. [13]; Wen et al. [20]	Experiment	Shark skin models were 3D printed using PLA material.	LEP airfoil section and bristled shark scale structures were 3D printed using PLA material.

MATERIALS AND METHODS

Experimental evaluations were performed to identify the effect of bristled shark scale structure affixed to biologically inspired LEP wing section at $Re=1.71 \times 10^5$. All the test cases were tested in the low-speed subsonic open-circuit wind tunnel facility available at SASTRA Deemed University. The cross-section of the rectangular test-section is $300 \times 300 \times 1500$ mm. The tunnel is activated by a fan powered by a 10HP motor and is capable of attaining a maximum speed of 60 m/s. The free-stream turbulence intensity at the test-section of the wind tunnel is 0.51%. A schematic representation of the wind tunnel with necessary equipment is shown in the Figure 1. NACA 63(4)-021 airfoil has been considered as the baseline airfoil profile in this study. The airfoil selection is based on the previous literatures. Several researchers have reported that NACA 63(4)-021 airfoil strongly reminiscent the cross-section of the flippers of humpback whale and hence NACA 63(4)-021 airfoil was chosen as the baseline airfoil profile. NACA 63(4)-021 airfoil is a member of 6-series airfoils and is symmetrical in nature. The detailed nomenclature based on the 6-series airfoil is as follows:

First digit “6” indicates the airfoil is from the 6-series. Second digit “3” describes the chordwise location of minimum pressure on the airfoil at the designed lift coefficient (Cl). This digit specifies that the minimum pressure is at 30% of the chord from the leading edge. Third digit “(4)” refers to the range of lift coefficient in which the favourable pressure gradient exists on both surfaces. Fourth digit “0” represents the design lift coefficient in tenths. Last two digits “21” represents the maximum thickness of the airfoil as a percentage of the chord length. In this case, the maximum thickness is 21% of the chord length.

Aiming at incorporating protuberances at the leading-edge, the sinusoidal protuberances of the baseline model was greatly influenced by two parameters namely amplitude (A) and wavelength (λ) were utilized as outlined in previous literatures [28], [32]. Since the present study focuses its attention to assess the effect of bristled shark scale on the LEP wing, the parameters A and λ of the sinusoidal LEP has been kept constant as 0.12C and 0.5C respectively throughout this study. The test model considered in this study has a mean chord length of chord length of 100 mm and span 300 mm the entire test-section thus making it as an infinite model. The baseline LEP model was then fabricated using Polylactic acid (PLA) material via 3D printing at a resolution of 100μ . Since the test model possess a varying chord, the surface pressure was measured at two different locations namely peak (i.e., chord maxima) and the trough (i.e., chord minima). The surface pressure tapings were equi-distributed over the LEP wing. A total of 50 pressure taps were made over the airfoil surface. 20 pressure taps were equi-distributed over the peak region and 16 pressure taps over trough region while remaining taps were distributed along the span of the model. The spacing between each taping is 9mm and the diameter of each pressure tap is approximately 1 mm.

The fabrication of bristled shark scale involves 3D printing utilizing the same poly lactic acid material. The shark scale models featuring chord length of 10 mm and span 6 mm with leading-edge amplitude of 2 mm has a bristling angle of 20° . The shark scale model is resting in a pillar with a height of 5 mm as shown in Figure (c) & (d). This 3D printed bristled shark scale models were affixed to the baseline LEP wing at 60% of the mean chord length (hereafter named as M1). To gain some insights on the effect of shark scale structures on the flow over the LEP wing, the structures were also arranged continuously from 60% to

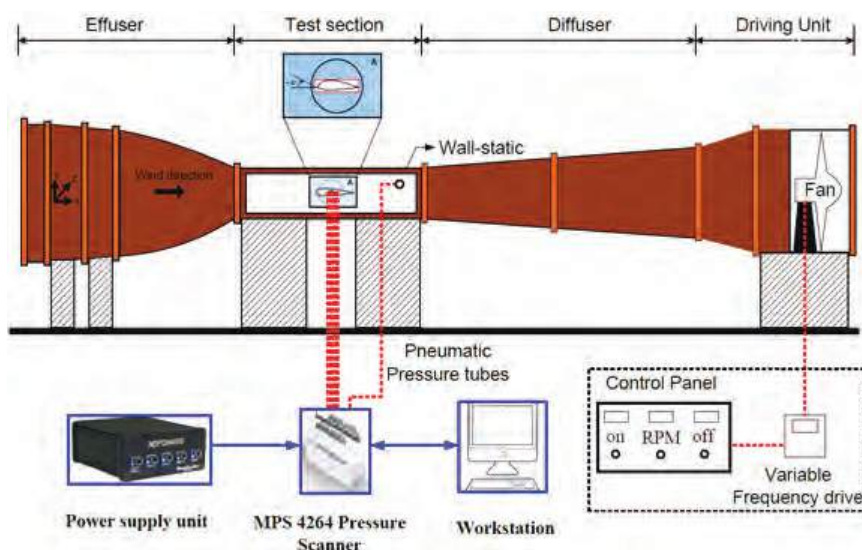


Figure 1. Schematic representation of wind tunnel setup.

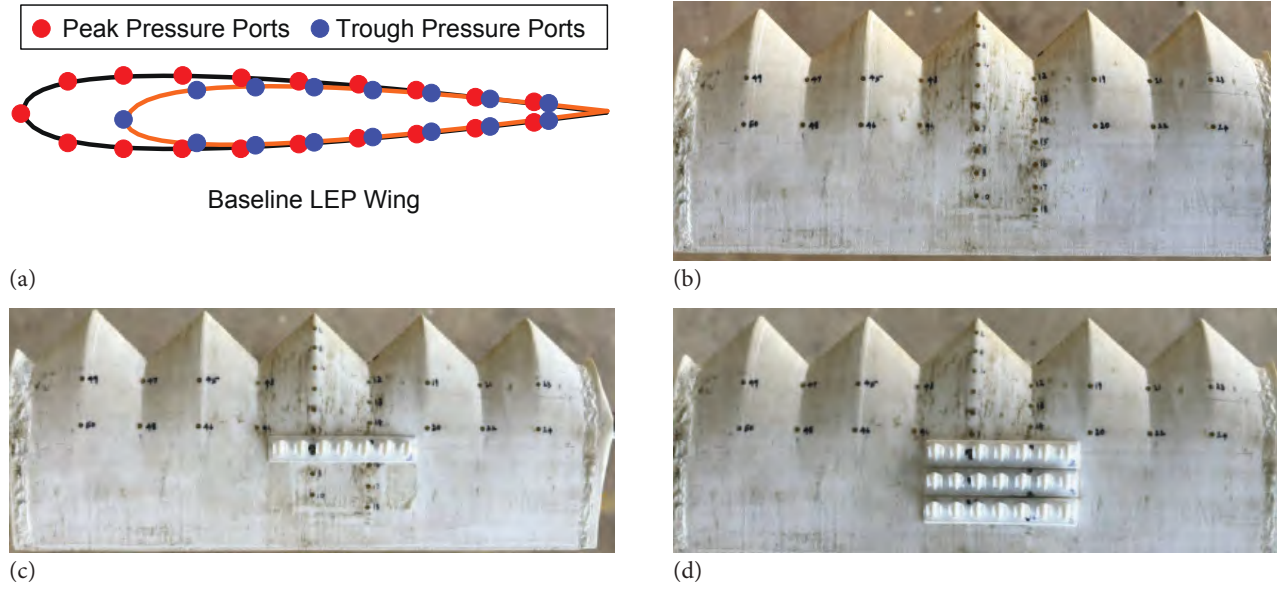


Figure 2. (a) Schematics of distribution of pressure ports (b) Real image of LEP baseline airfoil (c) Real image of modified M1 model (d) Real image of modified M2 model.

80% of the mean chord length of the LEP wing (hereafter named as M2). A visual representation is shown in the Figure 2. The surface pressure of the test models were then measured through the pressure taps which are pneumatically connected to simultaneous MPS4264 Scanivalve pressure scanner. The aerodynamic lift, drag forces and surface pressure distribution acting over the modified and unmodified equivalents were then estimated using pressure-integration technique [33-35] to yield lift, drag and pressure coefficients.

$$F_D = \sum (\Delta P) \times S_i \times \cos\left(a + \frac{\pi}{180}\theta\right) \quad (1)$$

$$F_L = \sum (\Delta P) \times S_i \times \sin\left(a + \frac{\pi}{180}\theta\right) \quad (2)$$

$$C_D = F_D / 0.5\rho v^2 s \quad (3)$$

$$C_L = F_L / 0.5\rho v^2 s \quad (4)$$

$$C_P = \Delta P / 0.5\rho v^2 s \quad (5)$$

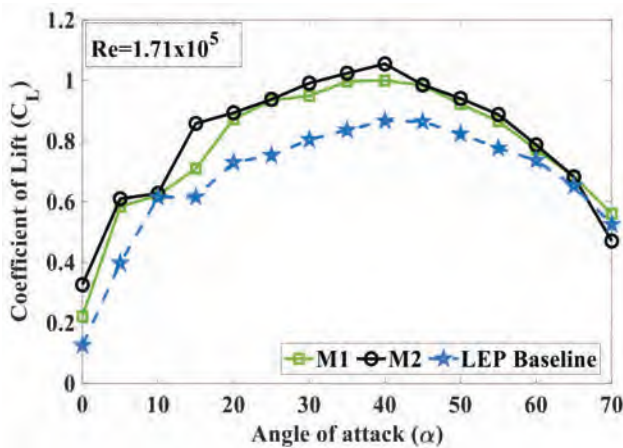
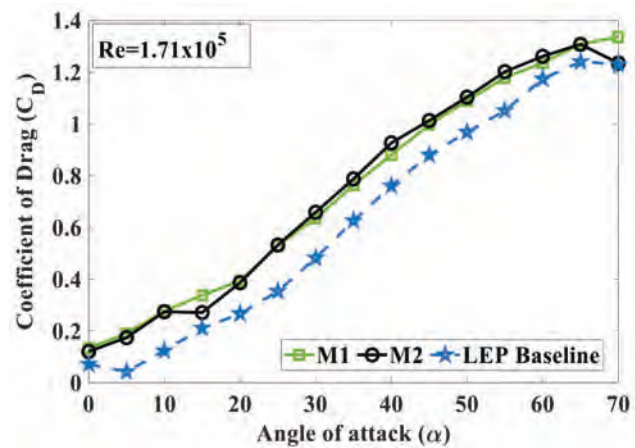
The uncertainties involved in the experiments like Buoyancy, solid blockage, wake blockage etc. and the instrumental uncertainties included in the present study along with their correction factors are tabulated as Table 2.

RESULTS AND DISCUSSION

The shark scale models affixed to the clean baseline LEP wing was primarily evaluated to identify its influence on the flow characteristics and aerodynamic behaviour of the bio-inspired LEP wing section. The modified models (M1 and M2) were tested in the wind tunnel using surface pressure measurement at $Re=1.71 \times 10^5$. The results obtained for the modified models were compared against the baseline LEP wing section as shown below. Figure 3 represents the variation of time-averaged coefficient of lift (C_L) vs the angle of attack (α) for all the test cases operating at $Re=1.71 \times 10^5$. In the figure, square denotes the C_L values for model M1, circle denotes the C_L values for model M2 and star denotes the C_L values for LEP Baseline model. It can be observed from the graph that the C_L curve gradually increases in the direction of increasing angle of attack until $\alpha = 40^\circ$. For instance, it can be observed that the baseline LEP model exhibits a maximum lift coefficient ($C_{L,max}$) of 0.84 at $\alpha = 40^\circ$. Beyond which with the further increase in the angle of attack, the lift coefficient gradually decreases. It can be easily seen from the figure that both the bristled shark scale models (M1 and M2) affixed to LEP wing exhibits significantly higher lift coefficient at all angles of attack than its unmodified equivalent. Therefore, it becomes clear that the presence of 3D printed shark scale over the LEP wing alters the flow characteristics by effectively inducing the momentum into the boundary layer and thus rendering aerodynamic benefits in terms of C_L increment. For instance, M2 model exhibits the maximum lift coefficient (C_L) of 0.99 which is 17.8% higher than the baseline model. At the same $\alpha = 40^\circ$, the modified M1 model exhibits a maximum lift coefficient of about 1.05 representing lift

Table 2. Uncertainties and their correction factor

Uncertainty	Correction method	Substitution	Correction factor
Buoyancy correction	Glauert method: $D_B = \frac{1}{2} \pi \lambda_2 t^2 P'$	$\lambda_2 = 1.4$ [36] $t = 0.021 \text{ m}$ $P' = 0.01 \text{ N/m}^2$	0.00000969
Solid blockage correction	Thom's method: $\varepsilon_{sb} = \frac{K_1(\text{Model Volume})}{c^{3/2}}$ Model Volume = $0.7 \times \text{model thickness} \times \text{model chord} \times \text{model span}$;	$K_1 = 0.74$ Model Volume = 0.000441 $c = 0.09$	0.0120
Wake blockage correction	Allen and Vincentti method: $\Delta C_{d,wb} = \Lambda \sigma$ $\sigma = \left(\frac{\pi^2}{48} \right) \left(\frac{C}{h} \right)^2$	$\Lambda = 0.36$ [36] $\sigma = 0.0616$	0.0221
Instrument and Data Error	MPS4264 miniature Scanivalve pressure scanner – OEM error intimation	-	full-scale error of $\pm 0.06\%$

**Figure 3.** Aerodynamic lift coefficient (C_L) vs Angle of attack (α)**Figure 4.** Aerodynamic drag coefficient (C_D) vs Angle of attack (α)

increment of 25% when compared against the LEP baseline model. The similar trendline can be observed in the pre-stall angles as well. For instance, at $\alpha = 15^\circ$ the modified model M1 and M2 displays a $C_{L,max}$ of 0.85 and 0.70 which is 44% and 18.6% higher than baseline LEP respectively. Based on the obtained experimental results, it can be summarized as the addition of shark scale structures over the LEP baseline can augment the lift characteristics of the LEP wing section.

Figure 4 shows the comparison of drag characteristics of the modified models with the unmodified equivalent for various α . In the figure, square denotes the C_L values for model M1, circle denotes the C_L values for model M2 and star denotes the C_L values for LEP Baseline model. It is important to note here that the drag illustrated in this study corresponds to pressure drag alone and skin friction drag is not

considered. It is notable from the figure that the baseline LEP exhibits the lowest drag than the modified models throughout all angles of attack. It could be understood that bristled shark scale structures induces turbulence creating perturbations in the flow thereby resulting in higher pressure drag. To gain more insights on the aerodynamic behaviour, surface pressure distribution were plotted against the chordwise position (x/C) for $\alpha = 5^\circ$ as shown in the Figure 5, 6. As the test model is a non-constant chord model featuring varying chord length along the chord-maxima and the chord-minima section, it becomes important to investigate the pressure distribution over both the chord-maxima and chord-minima region as shown in Figure 5, 6.

The surface pressure distribution in terms of pressure coefficient (C_p) plotted against x/C for test models at $\alpha=5^\circ$ as presented in Figure 5, 6. It becomes clear from the figure

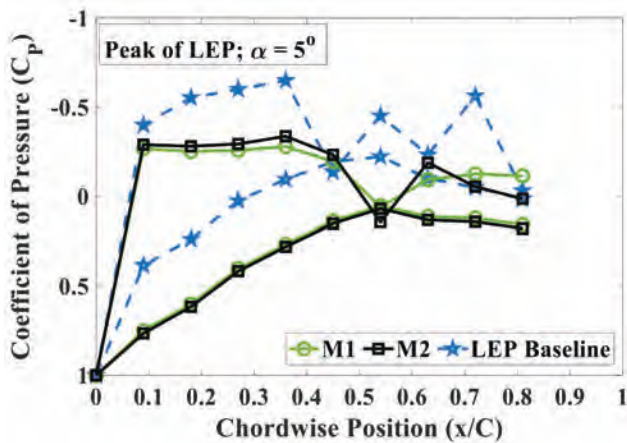


Figure 5. Surface pressure distribution along the chordwise position (x/C) for $\alpha = 5^\circ$ (peak region)

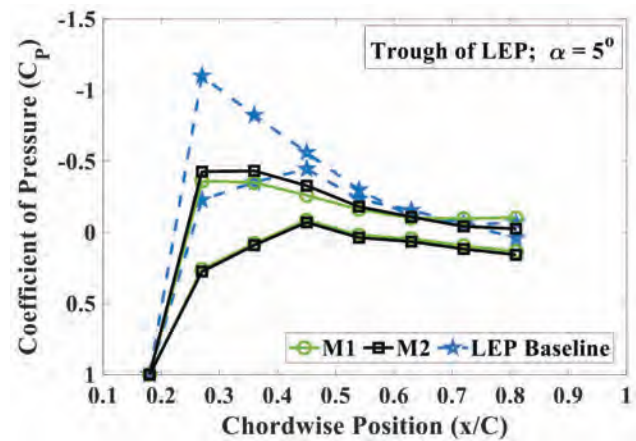


Figure 6. Surface pressure distribution along the chordwise position (x/C) for $\alpha = 5^\circ$ (trough region)

that the flow over the trough region remains attached in comparison against the peak region. This is because of the increased flow velocity over the trough region. For instance, it is notable from the figure 6 that peak negative suction pressure is seen at the trough section indicating that the majority of the freestream flow goes behind the trough region. It is understandable that a strong acceleration occurs at trough region when compared against the peak region thereby creating a spanwise flow. Evidently, the suction pressure on the trough region is lesser than the peak region indicating the accelerated flow at trough section. For Instance, if the velocity behind the trough section is very high, then the pressure will be relatively low in comparison against the peak region. However, as its well known that, when the flow over the peak is relatively smaller than the trough section, the high-pressure region from the peak tends to move to the low-pressure suction region happening behind the trough section thereby leading to reenergize the trough flow sacrificing the peak flow. This spanwise flow induced by the change in the pressure over the peak and the trough section keeps the flow attached over the trough section of the LEP. In simpler words, it could be explained that the spanwise pressure gradient created between the peak and the trough section draws low-inertial boundary layer molecules from the peak. This could be observed from the Figure 5 that at around $0.4C$ for the peak section, an increase in the pressure can be seen. As it is known that the increase in the pressure signifies reduction in velocity, this can be attributed to the following explanation: as the low-inertial boundary layer molecules from the peak are drawn towards the trough region an increase in the pressure is felt over the peak region at $0.4C$. Studies [37] suggest that when a low-inertial boundary layer molecules are transported away because of the spanwise pressure gradient, the flow will be replaced by a high-momentum fluid drawn from above fluid layers thereby reenergizing the flow. This in turn holds in good agreement with the surface pressure characteristics for the baseline LEP wing displayed in Figure 5 which

indicates the flow reattachment signified by the decrease in pressure around $0.5C$. Subsequently the flow undergoes similar phenomena at $0.7-0.8C$ as well. Therefore, based on these results, it can be claimed that the low-inertial boundary layer molecules driven from the peak section transported towards the trough section by the spanwise pressure gradient is held responsible for the intended aerodynamic benefit. In the case of the modified models affixed with shark scale structures, it is clearly evident that the peak negative suction pressure observed over both the peak and the trough section is significantly lesser when compared against the baseline LEP model. For instance, M1 model attains a peak negative pressure of -0.33 and -0.43 in the peak region and trough region respectively. It is speculated that the reduction in the flow acceleration over the upper surface of the modified LEP model is due to the presence of the shark scale structures. As it is well known that the pressure disturbances can propagate upstream in subsonic flow, the changes in the local pressure distribution induced by the shark scale structure influences the flow characteristics over the suction side of the airfoil. In other words, the reorganization of the flow upstream to accommodate the downstream disturbance induced by the shark scale structures results in the reduction of flow acceleration over the suction side of the airfoil signifying reduction in the peak negative suction pressure. Furthermore, it can be claimed that with the incorporation of shark scale structures the spanwise flow from peak to trough changes considerably. For instance, it could be seen that for the baseline LEP model, the peak negative suction pressure at the peak and the trough section corresponding to $0.1C$ is -0.40 and -1.09 whereas for the modified models M2 and M1, the peak negative suction pressure at $0.1C$ for the peak and the trough section reduces to -0.29 & -0.42 (for M2) and -0.26 and -0.36 (for M1) respectively. This clearly quantifies the reduction in the spanwise pressure gradient existing between the peak and the trough section for the modified models. In simple words, roughly 80% of the spanwise pressure gradient has reduced

with the introduction of the shark scale structures resulting in lesser suction at the trough region in comparison with the baseline LEP model. The reduction in the spanwise pressure prolongs the flow attachment regime over the peak section by limiting the transportation of the low-inertial boundary layer molecules from the peak section. However, as the name itself indicates “low-inertial boundary layer molecules” eventually they separate after a short while. Results indicate that following the peak negative suction pressure for the baseline LEP model at $0.1C$, the C_p increases with the increase in the chordwise position till $0.4C$ indicating the flow acceleration. Beyond $0.4C$ the pressure coefficient increases abruptly indicating the reduction in the flow velocity. It is speculated that this reduction in the flow velocity over the baseline LEP model is induced by the spanwise flow. However, in the case of the modified LEP models affixed with shark scale structures, as nearly 80% of the spanwise flow is diminished, the flow continues to attach till $0.5C$ for both the M1 and M2 models. Consequently, as the model M1 has even more reduction in the spanwise pressure gradient (approximately 85%) a similar pressure coefficient trendline can be observed but with more gradual slope (between 0.5 - $0.6C$) in comparison against the M2. Therefore, it becomes clear that with the incorporation of shark scale structures on the LEP wing, the shark scale structure reorganizes the flow altering it favourably redistributing the majority of the spanwise pressure

gradient in terms of prolonged flow attachment over peak section provides aerodynamic benefit. Furthermore, the alternate separation and reattachment pattern observed at the baseline LEP wing is stabilized with the introduction of shark scale structures as seen from the figure. It is believed that the small vortices induced by the shark scale structures mix higher-momentum fluid into the boundary layer thereby energizing the boundary layer, making it more resistant to separation. The delayed separation along with enhanced flow attachment observed over the modified models with shark scale structure increases the pressure gradient between the suction side and the pressure side of the model thereby resulting in higher lift coefficients in the modified model. It is of interest to note that out of the modified models M1(single-strip) and M2 (three-consecutive strips), M1 has the least drag profile. To further ascertain this behaviour, surface pressure contours were plotted for both the baseline and the modified M1 model is depicted in Figure 7-10.

It can be clearly seen from the figure that for the baseline LEP model, over the peak region (Fig. 7), followed by an initial increase in the pressure at the leading-edge signifying the stagnation point, with the acceleration in the flow, the pressure gradually decreases. But it could be observed at around $0.05m$ chordwise position, the pressure increases indicating the flow separation induced by the spanwise flow. Following which the flow reattaches and separates alternatively as

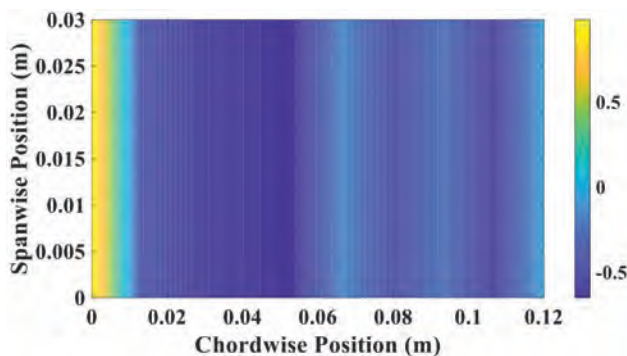


Figure 7. Surface pressure contour over LEP baseline peak at $\alpha=5^\circ$

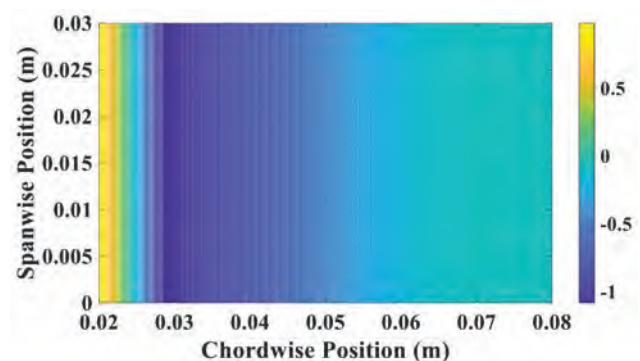


Figure 8. Surface pressure contour over LEP Baseline trough at $\alpha=5^\circ$

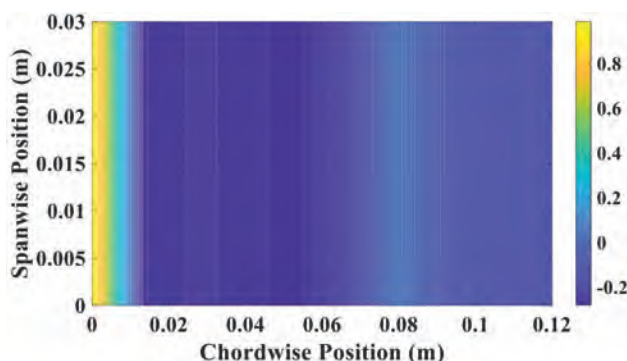


Figure 9. Surface pressure contour over M1 peak at $\alpha=5^\circ$

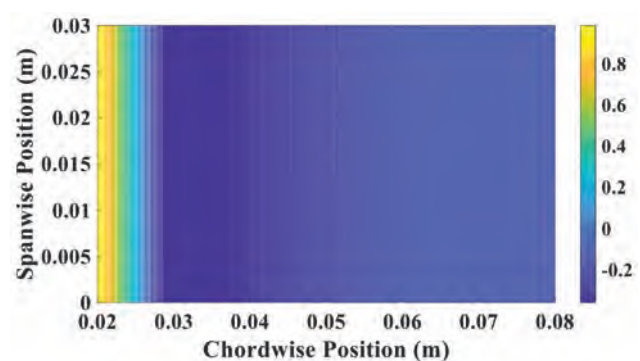
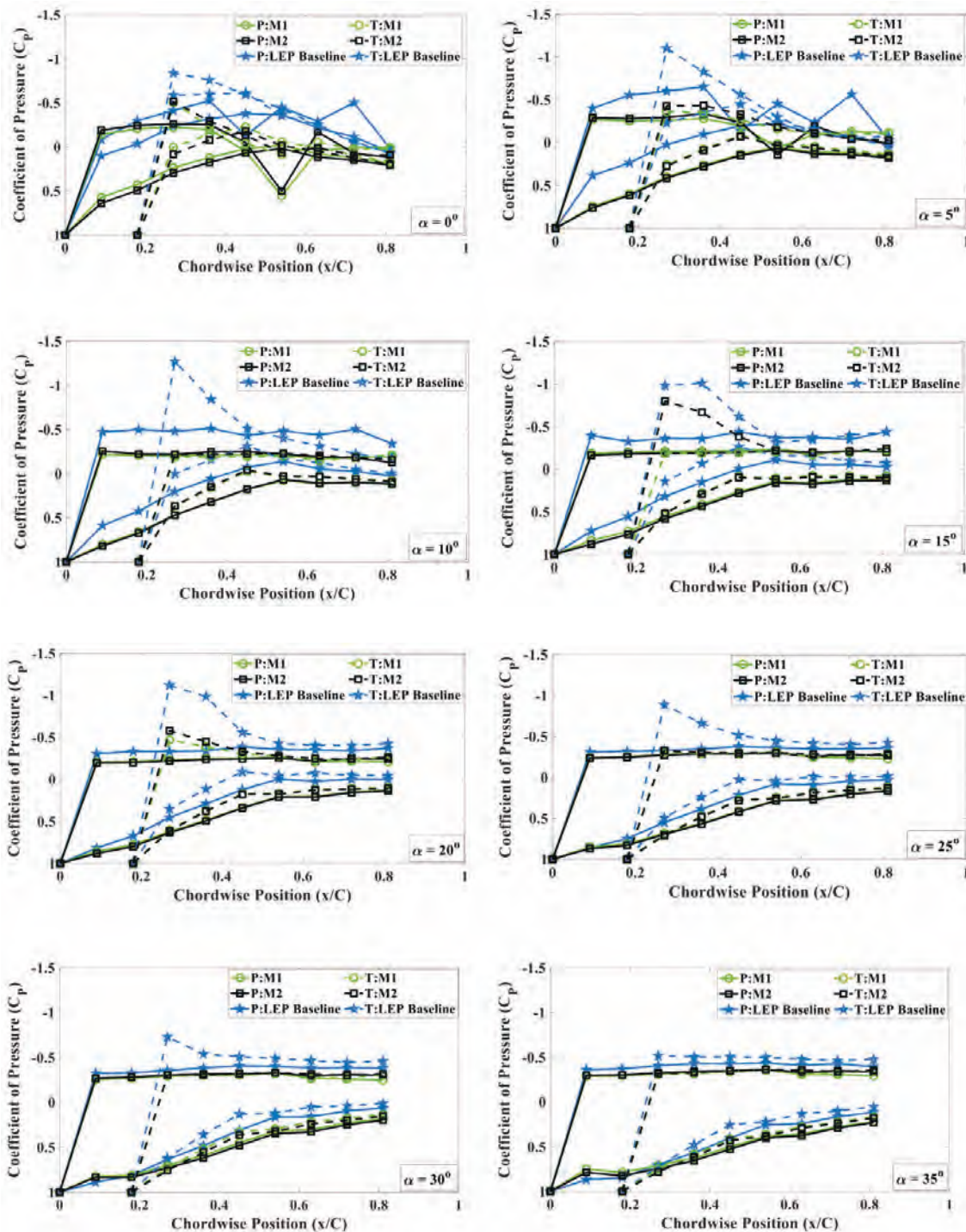


Figure 10. Surface pressure contour over M1 trough at $\alpha=5^\circ$

indicated in the Figure 6. On the other hand, the flow remains continuously attached over the suction side of the airfoil at the trough section as shown in Figure 8. Similarly, the peak and trough surface Pressure contours were plotted for the modified model with bristled shark scale structures (M1) in the Figure 9, 10. In the case of the M1 model, it is evident that the flow instability that previously occurred near the vicinity of the trailing edge in the LEP baseline model

disappears completely. Furthermore, the non-uniform separation characteristics observed in the LEP baseline gets reorganized and redistributed with the incorporation of bristled shark scale structures. It should however be noted that the presence of the bristled shark scale structures in fact give a slight increase in the pressure ahead of the model. The reduction in the spanwise flow with the introduction of shark scale structures keeps the flow attached over both the peak and the



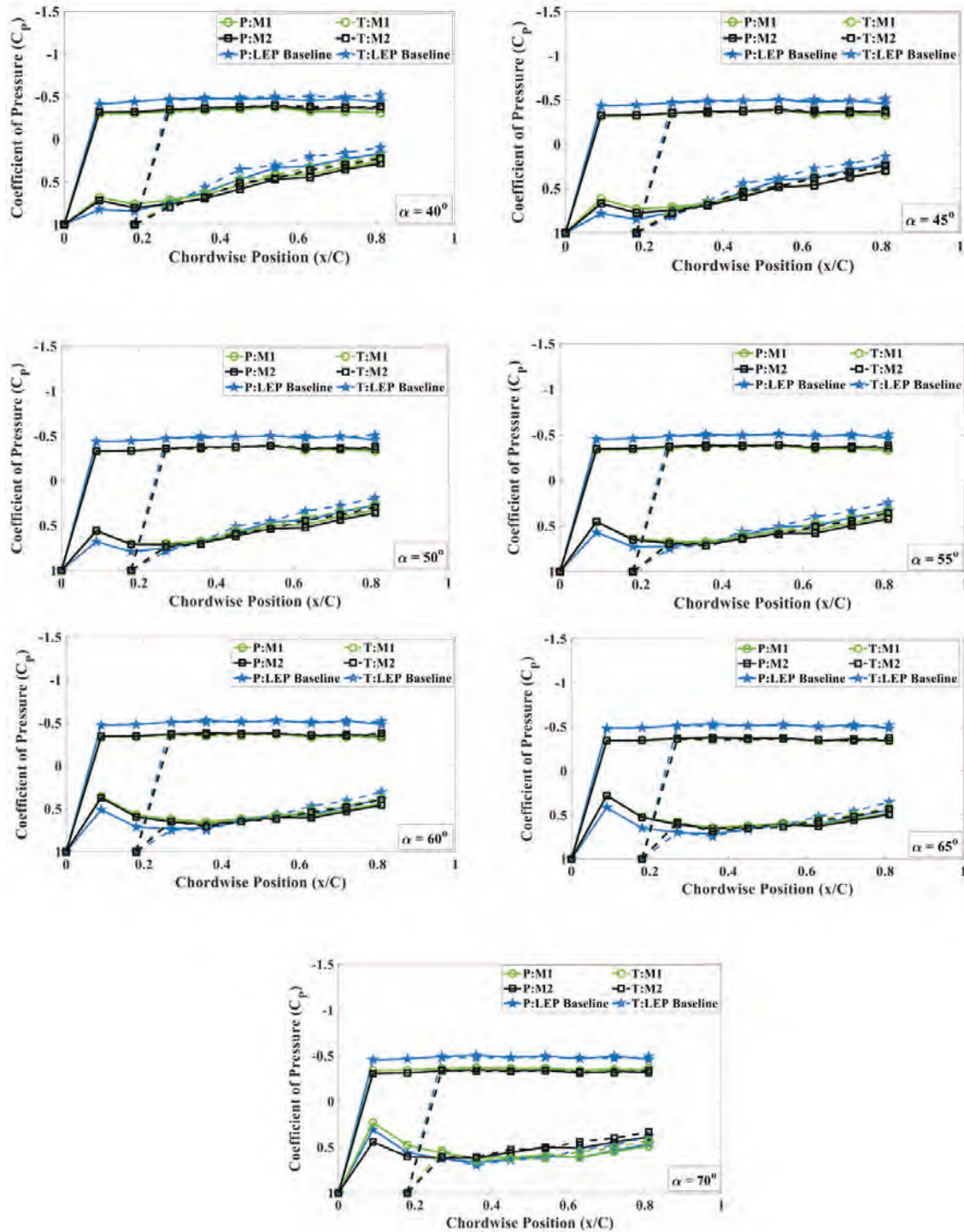


Figure 11. Surface pressure distribution along the chordwise position (x/C) from $\alpha = 0^\circ$ to 70°

trough for a longer distance thereby providing delay in flow separation.

Figure 11 compares the pressure distribution over the baseline LEP and the modified models (M1 and M2) for various α from 0° to 70° . The solid green line indicates the pressure distribution over the peak section of M1 model, solid black line refers the pressure distribution over the

peak section of M2 model while solid blue line for peak section of LEP baseline. Similarly, dashed line represents the pressure distribution over trough section of modified models and the LEP baseline model. Since LEP is a non-constant chord model, as discussed in the previous section, it could be seen that both the peak and the trough region experiences an altogether different flow pattern. Results reveal

that the extended flow attachment is seen over trough region rather than the peak section for the modified and LEP baseline models at lower α values. It is of interest to note that despite the addition of bristled shark scale structures on the modified model, the non-uniform separation characteristics prevalent over the baseline LEP wing could be still observed on the modified models. However, the addition of bristled shark scale structure influences the surface pressure characteristics, by energizing the boundary layer through the vortices produced by the shark scales. This in turn increases the net pressure gradient of the airfoil. At very larger values of α , the flow gets separated completely over both peak and trough section of the unmodified and modified equivalents. The graph plotted in the Fig. 11 reveals that the pressure gradient existing between the suction and the pressure side for the modified models (M1 and M2) is larger in comparison with the unmodified equivalent. The flow is attached over both peak and trough section of the bristled shark scale affixed models at $\alpha = 0^\circ$ & 5° . From $\alpha = 10^\circ$ to 20° , the flat section at the peak region depicts the separated flow whereas the trough section exhibits attached flow over the modified models as well as unmodified test case. The plateau region presented in the C_p vs x/C plots indicate constant pressure region, which in other words can be expressed as the flow separation regime. Though, the flow has been separated, the pressure gradient for the modified models is higher when compared against the baseline LEP model, thereby resulting in higher lift coefficients. Overall, it can be concluded that the modified shark scale models effectively alters the flow characteristics providing delayed separation, enhanced flow attachment, stable flow structure in proximity with the trailing edge and thus, rendering aerodynamic benefits.

CONCLUSION

The concluding remarks and findings are as follows:

- Bristled shark scale structured airfoil models exhibits enhanced lift characteristics at both the pre-stall and post-stall angles against its unmodified equivalent.
- At pre-stall angles ($\alpha = 15^\circ$), the modified model M1 (single strip at 0.6C) exhibits a maximum lift increment of 44%, while M2 (Shark scale strips between 0.6-0.8C) shows a peak lift increment of 18.6% in comparison against the LEP baseline model.
- At post-stall angles ($\alpha = 40^\circ$), M1 and M2 exhibits a maximum lift increment of 25% and 17.8% against the LEP baseline model.
- Surface pressure distribution over the test models with bristled shark scale structure clearly shows that the presence of shark scale structure effectively alters the flow characteristics.
- Results reveal that small vortices induced by the shark scale structures mix higher-momentum fluid into the boundary layer thereby energizing the boundary layer, making it more resistant to separation in addition to

the 85% reduction in the span wise flow, makes the modified model outperform the baseline unmodified equivalent.

Experimental results proved that the bristled shark scale structures can effectively alter the aerodynamic characteristics and therefore can be used as a viable passive flow control device. One of the challenges involved in this present study is the dislodging of 3D printed bristled shark scale structures affixed to the LEP wing section at high Reynolds number. Therefore, in the future, instead of affixing 3D Printed shark scale structures they should be printed directly over the airfoil section itself. But it should be noted that this will increase the model count and cost incurred for the research with the testing of different patterns and sizes of bristled shark scale structures. Attempts to decode the underlying flow physics in detail should be entertained in the near future to potentially utilize this technology in real-world applications. Since the experimental evaluation of biomimetic LEP airfoil with shark scale structures yielded a voluminous amount of pressure data, statistical analyses like 0-1 test, recurrence quantification analysis (RQA) etc. will also be a potential future work.

NOMENCLATURE

LEP	Leading-edge protuberance
θ	Angle of incidence on the i^{th} port ($^\circ$)
C_D	Coefficient of drag
VG	Vortex Generator
SSVG	Shark Scale Vortex Generator
SEM	Scanning Electron Microscopy
λ	Wavelength (m)
A	Amplitude (m)
C	Chord length (m)
PLA	Polylactic acid
v	Velocity (m/s)
M1	Bristled shark scale at 60% of the mean chord length
λ_2	body shape factor
P'	slope of longitudinal static pressure gradient curve
$\Delta C_{d,wb}$	Wake blockage correction
Λ	body shape factor
h	Tunnel height (m)
S_i	Area (m^2)
α	Angle of attack ($^\circ$)
C_L	Coefficient of lift
C_p	Coefficient of pressure
x/C	Chordwise location
Re	Reynolds Number
ΔP	Net Pressure (N)
F_L	Lift force (N)
F_D	Drag force (N)
ρ	Density of the fluid (kg/m^3)
s	Reference area (m^2)
M2	Bristled shark scale at 60% to 80% of the mean chord length

t	body thickness (m)
D_B	Glauert's buoyancy correction factor
c	test-section area (m ²)
K_1	Constant value for wing spanning the tunnel breadth

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Hussein EQ, Azziz HN, Rashid FL. Aerodynamic study of slotted flap for NACA 24012 airfoil by dynamic mesh techniques and visualization flow. *J Therm Eng* 2021;7:230-239. [\[CrossRef\]](#)
- [2] Mahmoud H. Stability of turbine blades, aircraft wings and their acoustic radiation. *J Therm Eng* 2015;1. [\[CrossRef\]](#)
- [3] Ayli E, Kocak E, Turkoğlu H. Numerical investigation of rod-airfoil configuration aeroacoustic characteristics using Ffowcs-Williams-Hawkings equations. *J Therm Eng* 2021;7:58-70. [\[CrossRef\]](#)
- [4] Alpman E. Aerodynamic performance of small-scale horizontal axis wind turbines under two different extreme wind conditions. *J Therm Eng* 2015;1:420-432. [\[CrossRef\]](#)
- [5] Maheri A. Simulation of wind turbines utilising smart blades. *J Therm Eng* 2016;2:557-565. [\[CrossRef\]](#)
- [6] Boumehani A, Noura B, Kerfah R, Khelladi S, Dobrev I. Numerical investigation of the blade profile effect on the aerodynamic performance of a vertical-axis wind turbine Darrieus H-rotor. *J Therm Eng* 2020;6:388-396. [\[CrossRef\]](#)
- [7] Şumnu A, Güzelbey İH. The effects of different wing configurations on missile aerodynamics. *J Therm Eng* 2023;9:1260-1271. [\[CrossRef\]](#)
- [8] Arunvinthan S, Raatan VS, Nadaraja Pillai S, Pasha AA, Rahman MM, Juhany KA. Aerodynamic characteristics of shark scale-based vortex generators upon symmetrical airfoil. *Energies (Basel)* 2021;14:71808. [\[CrossRef\]](#)
- [9] Domel AG, Saadat M, Weaver JC, Haj-Hariri H, Bertoldi K, Lauder GV. Shark skin-inspired designs that improve aerodynamic performance. *J R Soc Interface* 2018;15:139. [\[CrossRef\]](#)
- [10] Lang AW, Hidalgo P. Cavity flow characterization of the bristled shark skin microgeometry. *Bioinspir Biomim* 2008;3:046005. [\[CrossRef\]](#)
- [11] Lang AW, Motta P, Hidalgo P, Westcott M. Bristled shark skin: A microgeometry for boundary layer control? *Bioinspir Biomim* 2008;3:046005. [\[CrossRef\]](#)
- [12] Santos LM, Lang A, Wahidi R, Bonacci A, Gautam S, Parsons J. The effect of shortfin mako shark skin at the reattachment of a separated turbulent boundary layer. *Bioinspir Biomim* 2024;19:5. [\[CrossRef\]](#)
- [13] Natarajan E, Freitas LI, Rui Chang G, Abdulaziz Majeed Al-Talib A, Hassan CS, Ramesh S. The hydrodynamic behaviour of biologically inspired bristled shark skin vortex generator in submarine. *Mater Today Proc* 2021;46:3945-3950. [\[CrossRef\]](#)
- [14] Pu X, Li GJ, Liu YH. Progress and perspective of studies on biomimetic shark skin drag reduction. *ChemBioEng Rev* 2016;3:26-40. [\[CrossRef\]](#)
- [15] Han X, Zhang D. Study on the micro-replication of shark skin. *Sci China Ser E: Technol Sci* 2008;51:890-896. [\[CrossRef\]](#)
- [16] Zhao DY, Huang ZP, Wang MJ, Wang T, Jin Y. Vacuum casting replication of micro-riblets on shark skin for drag-reducing applications. *J Mater Process Technol* 2012;212:198-202. [\[CrossRef\]](#)
- [17] Walsh M, Lindemann A. Optimization and application of riblets for turbulent drag reduction. In: 22nd Aerospace Sciences Meeting. Reston, Virginia: American Institute of Aeronautics and Astronautics; 1984. [\[CrossRef\]](#)
- [18] Schumacher JF, Favier A, Pinelli A, Piomelli U. Engineered antifouling microtopographies—effect of feature size, geometry, and roughness on settlement of zoospores of the green alga *Ulva*. *Biofouling* 2007;23:55-62. [\[CrossRef\]](#)
- [19] Bixler GD, Bhushan B. Shark skin inspired low-drag microstructured surfaces in closed channel flow. *J Colloid Interface Sci* 2013;393:384-396. [\[CrossRef\]](#)
- [20] Wen L, Weaver JC, Lauder GV. Biomimetic shark skin: Design, fabrication and hydrodynamic function. *J Exp Biol* 2014;217:1656-1666. [\[CrossRef\]](#)
- [21] Fish FE. Influence of hydrodynamic design and propulsive mode on mammalian swimming energetics. *Aust J Zool* 1994;42:1-16. [\[CrossRef\]](#)

- [22] Google Patent. Scalloped wing leading edge. Jun. 2000. Available at: <https://patents.google.com/patent/US6431498B1/en> Accessed Sep 04, 2025.
- [23] Miklosovic DS, Murray MM, Howle LE, Fish FE. Leading-edge tubercles delay stall on humpback whale (*Megaptera novaeangliae*) flippers. *Phys Fluids* 2004;16:L39-L42. [CrossRef]
- [24] Custodio D, Henoch CW, Johari H. Aerodynamic characteristics of finite span wings with leading-edge protuberances. *AIAA J* 2015;53:1878-1893. [CrossRef]
- [25] Zhang MM, Wang GF, Xu JZ. Aerodynamic control of low-Reynolds-number airfoil with leading-edge protuberances. *J AIAA J* 2013;51:1960-1971. [CrossRef]
- [26] Yoon HS, Hung PA, Jung JH, Kim MC. Effect of the wavy leading edge on hydrodynamic characteristics for flow around low aspect ratio wing. *Comput Fluids* 2011;49:276-289. [CrossRef]
- [27] Kim MJ, Yoon HS, Jung JH, Chun HH, Park DW. Hydrodynamic characteristics for flow around wavy wings with different wave lengths. *Int J Nav Arch Ocean Eng* 2012;4:447-459. [CrossRef]
- [28] Arunvinthan S, Gouri P, Divysha S, Devadharshini RK, Nithya Sree RN. Effect of trough incidence angle on the aerodynamic characteristics of a biomimetic leading-edge protuberanced (LEP) wing at various turbulence intensities. *Biomimetics* 2024;9:354. [CrossRef]
- [29] Hansen KL, Kelso RM, Dally BB. Performance variations of leading-edge tubercles for distinct airfoil profiles. *AIAA J* 2011;49:185-194. [CrossRef]
- [30] Analysis of the streamwise vortices generated between leading edge tubercles. Available at: https://www.researchgate.net/publication/258246709_Analysis_of_the_Streamwise_Vortices_Generated_Between_Leading_Edge_Tubercles Accessed on Nov 25, 2024.
- [31] Flows in films and over flippers.. Available at: https://www.researchgate.net/publication/253904188_Flows_in_films_and_over_flippers Accessed on Nov 25, 2024
- [32] Arunvinthan S, Nadaraja Pillai S, Cao S. Aerodynamic characteristics of variously modified leading-edge protuberanced (LEP) wind turbine blades under various turbulent intensities. *J Wind Eng Ind Aerodyn* 2020;202:104188. [CrossRef]
- [33] Anderson JA. *Fundamentals of Aerodynamics*. Michigan: McGraw-Hill; 2001.
- [34] Li Q, Kamada Y, Maeda T, Murata J, Nishida Y. Effect of turbulent inflows on airfoil performance for a horizontal axis wind turbine at low reynolds numbers (part I: Static pressure measurement). *Energy* 2016;111:701-712. [CrossRef]
- [35] Arunvinthan S, Nadaraja Pillai S. Aerodynamic characteristics of unsymmetrical aerofoil at various turbulence intensities. *Chin J Aeronaut* 2019;32:2395-2407. [CrossRef]
- [36] Barlow JB, Rae WH, Pope A. *Low-speed Wind Tunnel Design*. New York: John Wiley Sons; 1999.
- [37] Skillen A, Revell A, Favier J, Pinelli A, Piomelli U. Investigation of wing stall delay effect due to an undulating leading edge: An LES study. In: 22nd Aerospace Sciences Meeting. Reston, Virginia: American Institute of Aeronautics and Astronautics; 2013. [CrossRef]



Research Article

Experimental investigation of the impact of electronic expansion valve opening on vcr system performance: comparative analysis with capillary tube and thermostatic expansion valve

Ankitsinh CHAUHAN¹, Ashok PAREKH¹, Vimal PATEL^{1,*}

¹Department of Mechanical Engineering, S. V. National Institute of Technology, Surat 395007 Gujarat, India

ARTICLE INFO

Article history

Received: 21 May 2024

Revised: 29 August 2024

Accepted: 05 September 2024

Keywords:

VCR System, Expansion Devices, Performance Analysis of VCR System, Comparison of Expansion Devices, EXV Openings

ABSTRACT

The Electronic Expansion Valve controls the amount of refrigerant flow into the evaporator. The percentage opening of the electronic expansion valve significantly impacts the performance of the vapor compression refrigeration system as it controls the refrigerant flow to the evaporator which alters the superheating temperature and thus providing different cooling loads. In order to protect the compressor from damage from the liquid refrigerant, it is essential to vaporize all refrigerant before it leaves the evaporator. The experiments were conducted with an air-cooled condenser, a compressor frequency of 45 Hz, and a water pump frequency of 50 Hz. The experiments ran until the cold water temperature reached 10°C. R-134a refrigerant was used as the working fluid in all experiments. The conditions were kept similar for the capillary tube, thermostatic expansion valve, and electronic expansion valve. The performance parameters including cooling load, compressor power, and cooling coefficient of performance, were investigated by varying the electronic expansion valve percentage opening. The best performance achieved was that of a cooling load of 6560 W and COP of 4.56, with a 60% electronic expansion valve opening. In comparison to the capillary tube, the 60% electronic expansion valve opening provides an increase of 65% of the cooling load and increase of 39% of the coefficient of performance, respectively, and compared to thermostatic expansion valve, it provides an increase of 35% of the cooling load and increase of 10% of the coefficient of performance, respectively.

Cite this article as: Chauhan A, Parekh A, Patel V. Experimental investigation of the impact of electronic expansion valve opening on vcr system performance: comparative analysis with capillary tube and thermostatic expansion valve. J Ther Eng 2025;11(5):1468–1482.

INTRODUCTION

Air conditioning and refrigeration systems are increasingly prevalent in daily life, reflecting the rise in living standards in contemporary society. In Europe, heating

and cooling account for over half of all energy consumption [1]. Cooling and air conditioning systems currently represent approximately 20% of global energy consumption, and this share is projected to rise with the growing

*Corresponding author.

*E-mail address: vkp@med.svnit.ac.in

This paper was recommended for publication in revised form by Editor-in-Chief Ahmet Selim Dalkılıç



demand for cooling [2]. Consequently, there is a pressing need for energy-efficient technologies. Enhancing the efficiency of Vapor Compression Refrigeration (VCR) systems is crucial, and the expansion process plays a vital role in this improvement. In vapor compression refrigeration (VCR) systems, expansion devices are critical for regulating the flow and pressure of the refrigerant. These devices transform high-pressure, high-temperature refrigerant into a low-pressure, low-temperature state without the consumption of work [3]. Capillary tubes are the simplest form of refrigerant flow control, characterized by their design caused by the long and narrow tubes. These tubes are cost-effective and require a low starting torque [4]. The thermostatic expansion valve (TXV) is a vital component in HVAC systems and commercial refrigeration applications, particularly in walk-in coolers and freezers. Its main function is to precisely regulate the refrigerant flow into the evaporator coil by adjusting the flow passage opening, thereby controlling the refrigerant mass flow rate within the system. The TXV operates based on a suction superheat set point, calibrated to keep the refrigerant gas temperature above its saturation point at the compressor inlet [5-6]. Thermostatic expansion valves (TXV) can encounter difficulties in adjusting to substantial and quick changes in refrigerant flow, especially when variable-speed compressors are used. Traditional TXV, which are typically set to maintain a constant refrigerant flow rate under stable operating conditions, may require assistance to effectively respond to these dynamic changes [7]. In TXV-controlled direct expansion evaporator systems, hunting can occur under certain conditions where there are sustained oscillations in refrigerant flow rate, evaporating pressure, and superheat temperature [8].

Electronic expansion valves (EXV) are considered an alternative to TXVs for enhancing the performance of refrigeration systems in terms of operating efficiency and evaporator capacity. An EXV functions similarly to a TXV but employs a different control mechanism. Instead of relying solely on temperature sensing with a thermostatic element, an EXV compares the superheat level against a specified set point. This comparison generates a signal that regulates the opening or closing of the valve. The valve position is adjusted by a stepper motor controlled by a microprocessor. This electronic control system enables more precise and responsive regulation of refrigerant flow based on real-time superheat measurements. By continuously adjusting the valve opening according to the set superheat target, EXV can optimize system performance and efficiency, particularly in applications with variable operating conditions and load demands. The use of electronic control technology in EXV enhances the adaptability and effectiveness of refrigeration systems, providing improved reliability and energy efficiency compared to traditional TXV [9]. Many researchers believe that EXV offer superior control over superheat compared to TXV. Typically, TXV are set to maintain a degree of superheat between 5 to 8

°C under normal operating conditions. In contrast, EXV enable more efficient utilization of evaporator capacity by reducing the degree of superheat to approximately 3°C [10]. Thermostatic expansion valves (TXV) regulate the degree of superheat, but the stability of superheat can fluctuate if the control algorithm does not precisely adjust the valve's opening position. TXV usually employ proportional (P) control alone, lacking integral (I) control which is found in electronic expansion valves (EXV) equipped with proportional-integral-derivative (PID) control. EXV using PID control maintain more stable and precise superheat levels by incorporating integral control to correct deviations from the set point. This capability enhances the overall efficiency and effectiveness of refrigeration systems compared to traditional TXV [11]. Experimental analysis of an automotive air conditioning system with a thermostatic expansion valve (TXV) using two refrigerants, R-134a and R-1234yf, revealed that R-1234yf exhibited lower refrigeration capacity and COP compared to R-134a refrigerant [12]. The experimental results from the chiller system equipped with an EXV show a 30% reduction in superheat value and 6% lower power consumption compared to systems using TXVs. The control of superheat temperature in this setup utilizes a stepper control circuit with a fuzzy logic algorithm [13]. An examination of the refrigeration system with a variable speed compressor and different EXV openings reveals the use of a PI controller to regulate the EXV opening. The system maintains a superheat temperature of 6°C [14]. Many researchers have investigated various superheat temperature controllers, with studies employing fuzzy controllers [13, 15-16], predictive functional control (PFC) [17-19], and PI and PID controllers [20-23]. Among these, PID controllers are noted for providing superior control of superheat temperature compared to other types of controllers. Additionally, researchers have analyzed and modeled the adiabatic flow in capillary tubes for refrigerants such as R-22 and R-407C.

Many researchers have studied various configurations of EXV openings. It has been found that using an EXV consumes 9% less energy compared to a capillary tube in refrigerators [24]. Furthermore, a variable speed compressor paired with different EXV openings results in a chiller system with 30% lower superheat temperature and 6% lower power consumption compared to systems operated with TXV [13]. In the experimental study of a dual evaporation temperature-based chiller, it was observed that the cooling and heating capacity typically varies proportionally with compressor frequency. However, when considering the combination of EEV1 and EEV2 openings, there is an inverse relationship noted in the experimental results [25]. Additionally, researchers have conducted analysis and modeling of adiabatic flow in capillary tubes using refrigerants like R-22 and R-407C. The results of these models were compared with experimental data, demonstrating that the mathematical model yields optimal outcomes for air conditioning systems [26]. The numerical modeling

of a wire and tube condenser used in domestic refrigerators concluded that this modeling approach can be effectively utilized to design condensers for VCR systems [27]. An experimental investigation was conducted to enhance the performance of a domestic refrigerator by varying the length of the capillary tube (from 3.96 m to 5.18 m) with different combinations. The study concluded that the least energy consumption occurred with a capillary tube length of 4.3 m when using R-134a refrigerant [28].

Based on the literature reviewed, there is a limited number of studies focusing on enhancing the performance of Vapor Compression Refrigeration (VCR) systems. No specific experimental research has been conducted to investigate the performance of VCR systems using various openings of electronic expansion valves (EXV) at specific compressor frequencies. Current research trends suggest a shift towards adopting EXV to improve energy efficiency in refrigeration systems. PID controllers are favoured for their effectiveness in maintaining stable superheat temperatures. This study aims to investigate the performance of the VCR system by varying EXV openings and comparing them with other expansion devices such as capillary tubes and TXV under identical input parameters. The objective is to assess and compare the efficiency and effectiveness of these different expansion devices in VCR systems.

EXPERIMENTAL INVESTIGATION

Experimental Setup Description

The present study aims to investigate an experimental Variable Compression Refrigeration System (VCRS) with different Electronic Expansion Valve (EXV) openings and compare the performance using a capillary tube and TXV expansion devices. The experimental schematic diagram consists of the compressor, an air-cooled condenser (fin and tube), expansion devices (capillary tube, TXV, and EXV), and a plate heat exchanger that acts as the evaporator. Water from a 60-liter tank was circulated to the evaporator using a water flow pump. Various sensors were used, including RTD PT-100 temperature sensors, pressure transducers, a rotameter for refrigerant flow measurement, a paddle-wheel-type water flow meter, and humidity sensors. Data acquisition and control systems were integrated into a human-machine interface (HMI), facilitating seamless operation. Within the system, solenoid valves were used to control the equipment. A Eureka variable area glass tube rotameter, which provides $\pm 2\%$ accuracy over the flow range, was used to measure the refrigerant flow rate. The schematic diagram is shown in Figure 1.

The frequency of the semi-hermetic reciprocating variable speed compressor is regulated by a PLC (programmable logic controller). Suction and discharge pressure

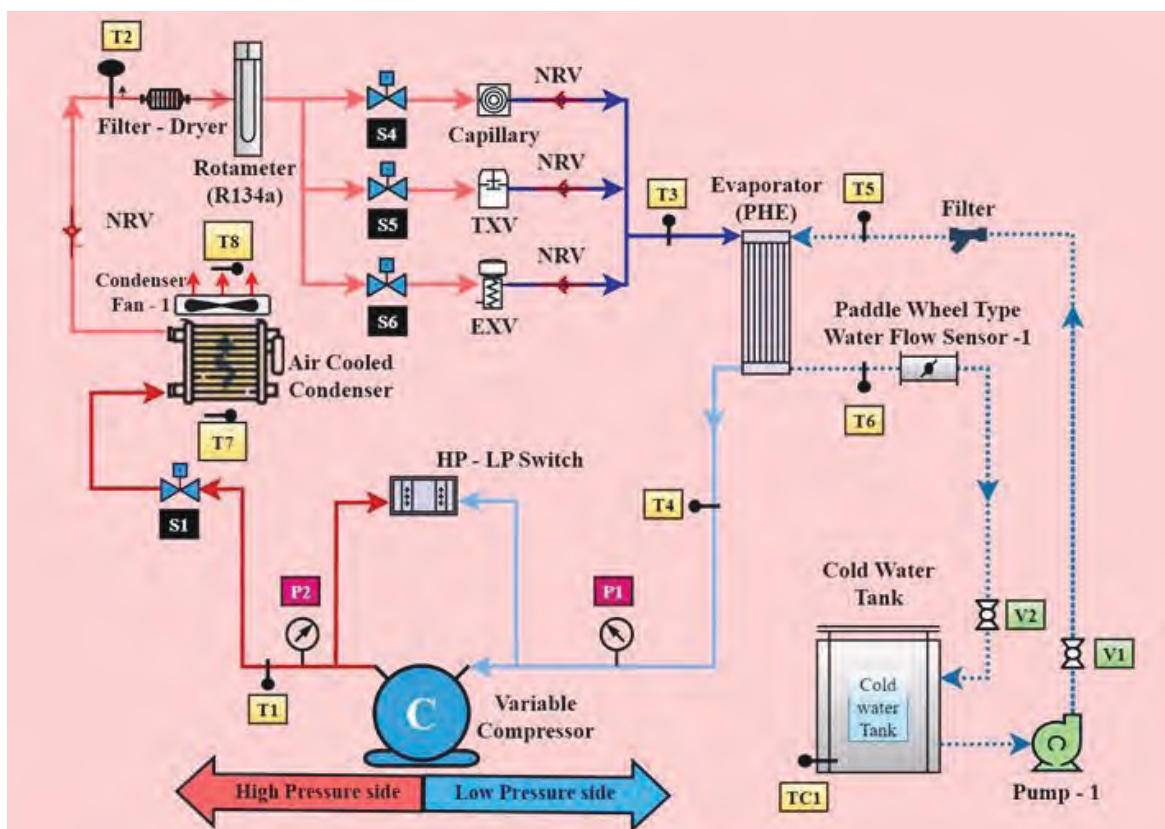


Figure 1. Schematic diagram of experimental setup.

transducers are installed to measure compressor pressure. Once compressed, the refrigerant R134a moves to an air-cooled fin and tube condenser. The refrigerant passes through a filter and then through a glass tube rotameter to measure the flow rate. Three types of expansion devices; thermostatic expansion valve (TXV), electronic expansion valve (EXV), and capillary tube are each controlled by their respective solenoid valves.

The water-cooled plate heat exchanger acts as an evaporator, with water circulated by a 60-liter stainless steel water tank using a PLC-controlled water pump. The flow rate of water is measured by a paddle wheel-type water flow meter, while the temperature is monitored using an RTD Pt 100 stainless steel sensor.

The RTD Pt 100 stainless steel sensors are installed to measure the inlet and outlet temperatures of the refrigerant at all four main components. Precisely, temperature sensors T1 measures the outlet and T4 measure the inlet temperature of the compressor respectively, T1 and T2 monitor those of the condenser, T2 and T3 measure the temperatures at the expansion device, and T3 and T4 measure the temperatures at the evaporator. The front view of the experimental setup, which includes the HMI screen, glass tube rotameter, and expansion devices with their respective solenoid valves, is depicted in Figure 2. The back view of the experimental setup, showcasing other components of the VCR system, is illustrated in Figure 3. The water-cooled and cooling tower based condenser is not used in present study.

The specifications of the different components of the VCR system are listed in Table 1. A semi-hermetic reciprocating compressor is used with refrigerant R134a, offering a capacity range of 0.75 to 1.25 tons of refrigeration (TR). The temperature of the refrigerant and water is measured using RTD Pt 100 stainless steel sensors, which provide

an accuracy of $\pm 0.1\%$ of full scale. Suction and discharge pressures are monitored using pressure transducers with an accuracy of $\pm 0.8\%$ of full scale. The measured data, including temperature, pressure, and water flow rate, is stored in the data acquisition system. The PLC controller regulates the Electronic Expansion Valve (EXV) opening.

Investigated Parameter

The performance of the VCR system is investigated with different expansion devices, including a capillary tube, thermostatic expansion valve (TXV), and various percentages of electronic expansion valve (EXV) openings (ranging from 30% to 70% in 5% increments). The following parameters are held constant for all experiments: compressor frequency set at 45 Hz, air-cooled fin and tube type condenser, water pump flow rate fixed at 50 Hz (23.1 LPM), initial water temperature in the evaporator tank maintained at 28°C, and the target cold water temperature set at 15°C. Measurements include the evaporative cooling tank's inlet and outlet water temperatures and different refrigerant temperatures at all four VCR components. Compressor power consumption is monitored using an energy meter. The cooling load, cooling COP, and system cooling COP are derived using mathematical equations.

Experimental Procedure

The experimental procedure commences by powering up the VCRES and configuring the desired parameters on the HMI screen, as specified in Table 2. It is crucial to allow a 30-minute stabilization period after making the selections on the HMI to achieve steady-state conditions within the system.

The initial water temperature in the evaporator tank is set at 28°C with a target temperature of 15°C. The compressor frequency is fixed at 45 Hz for all test runs. Data,



Figure 2. Front view of experimental setup.

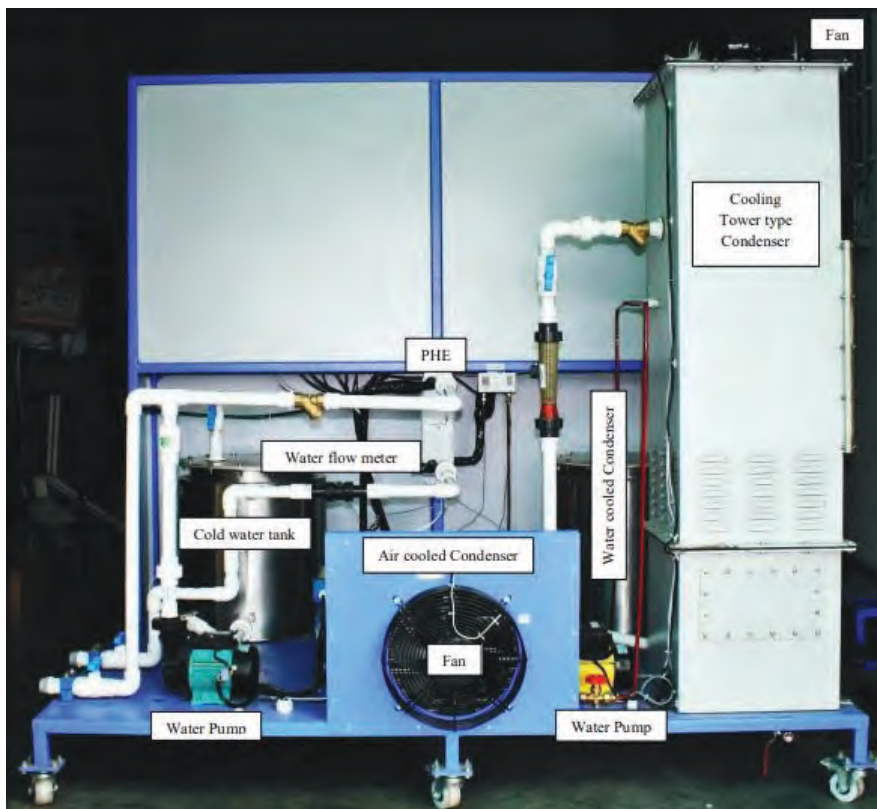


Figure 3. Back view of experimental setup.

Table 1. Specifications of different components

Name of components	Description
Compressor	Type: Semi-hermetic reciprocating Make: Frascold A1-6Y Refrigerant: R-134a Capacity: 0.75 – 1.25 TR, 400V, 30-60 Hz
Condenser	Type: Air cooled fin and tube
Expansion device	Type: Capillary tube, TXV and EXV
Evaporator	Type: Plate heat exchanger Suitable with most HFC, HFO and natural refrigerants.
Temperature sensor	Type: RTD Pt100, stainless steel sensor Accuracy: $\pm 0.1\%$ of full scale
Pressure transducers	Type: Sanhua YCQB Accuracy: $\pm 0.8\%$ full-scale
Water flow meter	Type: Peddle wheel Range: 20 – 50 lpm Accuracy: $\pm 4\%$
R-134a flow meter	Type: Eureka PG-5 (Glass tube rotameter) Range: 15-140 lpm
Data acquisition & control unit	VFD drive, Water flow controller, PLC control and HMI

Table 2. Input parameters of present experiment

Parameter	Details
Compressor frequency (Hz)	45 [25]
Condenser type	Air cooled fin and tube type with axial fan
Water pump flow rate (LPM)	23.1
Initial temperature of water in evaporator water tank ($^{\circ}\text{C}$)	28
Targeted cold water temperature in evaporator water tank ($^{\circ}\text{C}$)	15
EXV opening	30% to 70% in the increment of 5%.

including pressure, temperature, and water flow rate, are recorded and stored at 30-second intervals throughout the experiments until the water temperature reaches the target of 15°C. The recorded data is consistently saved in an Excel document during the experiment, facilitating subsequent performance measurements and analysis. The experiments are repeated for various EXV openings, ranging from 30% to 70% in 5% increments. Additionally, experiments are conducted using a capillary tube and TXV while keeping the same input parameters, and data are recorded for analysis purposes.

Mathematical Calculation

During the experiments conducted with the experimental setup, the temperature at the inlet and outlet of the evaporative cooling system is measured using the HMI system. The mass flow rates of water and refrigerant are measured using a paddle wheel-type flow meter and a rotameter, respectively. The compressor frequency is maintained at a constant 45 Hz throughout the experiment. Based on the data collected during these experiments, calculations are performed to determine the cooling load, cooling COP (Coefficient of Performance), and system cooling COP. The following is a brief description of these parameters:

Cooling load (Q_e)

The amount of heat absorbed per unit time by refrigerant in evaporator is termed as cooling load. It is calculated using Equation (1).

$$Q_e = \dot{m}_w * C_{pw} * (T_5 - T_6) \quad (1)$$

Cooling COP (COP)

The coefficient of performance (COP) describes the cooling capacity provided by compressor power. The cooling coefficient of performance is determined using Equation (2) as the ratio of the cooling load to work done by the compressor [29].

$$COP = \frac{Q_e}{W_c} \quad (2)$$

System cooling COP (COP_s)

A water pump facilitates water circulation in the chilled water tank, and an axial fan is used to flow air in a condenser. When calculating the system coefficient of performance (COP), it is essential to account for the power consumed by auxiliary devices. The total power (TP) of the system includes the power consumption of the water circulating pump, condenser axial fan, and compressor. Equation (3) provides the system cooling COP.

$$COP_s = \frac{Q_e}{TP} \quad (3)$$

Degree of superheating of refrigerant (ΔT_{sh})

The degree of superheating involves increasing the temperature of a vapor refrigerant above its saturation

temperature. Superheating is quantified as the temperature difference between the actual vapor and saturation temperatures corresponding to the refrigerant's pressure. It is determined by measuring the pressure and temperature, determining the saturation temperature, and calculating the difference. The superheating temperature is computed using Equation (4).

$$\Delta T_{sh} = T_{act} - T_{sat} \quad (4)$$

Where; T_{act} is an actual temperature of vapour and T_{sat} is a saturated temperature of refrigerant

RESULTS AND DISCUSSION

During the experiment, data collection was conducted at regular 60-second intervals. Various primary parameters, including pressures, temperatures, and flow rates, were measured and recorded. The experiments continued until the water temperature in the cold-water tank reached the desired temperature of 15 °C. The effects on various parameters are discussed in detail in the following paragraphs.

Compressor Discharge Temperature and Pressure Ratio

The Electronic Expansion Valve (EXV) regulates refrigerant flow into the evaporator coil based on the system's demands. The refrigerant flow rate increases when the EXV opens more comprehensively, allowing more refrigerant to enter the evaporator. Consequently, the pressure ratio also decreases. Essentially, as the EXV opens further, permitting more refrigerant to flow through the evaporator its pressure rises. The variation of pressure ratio at different valve opening is shown in Figure 4.

The increase in evaporator pressure subsequently increases the suction pressure at the compressor inlet. The compressor requires more work to achieve the desired discharge pressure when operating under increased suction pressure due to the broader EXV opening, because the overall work required is increased majorly due to higher mass flow rate. The initial refrigerant flow rate and compression workload may increase when the refrigeration system starts operating. This results in an initial rise in discharge temperature as the compressor compresses the incoming refrigerant to achieve the desired pressure and temperature conditions. The compressor may generate more heat during this phase, leading to a higher discharge temperature. As the system stabilizes and reaches a steady-state condition, the discharge temperature increases until it reaches a peak value. Then, the compressor adjusts to the required refrigeration load, stabilizing the refrigerant flow rate. As a result, the compressor operates more efficiently with reduced heat generation, gradually reducing and stabilizing the discharge temperature over time. Figure 5 illustrates the variation in compressor discharge temperature.

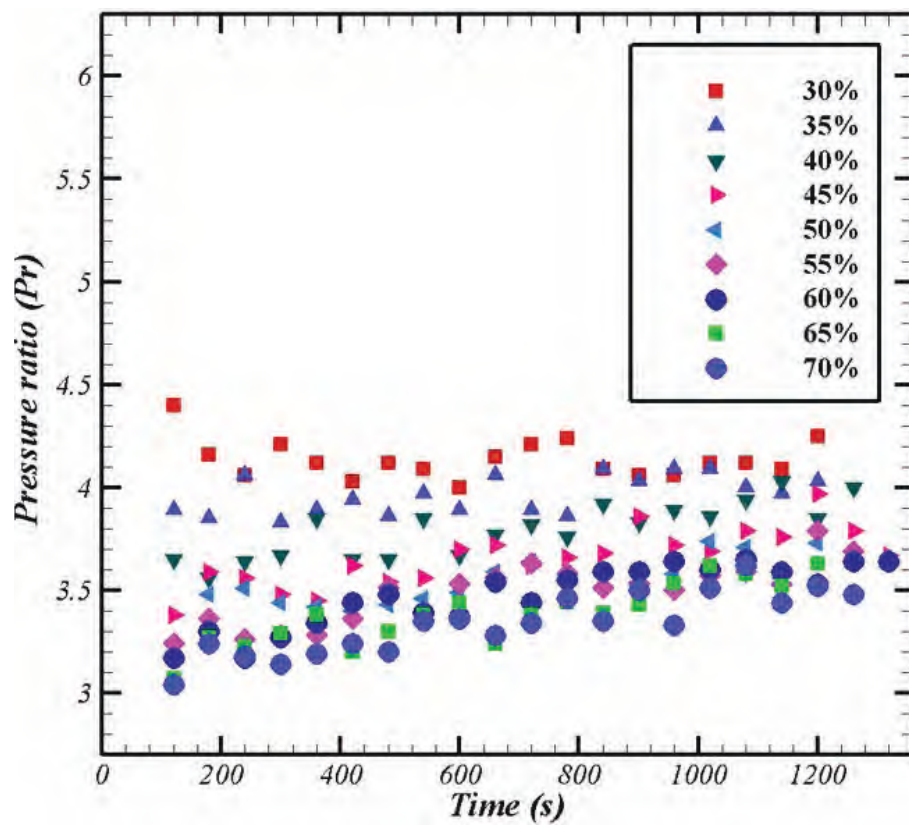


Figure 4. Effect on pressure ratio of EXV opening.

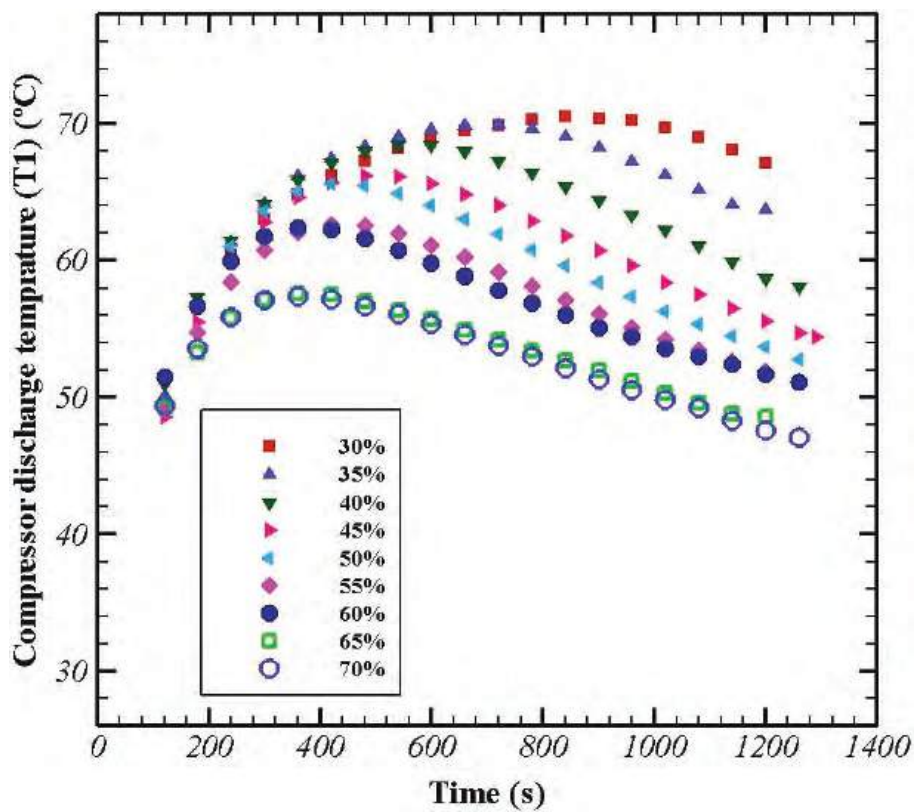


Figure 5. Effect on compressor discharge temperature of EXV openings.

Superheating of Refrigerant

With increase in EXV opening (increased refrigerant flow rate) corresponds to an increase in evaporator pressure, which reduces the refrigerant's saturation temperature within the evaporator coil. When the EXV opening is increased from 30% to 70%, there may be a transient period where system parameters, including superheat temperature, undergo rapid changes. The superheat temperature may initially decrease sharply as the EXV allows more refrigerant into the evaporator, leading to lower evaporator temperatures. As the system stabilizes with the wider EXV opening, the superheat temperature reaches a new equilibrium level. The reduction in superheat temperature becomes more gradual and stabilizes at a lower value compared to the initial conditions.

With lower EXV opening the inlet temperature of evaporator is low with lower evaporator pressure, this results in more sensible heat transfer to the vapor phase resulting in high superheating temperature. As the EXV opening is increased, the liquid fraction is increased along with increase in pressure in the flow, which gives lower superheating temperature due to latent of vaporisation. However, on further opening the EXV increase in pressure dominates thus reducing the saturation temperature and increasing the superheating temperature. Effect of EXV opening on superheating temperature is shown in Figure 6.

Condenser outlet temperature

The condenser outlet temperature in a vapor compression refrigeration system refers to the temperature of the refrigerant leaving the condenser. It plays a crucial role in the overall system performance and efficiency. The condenser's primary function is to remove heat from the refrigerant and reject it to the surrounding environment. The condenser outlet temperature reflects the effectiveness of this heat rejection process. A higher condenser outlet temperature indicates that the refrigerant is not being adequately cooled, and the heat transfer process is less efficient. The condenser outlet temperature is closely related to the refrigerant's saturation pressure. As the condenser outlet temperature increases, the saturation pressure of the refrigerant also tends to increase. The outlet temperature of the condenser observed over time typically exhibits dynamics influenced by system operation and external factors. Initially, the condenser outlet temperature may show fluctuations as the system stabilizes, adjusting to varying load conditions or EXV settings. Over time, as the system reaches steady-state operation, the condenser outlet temperature tends to stabilize within a specific range, reflecting consistent heat rejection performance. Factors such as ambient conditions, system load variations, and EXV control adjustments can influence the long-term trend of the condenser outlet temperature. The effect of EXV opening on condenser outlet temperature is shown in Figure 7.

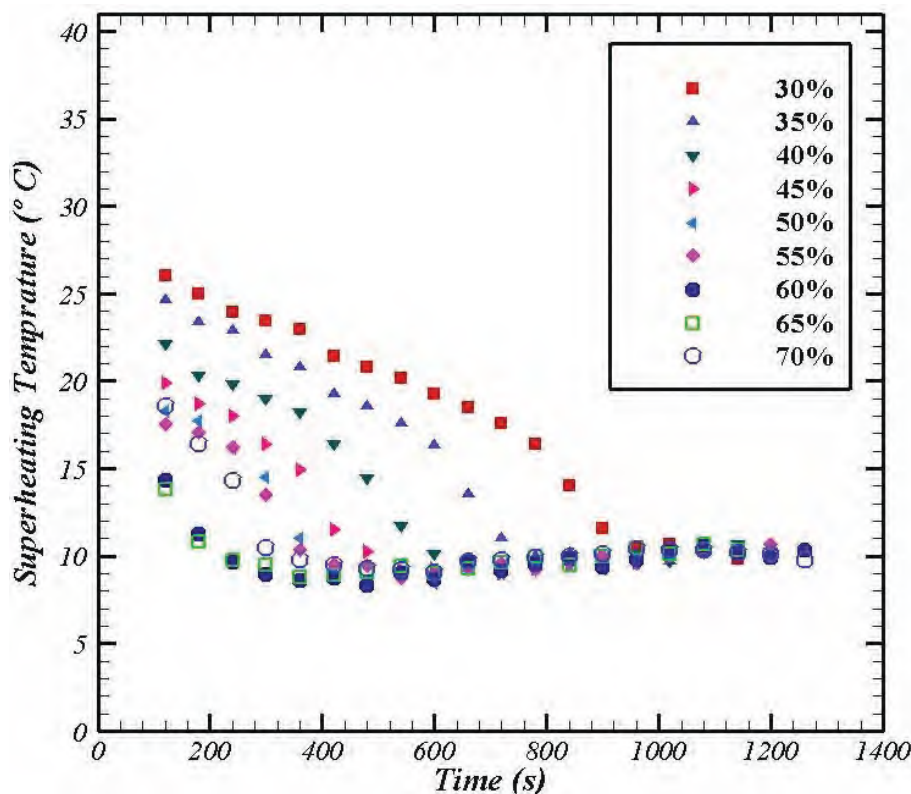


Figure 6. Effect on degree of superheating of EXV opening.

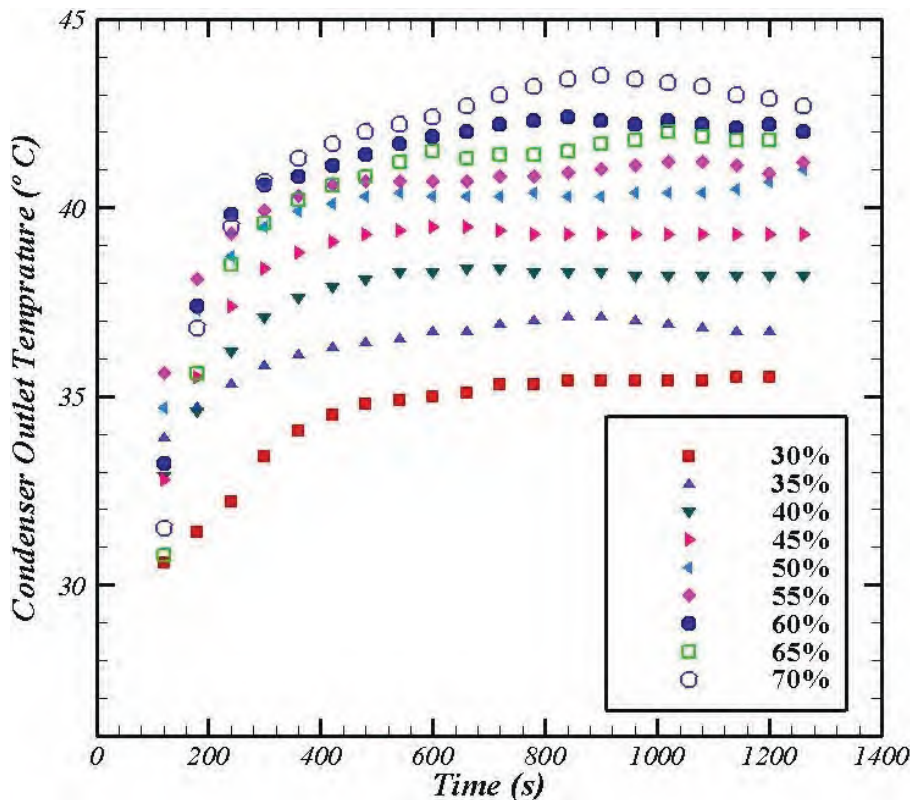


Figure 7. Effect on condenser outlet temperature of EXV opening.

Cooling load (Q_c)

The data reveals a clear trend in the relationship between EXV opening and cooling load in the refrigeration system. Initially, as the EXV opening increases from 30% to 60%, there is a substantial and proportional increase in cooling load. Specifically, the average cooling load at a 60% EXV opening is significantly higher compared to lower openings, with a 22.58% more compared to a 30% opening and a 20.15% higher compared to a 35% opening. This demonstrates that widening the EXV opening up to 60% allows for a more substantial refrigerant flow, leading to enhanced heat transfer and greater cooling capacity. Beyond an EXV opening of 60%, the cooling load continues to increase but at a diminishing rate, as indicated by the smaller percentage increases at 65% (3.4% increase) and 70% (6.1% increase) EXV openings. This diminishing return suggests that beyond a certain point (around 60% opening), additional increases in the EXV opening yield relatively smaller gains in cooling load. Therefore, the optimal EXV opening for maximizing cooling load efficiency is around 60%, highlighting the importance of careful EXV control to achieve optimal system performance and efficiency. Effect of EXV opening on cooling load is shown in Figure 8.

Compressor Work (W_c)

The electronic expansion valve's (EXV) opening position significantly impacts the compressor's performance

in a refrigeration system. The EXV opening directly affects the pressure and mass flow rate of the refrigerant entering the compressor, affecting the compressor's workload. As the EXV opening increases, the pressure at the compressor inlet increases, resulting in a lower compression ratio. This decrease in compression ratio reduces the work required by the compressor per unit mass, as it operates at a lower pressure and temperature. However, with further increases in the EXV opening, the mass flow rate of refrigerant entering the compressor also increases. Consequently, the compressor must work harder to accommodate the higher refrigerant flow rate. This higher flow rate requires the compressor to perform more work to compress the larger refrigerant volume, leading to an initial increase in compressor work. As the refrigerant flow rate continues to increase and reaches a point where the compressor's capacity is maximized, further increases in flow rate do not proportionally increase the compressor's workload. This results in an initial rise in power consumption. As mentioned above, on further increasing the EXV opening beyond a certain threshold, the decrease in pressure ratio dominates the mass flow rate of the refrigerant. As a result, the compressor's power consumption decreases, leading to the observed declining trend in power consumption with wider EXV openings.

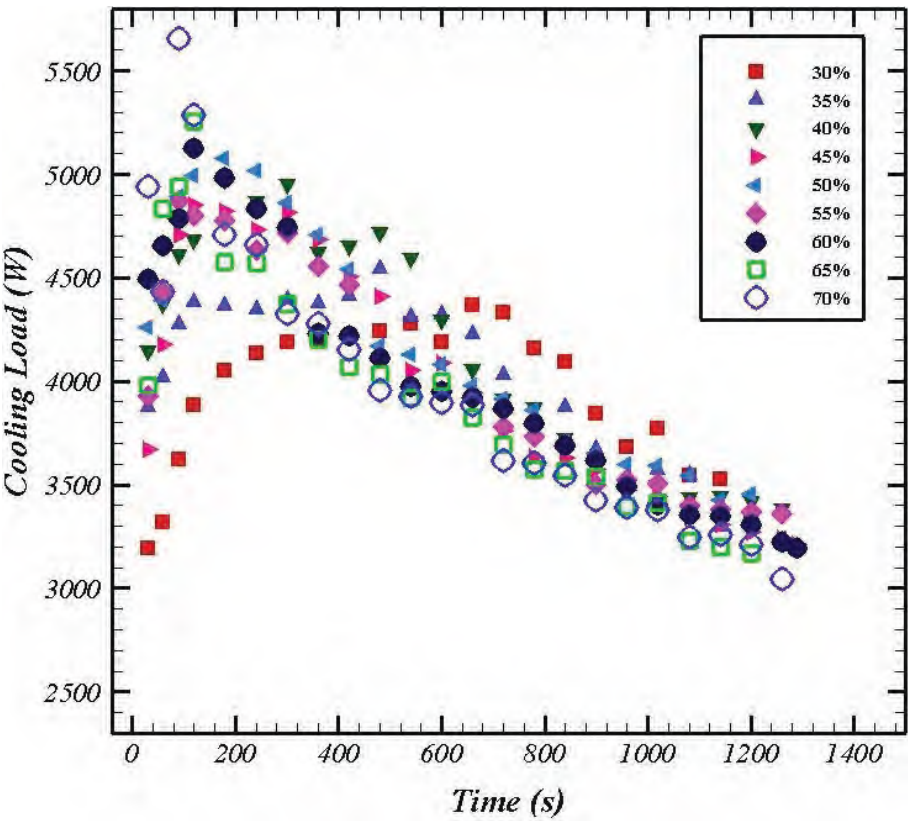


Figure 8. Cooling load for different EXV opening.

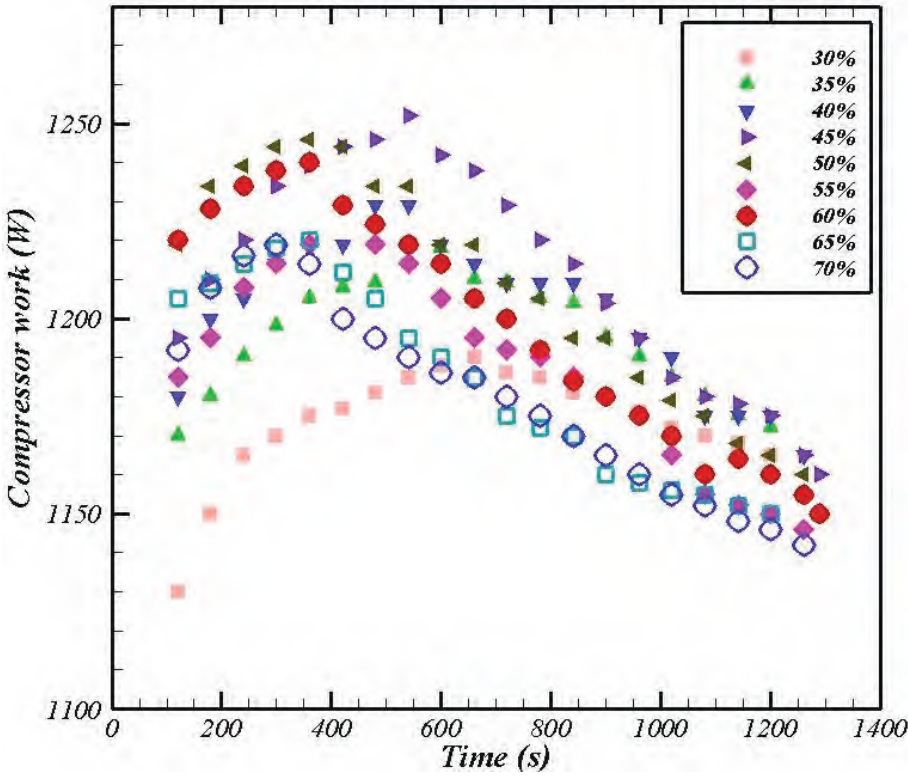


Figure 9. Compressor work for different EXV opening.

Cooling COP (COP_C)

The coefficient of performance (COP) in a refrigeration system, which quantifies heat transfer efficiency, is significantly influenced by the opening position of the electronic expansion valve (EXV). As the EXV opening increases, the cooling load also tends to rise due to the higher mass flow rate of refrigerant entering the evaporator, reaching its peak efficiency at around 60% EXV opening. However, further widening of the EXV opening results in increased compressor power consumption beyond this optimal point. This occurs because the compressor must handle a larger refrigerant volume at higher flow rates, leading to elevated workload and energy consumption. Consequently, the COP begins to decline for EXV openings of 65% and 70% compared to the optimal 60% opening. The observed trend highlights the trade-off between cooling performance and energy efficiency, emphasizing the importance of selecting an EXV opening that maximizes COP by effectively

balancing cooling load and compressor power consumption. The average COP for 60% of openings is 19% higher than for 30 and 35% of openings of EXV and 10% higher than for 40, 45 and 50% of EXV openings. The average COP for 65 and 70% opening is 1% and 5% lower than 60% of EXV opening. The effect of EXV opening on cooling COP of system is shown in Figure 10.

Validation

For validation, the experimental data compared with the theoretical calculation use of data provided by CoolPack software. The software works on engineering equation solver (EES). The input conditions are taken from the recorded data of the experiments. The percentage error is below 5% in the all cases. Table 3 shows the validation of experimental data with theoretical data.

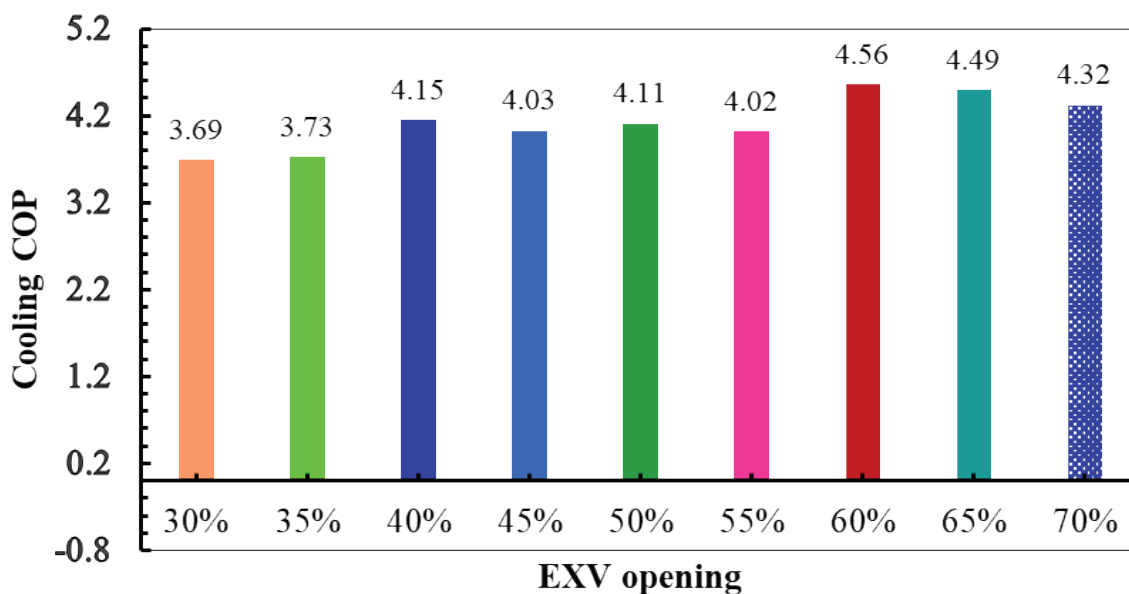


Figure 10. Cooling COP for different EXV opening.

Table 3. Validation of experimental data with theoretical data

EXV opening	Cooling COP (Experimental)	Cooling COP (Theoretical)	% Error
30	3.69	3.52	4.6
35	3.73	3.79	1.6
40	4.15	4.26	2.6
45	4.03	4.21	4.4
50	4.11	4.07	0.9
55	4.02	4.11	2.2
60	4.56	4.69	2.8
65	4.49	4.53	0.9
70	4.32	4.52	4.6

COMPARISON OF CAPILLARY, TXV AND EXV

All experimental work was conducted under similar input conditions for the capillary tube, thermostatic expansion valve (TXV), and electronic expansion valve (EXV), as detailed in Table 2. Figure 11 compares compressor power consumption, cooling load, and cooling coefficient of performance (COP) for these expansion devices. The results indicate that the cooling load achieved with the EXV is higher than that of the TXV and capillary tube. Specifically,

at a 60% EXV opening, the cooling load is 65% and 35% greater than that of the capillary tube and TXV, respectively. Additionally, the compressor requires more power due to higher superheat temperatures associated with the EXV compared to the TXV. Despite this, the cooling load achieved with the EXV remains significantly higher than that of the TXV and capillary tube, with relatively minimal changes in power consumption between the EXV and capillary tube. Consequently, the cooling COP of the EXV (at 60% opening) exceeds that of the capillary tube and TXV by

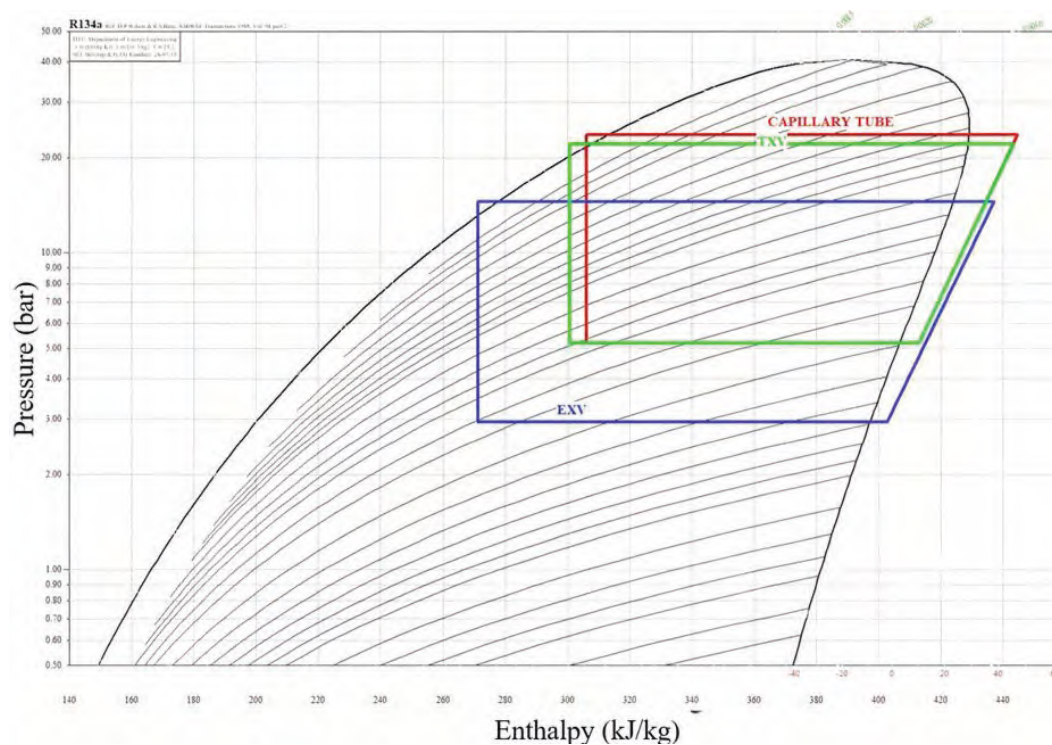


Figure 11. P-h diagram of expansion devices used.

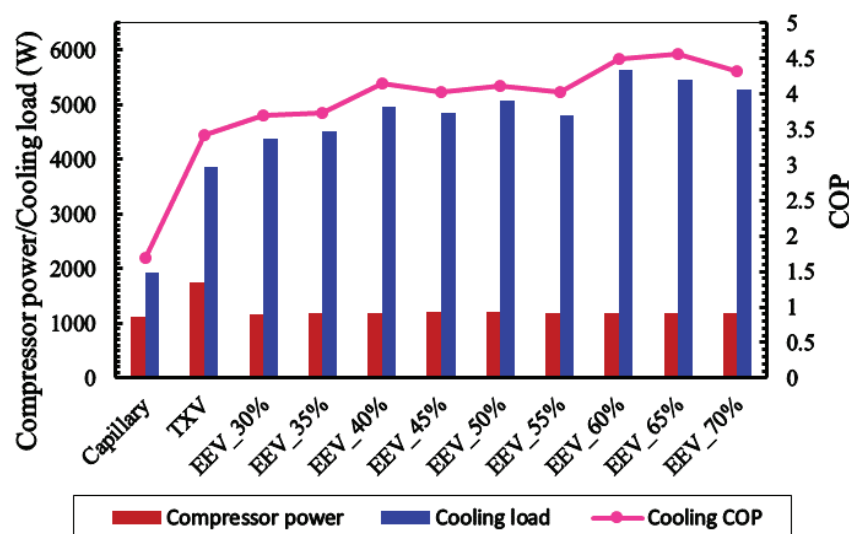


Figure 12. Comparison of performance for different expansion devices.

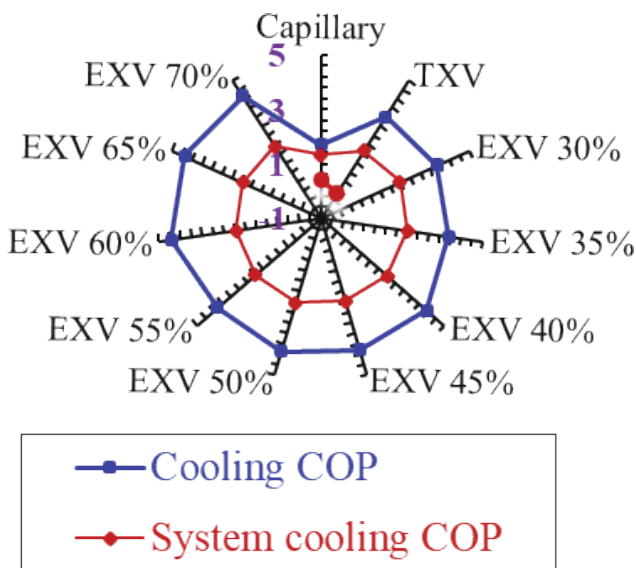


Figure 13. Comparison of performance for different expansion devices.

39% and 10%, respectively. The comparison of performance for different expansion devices is shown in Figure 12.

The pressure enthalpy diagram of VCR system with capillary tube, TXV and EXV expansion devices is shown in Figure 11.

The cooling COP and system cooling COP for capillary tube, TXV, and EXV are plotted on the radar graph and shown in Figure 13. The COP of the system is defined as the ratio of cooling load to total power consumed by the system. The total power (TP) of the system includes the power consumption of the water circulating pump, condenser axial fan, and compressor. The maximum cooling COP achieved is 4.56 for EXV opening of 60% which is 39% and 10% higher than capillary and TXV respectively.

CONCLUSION

The performance of the VCR system was investigated using various expansion devices, including a capillary tube, a thermostatic expansion valve (TXV), and an electronic expansion valve (EXV) at different opening percentages ranging from 30% to 70% in 5% increments. The experiments were conducted at a compressor frequency of 45 Hz. As the EXV opening increased from 30% to 70%, the discharge temperature and pressure ratio decreased, resulting in improved cooling efficiency and reduced compressor power consumption. This significant reduction in superheat temperature, attributed to the higher refrigerant flow rate associated with wider EXV openings, contributed to enhanced cooling rates and overall system performance. However, it is noteworthy that the condenser outlet temperature increased as the EXV opening widened, suggesting

that a broader EXV opening may lead to less effective heat rejection and lower system efficiency.

1. The maximum cooling load of 6560 W is achieved at a 60% EXV opening. At a 60% EXV opening, the cooling load was 22.58% higher compared to a 30% opening and 20.15% higher compared to a 35% opening.
2. The cooling load remained consistently higher, increasing by around 10 to 12%, in the EXV opening range of 40% to 55%. However, for EXV openings of 60% and higher, the cooling load exhibited diminishing returns, with a minor increase of 3.4% for a 65% EXV opening and a 6.1% increase for a 70% EXV opening.
3. The highest COP value is achieved at a 60% EXV opening, with a 19% increase compared to a 30% and 35% EXV opening and a 10% increase compared to a 40%, 45%, and 50% EXV opening.
4. The COP for 65% and 70% EXV openings was slightly lower, with a 1% and 5% decrease, respectively, compared to the 60% EXV opening.
5. The cooling COP of the EXV (at 60% opening) was 39% and 10% higher compared to the capillary tube and TXV.

NOMENCLATURES

C_{pw}	Specific heat of water (J/kg.K)
m_w	Mass flow rate of water (kg/s)
Q_e	Cooling load (W)
T_1	Compressor discharge temperature of refrigerant (°C)
T_2	Condenser outlet temperature of refrigerant (°C)
T_3	Evaporator inlet temperature of refrigerant (°C)
T_4	Compressor suction temperature of refrigerant (°C)
T_5	Inlet temperature of water of evaporator (°C)
T_6	Outlet temperature of water of evaporator (°C)
TC_1	Cold water tank controlling temperature (°C)
T_{10}	Air cooled condenser air outlet temperature (°C)
T_{sat}	Refrigerant saturated temperature (°C)
T_{act}	Refrigerant actual temperature (°C)
T_{sh}	Degree of superheating °C
TP	Total power (W)
WC	Compressor work (W)

Abbreviations

CCD	Central composite design
COP	Cooling coefficient of performance
COPS	System cooling coefficient of performance
DF	Degree of freedom
EXV	Electronic expansion valve
HFC	Hydro-fluorocarbons
HFO	Hydro-fluoroolefin
HMI	Human Machine Interface
HVAC	Heating ventilation and air conditioning
LPM	Liters per minutes
PHE	Plate heat exchanger
PF	Proportional-Fuzzy
PI	Proportional-Integral
PID	Proportional-Integral-Derivative

PLC	programmable logic controller
RSM	Response surface method
TXV	Thermostatic expansion valve
VCRS	Vapour compression refrigeration system
VFD	Variable frequency drive

Subscripts

act.	Actual
c	Compressor
s	System
sat.	Saturated
sc	Subcooling
sh	Superheating
w	Water

ACKNOWLEDGEMENT

I sincerely thank Sardar Vallabhbhai National Institute of Technology (SVNIT), Surat, for providing the necessary laboratory facilities and resources for this project. The support and infrastructure made available greatly contributed to the successful completion of my work. I would like to extend my heartfelt gratitude to Ishant Patil and Nachiket Shah for their invaluable assistance in conducting the experiments. Their dedication, hard work, and keen insight were instrumental in the successful completion of this research. Thank you for your unwavering support and contributions.

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article

REFERENCES

- [1] Urchueguía JF, Alakangas E, Berre I, Cabeza LF, Grammelis P, Haslinger W, van Helden W. Common implementation roadmap for renewable heating and cooling technologies: European Technology Platform on Renewable Heating and Cooling 2014.
- [2] Dupont JL, Domanski P, Lebrun P, Ziegler F. The role of refrigeration in the global economy. Informatory Note on Refrigeration Technologies 2019.
- [3] Cengel YA, Boles MA, Kanoğlu M. Thermodynamics: an engineering approach. New York (NY): McGraw-Hill; 2011. p. 445.
- [4] Zhang CL. Generalized correlation of refrigerant mass flow rate through adiabatic capillary tubes using artificial neural network. Int J Refrigeration. 2005;28:506-514. [\[Crossref\]](#)
- [5] Wichman A, Braun JE. Fault detection and diagnostics for commercial coolers and freezers. HVAC&R Res 2009;15:77-99. [\[Crossref\]](#)
- [6] ASHRAE A. Advanced Energy Design Guide for K-12 School Buildings: Achieving 30% energy savings toward a net zero energy building, W Stephen Comstock. [place unknown]: ASHRAE; 2008.
- [7] Chen W, Zhijiu C, Ruiqi Z, Yezheng W. Experimental investigation of a minimum stable superheat control system of an evaporator. Int J Refrigeration 2002;25:1137-1142. [\[Crossref\]](#)
- [8] Mithraratne P, Wijesundera NE. An experimental and numerical study of hunting in thermostatic-expansion-valve-controlled evaporators. Int J Refrigeration 2002;25:992-998. [\[Crossref\]](#)
- [9] Yu FW, Chan KT, Chu HY. Constraints of using thermostatic expansion valves to operate air-cooled chillers at lower condensing temperatures. Appl Thermal Eng 2006;26:2470-2478. [\[Crossref\]](#)
- [10] Roper, M. A. (2000). Energy efficient chiller control. BSRIA
- [11] Finn DP, Doyle CJ. Control and optimization issues associated with algorithm-controlled refrigerant throttling devices. Univ Coll of Dublin (IE); 2000.
- [12] Alkan A, Kolip A, Hosoz M. Energetic and exergetic performance comparison of an experimental automotive air conditioning system using refrigerants R1234yf and R134a. J Therm Eng 2021;7:1163-1173. [\[Crossref\]](#)
- [13] Ekren O. Energetic assessment of an electronic and a thermostatic expansion valve for a variable capacity compressor. J Adv Therm Sci Res 2019;6:51-57. [\[Crossref\]](#)
- [14] Li H, Jeong SK, You SS. Feedforward control of capacity and superheat for a variable speed refrigeration system. Appl Thermal Eng 2009;29:1067-1074. [\[Crossref\]](#)
- [15] Chia PK, Tso CP, Jolly PG, Wong YW, Jia X. Fuzzy control of superheat in container refrigeration using an electronic expansion valve. HVAC&R Res 1997;3:81-98. [\[Crossref\]](#)

- [16] Shuai Y, Wen M, Wen X, Li Z, P Shao-fei. Fuzzy control application of electronic expansion valve regulation in heat pump system. *Autom Instrum* 2016;6:13.
- [17] Changenet C, Charvet JN, Gehin D, Sicard F, Charmel B. Study on predictive functional control of an expansion valve for controlling the evaporator superheat. *Proc Inst Mech Eng Part I J Syst Control Eng* 2008;222:571-582. [\[Crossref\]](#)
- [18] Fallahsohi H, Changenet C, Placé S, Ligeret C, Lin-Shi X. Predictive functional control of an expansion valve for minimizing the superheat of an evaporator. *Int J Refrigeration* 2010;33:409-418. [\[Crossref\]](#)
- [19] Saleh B, Aly AA. Flow control methods in refrigeration systems: A. *Int J Control Autom Syst* 2015;4.
- [20] Maia AAT, Silva MDA, Koury RNN, Machado L, Eduardo AC. Control of an electronic expansion valve using an adaptive PID controller.
- [21] Tesfay M, Alsaleem F, Arunasalam P, Rao A. Adaptive-model predictive control of electronic expansion valves with adjustable setpoint for evaporator superheat minimization. *Build Environ* 2018;133:151-160. [\[Crossref\]](#)
- [22] Rasmussen H, Thybo C, Larsen LFS. Nonlinear superheat and evaporation temperature control of a refrigeration plant. *IFAC Proc Vol* 2006;39:251-254. [\[Crossref\]](#)
- [23] Al-Badri AR, Al-Hassani AH. A control method using adaptive setting of electronic expansion valve for water chiller systems equipped with variable speed compressors. *Int J Refrigeration* 2020;119:102-109. [\[Crossref\]](#)
- [24] Knabben FT, Ronzoni AF, Hermes CJ. Application of electronic expansion valves in domestic refrigerators. *Int J Refrigeration* 2020;119:227-237. [\[Crossref\]](#)
- [25] Li R, Zhu Y, Yang Y, Li K, Zhang R, Sun J, Sun Z. The effects of the opening of the electronic expansion valve in the high-stage cycle on the performance of a cascade heat pump water heater. *J Build Eng* 2021;42:103015. [\[Crossref\]](#)
- [26] Imran AA. Adiabatic and separated flow of R-22 and R-407C in capillary tube. *Eng Tech J* 2009;27. [\[Crossref\]](#)
- [27] Imran AA, Jafal HM. Numerical modeling of wire and tube condenser used in domestic refrigerators. *J Eng Sustain Dev* 2009;13:1-17.
- [28] Gugulothu S. Enhancement of household refrigerator energy efficiency by studying the effect of refrigerant charge and capillary tube length. *J Therm Eng* 2021;7:1121-1129. [\[Crossref\]](#)
- [29] Selloum A, Triki Z, Chiba Y. Thermodynamic analysis of a solar-driven vapor compression refrigeration system using R1234ze for cooling applications in Ghardaïa region (Southern Algeria). *J Therm Eng* 2021;10:130-141. [\[Crossref\]](#)



Research Article

Thermal performance of drying mango slices using a baffled-type hybrid solar dryer with exhaust hot air recirculation

Debashree Debadatta BEHERA¹, Shiv Sankar DAS², Ramesh CHANDRA MOHANTY^{1,*},
Raj Kumar SATANKAR³

¹Department of Mechanical Engineering, Centurion University of Technology and Management, Odisha 752050, India

²School of Management, Centurion University of Technology and Management, Odisha 752050, India

³Department of Mechanical Engineering, Poornima College of Engineering, Jaipur, Rajasthan 302022, India

ARTICLE INFO

Article history

Received: 01 June 2024

Revised: 18 September 2024

Accepted: 21 September 2024

Keywords:

Solar Drying, Forced Convection, Development of Hybrid Solar Dryer, Experimental Investigation, Performance Analysis, Uncertainty Analysis

ABSTRACT

In this research, a hybrid solar dryer has been designed for day and night operation through forced convection for drying mango slices and preparation of bread toast. As studied from the literature, enough research is not done on the use of recirculation of exhaust hot air and baffle in a solar dryer. The thermal and economic performance of the dryer has been assessed by calculating collector efficiency, drying efficiency, drying rate, payback period and cost-benefit ratio. It is found that the maximum collector efficiency is 74.1% and 66.96% with and without the use of exhaust hot air recirculation. Performance parameters such as drying rate (0.95 kg/hr), drying efficiency (32.89%), payback period (1.4394 years), and cost-benefit ratio (2.08) have been evaluated. An effective drying effect is produced by using an electric coil for night operation of dryer. An increase in the solar radiation increases the temperature of air inside the collector, thereby increasing the collector efficiency, drying efficiency and drying rate. The coefficient of determination for outlet collector temperature during consecutive three days is estimated above 95%, thus signifying a higher order of fit. Performing an uncertainty analysis of measurement, the uncertainty error is calculated below 10% exhibiting a reliable experimental result. Solar drying is considered as a sustainable food preservation method without negatively impacting the environment and finds wide applications in the fruit and food processing industries, domestic purposes etc.

Cite this article as: Behera DD, Das SS, CHANDRA MOHANTY R, Satankar RK. Thermal performance of drying mango slices using a baffled-type hybrid solar dryer with exhaust hot air recirculation. J Ther Eng 2025;11(5):1483–1496.

INTRODUCTION

Since ancient times, firewood, fossil fuels, open solar drying, and carbon-releasing coals have been used to dry

and preserve food grains. These techniques are unhealthy, expensive, and unreliable. Solar drying technology is a sustainable way to preserve food by harnessing solar energy. A

*Corresponding author.

*E-mail address: rcmohanty@cutm.ac.in

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkılıç



good alternative for operating in all weather conditions and at night is a hybrid solar dryer.

Şirin et al. [1] used an indirect photovoltaic dryer to test its performance with and without aluminium oxide nano-particles. Using nano-particles in thermal energy storage units reduced the drying period by 15 to 22%. The energy and exergy efficiency of collector was increased by 7% and 9%, respectively. A study on drying of cassava slices using a hybrid solar dryer that included six different types of evacuated tube collectors and a centrifugal pump driven by solar PV panel was carried out by Veeramanipriya and Sundari [2]. It required 8 hours to reduce the moisture content from 91.5 to 10.67%, whereas open sun drying took 13 hours to reduce the moisture content to 22.5%. The temperature within the drying chamber is between 30 and 40°C higher than that of the surrounding air. Singh and Gaur [3] studied on a hybrid greenhouse solar dryer using a heat exchanger as the drying bed. They found that the maximum drying period of food products was reduced by 61.9%, and the maximum payback time was determined as 2.87 years. Singh and Gaur [4] investigated drying tomato, ginger, and bottle guards using a hybrid solar dryer with and without an evacuated tube collector. For the crops mentioned above, the drying efficiency with an evacuated collector ranged from 14.22 to 27.99%, whereas the drying efficiency without one ranged from 9.63 to 24.88%.

In an experiment on a hybrid solar dryer, Nwakuba et al. [5] used an electric heater to dry the red pepper. Their observations showed that energy and collector efficiency varied from 13.2 to 35.6% and 23.7 to 72.4%, respectively by using a 1500W electric heating wire. The use of a hybrid solar dryer to dry vermicelli was investigated by Suherman et al. [6]. According to the study, the dryer efficiency, energy ratio for utilization, and ultimate energy efficiency increased to 17.02%, 0.32, and 83.6%, respectively. In order to improve heat transfer, Srithanyakorn et al. [7] conducted research on a mixed-type forced convection solar dryer employing a stainless steel wire mesh collector. The drying efficiency and drying chamber temperature were increased by 23.07% and 9.57°C, respectively. Deef et al. [8] developed a novel hybrid solar dryer to dry fish waste. For three samples of fish waste, the moisture content reduced from 75.2% to 24.8% throughout 10, 7, and 5 hours of drying. Maximum drying rates were recorded as 1.1, 1.22, and 1.41 kg/h at 45, 50, and 55°C. Salve and Fulambarka [9] have developed a solar collector for drying green chillies using an aluminium can and selective coating material. The maximum moisture removal rate and collector efficiency at the maximum mass flow rate have been calculated to be 88% and 50.27%, respectively. Srimanickam and Sunil [10] investigated a photovoltaic thermal collector with a solar dryer for drying 4 kg of coriander seeds. The moisture content has been reduced by weight from 68 to 9% within six days. For the purpose of drying mango leather, Shrivastava et al. [11] evaluated the efficiency of a hybrid greenhouse solar dryer integrated with an evacuated tube collector.

Compared to open sun drying (OSD), the drying time was reduced by 7 hours. The average convective heat transfer coefficient in OSD was 0.07 W/m²°C, but in the solar dryer, it was 0.11 W/m²°C and 0.08 W/m²°C, respectively. In order to dry vegetables, Behera et al. [12] conducted an experimental investigation on a hybrid solar dryer with and without phase change material. The maximum collector efficiency, drying efficiency, and drying rate were increased to 64.23%, 33.16%, and 1.351 kg/hr, respectively, through the use of the turbulence method. Heydari et al. [13] performed a study on the exergy and energy efficiency of a hybrid air heater integrated to a solar dryer. By adding auxiliary thermal element to solar air heater, temperature, thermal efficiency and second law efficiency was increased.

In previous studies, researchers have developed a hybrid type of solar dryer with a heat exchanger using an evacuated tube collector, flat plate collector, and phase change material. Their results did not show an appreciable rise in the drying rate and collector efficiency. Further, there has been no significant work on solar collectors using a baffle, recirculation of exhaust hot air, or electric heating coils during night operation. The authors are motivated to do further experimental study in order to improve the thermal performance of a hybrid solar dryer by these research gaps or limitations in the literature. The novelty of this study is the recirculation of hot exhaust air and addition of baffle for generation of turbulence effect. Because of these provisions, the drying parameters can be improved. As hot air generated inside the collector strikes the baffles, more fluid friction inside the collector randomly increases due to the turbulence effect and increasing the collector's outlet temperature. As a result, drying efficiency and drying rate are increased. Hot exhaust air is circulated by means of a pipe that connects the inlet of collector and the outlet of drying chamber. With an electric coil, the drying chamber temperature is raised to achieve the required drying effect for use at night. In the current research, drying of mango and preparation of bread toast have been experimentally studied. Various performance parameters including payback period, and the cost-benefit ratio have been evaluated. Finally, the current experimental results have been compared with the previous literature. Solar drying is a sustainable method that produces products of appropriate quality with minimal negative environmental impact. Solar dryers can be utilized for various applications in the fruit and food processing industries, agro-based industries, domestic purposes etc.

MATERIALS AND METHODS

Working Principle of Hybrid Solar Dryer

The hybrid solar dryer consists of major components such as a blower, drying chamber, solar photovoltaic system, flat plate collector, and electric heating coil. The heated air produced by the solar collector is circulated to

the drying chamber by blowers during the drying process. It is based on forced convection mode. Two forms of energy are required for drying, i.e. solar energy and electrical energy. The dryer needs solar energy during the day and electricity for night operation. Finally, the moisture found in food products is allowed to escape into the atmosphere. The flow diagram of the hybrid solar dryer operating in the day time is shown in Fig. 1.

Fabrication Process

The dimensions of each component of the solar dryer are first calculated before fabrication. The specifications of the main components are presented in Table 1. Using these dimensions of each component, a digital model of the front and rear sides of the dryer and its schematic diagram is developed using CATIA software, as displayed in Fig. 2.

The experimental set-up is constructed after the completion of the 3D model design. The material required for constructing a solar dryer has been procured from a local store. The components are manufactured, assembled, and installed in the University Mechanical Workshop. A 2.5 mm aluminium sheet has been used to fabricate the dryer. Because of the excellent heat conductivity and resistance to corrosion, aluminium sheet is selected for fabrication. Initially, aluminium sheets are cut as per the given size through a pipe cutter and sheet bending operation. Mild steel angle frames are used to join all the sheets, and finally, the sheets are riveted using a riveter. The drying chamber

constructed as a rectangular stainless steel box and is capable of carrying 15 kg of fresh food products for drying. The drying chamber is structured with hinges for opening and closing the door. Three L-shaped mild steel frames are riveted inside the drying chamber to each side of the vertical walls to support three horizontal trays. Within the drying chamber, three vertical stainless steel wire mesh trays are placed uniformly at 110 mm spacing to provide a uniform air heating.

An inclined mounting structure is fabricated to support and position the solar panel using a mild steel angle frame. To maximize the solar energy incident on the collector, the solar panel and collector are oriented 20° to the south as per the latitude of the location (20.2961°N , 85.8245°E) [14]. An angle protractor is used to measure the angle of the tilted structure. The collector glass plate experiences an increase in the incident solar radiation from 9:00 AM to 1:00 PM, followed by a decrease from 1:00 PM and 4:00 PM. The collector receives a maximum solar radiation of 670 W/m^2 . The charge controller and battery are placed in the basement of the drying chamber so that it will not be exposed directly to solar radiation.

A rectangular box-type solar flat plate collector is fabricated using a 2.5 mm thickness aluminium sheet. The sides and bottom of the collector are insulated with glass wool of 75 mm thickness. Two glass plates of varying thickness are fixed to the top of the solar collector. A gap of 38 mm is provided between two glass plates to transfer enough solar energy to the absorbing plate. The upper

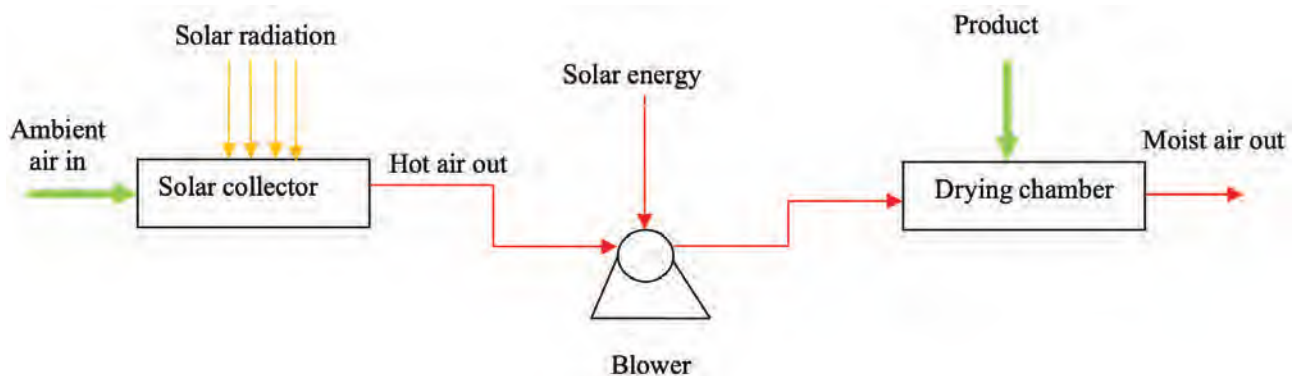


Figure 1. Flow diagram of hybrid solar dryer during day time.

Table 1. Design specifications of main components of solar dryer

Sl. No	Parameter	Specifications
1	Solar flat plate collector	1 m × 1 m × 0.22 m
2	Drying chamber	1 m × 0.5 m × 0.5 m
3	Two glass plates	0.98 m × 0.98 m of varying thicknesses of 5 mm and 4 mm
4	Glass wool	75 mm thickness

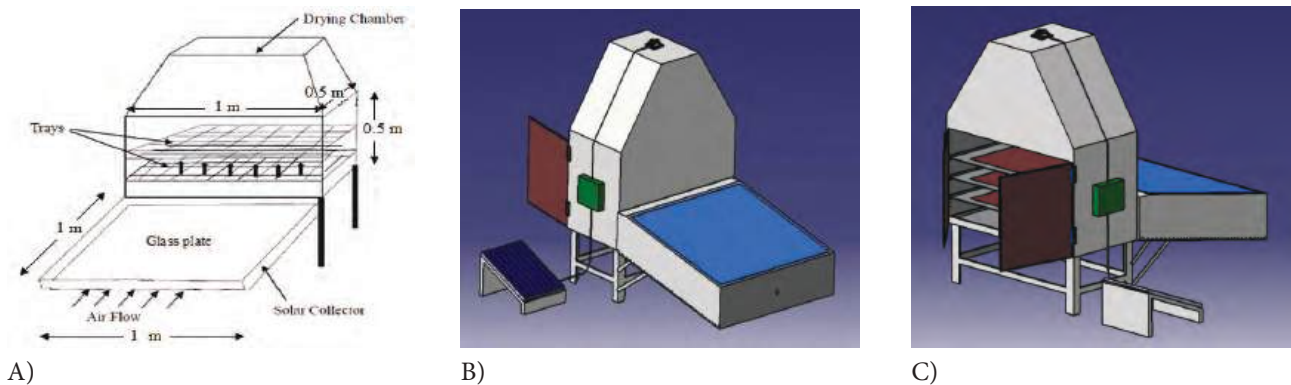


Figure 2. A. Schematic diagram; B. CATIA model of front side; C. Rear side.

glass plate is movable while the lower glass plate is fixed for ease of maintenance. Using two glass plates makes it possible to enhance solar radiation transmissivity and reduce the amount of heat dissipated from the absorbing plate as a result of reradiation. The absorbing plate is constructed from a 1.5 mm thick aluminium sheet and is made of a V-shaped wavy structure by bending operation. It is a wavy structure having a larger surface area, which enhances the rate of heat transfer. A baffle provided over the absorbing plate enhances turbulence and hot air flow into the drying chamber. To maximize its ability to absorb solar energy, the dryer's outside and interior surfaces have been painted in black. By adequately sealing with glass wool, the heat losses from the side walls of the collector and drying chamber have been minimized. The escaped hot exhaust air from the drying chamber can be further recycled into the collector inlet through a pipe connection. The front and rear sides of

the experimental setup for the hybrid solar dryer have been illustrated in Fig. 3.

Proper sizing and capacity of solar panel, battery, and charge controller is required to run four DC blowers. (i) The solar panel sizing is calculated for four blowers @3W, considering five sunshine hours per day that produce a 60Wh load. For climate conditions, the total load will be double, i.e., 120Wh. Considering five sunshine hours, a $120\text{Wh}/5=24\text{W}$ solar panel is needed. As per availability in the market, 20W capacity solar panel has been used. (ii) 12V battery is generally used for 3-100W solar panel. The Ampere hour of the battery is calculated as $120\text{Wh}/12=10\text{Ah}$. Therefore, a lead acid battery with a capacity of 12V and 10Ah was used. (iii) The sizing of the charge controller is calculated as $24\text{W}/12\text{V}=2\text{A}$. However, as per the market's availability, a 12V and 10A capacity MPPT charge controller has been taken. Therefore, a solar panel (20W), a battery (12V, 10Ah), and a charge controller

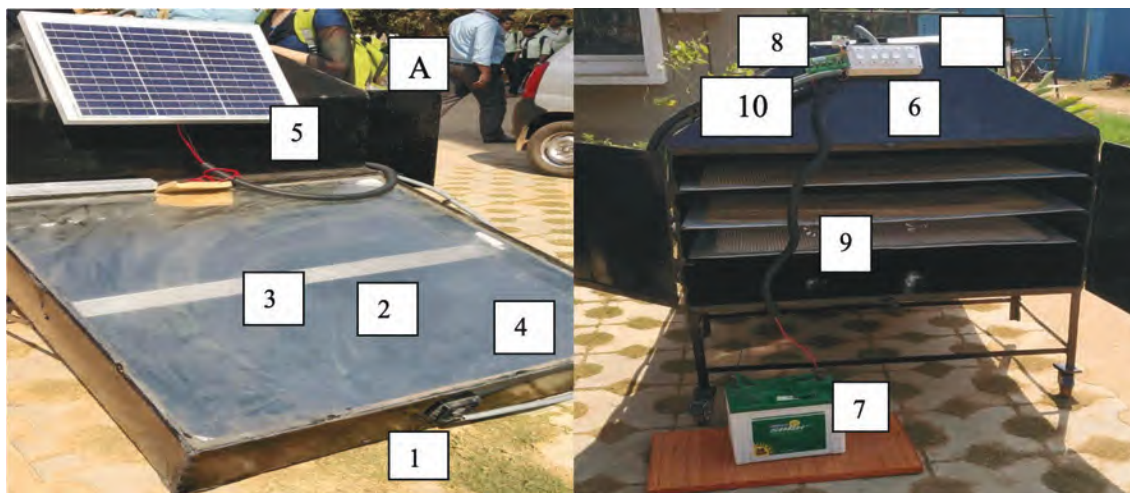


Figure 3. Experimental set-up: A; B. Front side; C. Rear side of hybrid solar dryer.

1, 8; DC blower at the inlet and outlet of the collector, 2. Absorbing plate, 3. Baffle, 4. Connecting pipe at the inlet of collector, 5. Solar panel, 6. Drying chamber, 7. Battery, 9. Tray, 10. Charge controller

Table 2. Specifications of different electrical components

Sl.No	Parameter	Specifications
1.	Polycrystalline type solar photovoltaic panel	Maximum power output =20W, Maximum voltage =17V, Maximum current =1.18A, Open circuit voltage =21.50V, Short circuit current =1.29A
2.	MPPT charge controller	12V, 10A
3.	Inverter	100W, IGBT model
4.	DC blower	3W, 12V, 0.6A
5.	Lead acid type battery	12V, 10Ah
6.	Electric heating coil	50W, 5A

(12V, 10A) have been used in the present work to power four DC blowers.

The first DC blower is placed at the collector inlet, bringing enough ambient air into the solar collector. To ensure adequate circulation of hot air within the drying chamber, the two additional DC blowers are connected electrically at the collector outlet. Moist air is released into the atmosphere by the fourth one, which is located at the drying chamber's outlet. Table 2 presents the specifications for various electrical components.

Measuring Instruments

Various measuring devices such as infrared thermometer, hygrometer, lux meter, anemometer, and digital balance weighing machine are used to measure various parameters like temperature, relative humidity, intensity of solar radiation, air velocity, and weight of food product respectively. Different measuring devices and their specifications are presented in Table 3.

Uncertainty Analysis of Measurement

The uncertainty surrounding any measurement originates from variations in the data collected and can be caused by either random or systematic effects. Determination of measurement uncertainty is essential because it allows assessments of experimental results to be made while considering their effect. The standard deviation of the

measurement is to be determined for finding the uncertainty error. The reading of the outlet collector, drying chamber temperature, and relative humidity of the drying chamber are recorded five times at regular intervals at some particular operating conditions. The final result is obtained by taking an average of all the recorded data. The standard deviation of the experimental result is obtained by using Equation (1) [15];

$$\sigma = \sqrt{\frac{\sum_{i=1}^n (Z_i - \bar{Z})^2}{n-1}} \quad (1)$$

Where, σ = standard deviation, Z_i = recorded data in i^{th} time, $\bar{Z}$ = mean value of Z_p , n = number of recorded data.

The uncertainty error of measurement is determined by using Equation (2);

$$U_{T_{CO}} = \frac{\sigma_{T_{CO}}}{T_{CO_{MEAN}}} \times 100 \quad (2)$$

Where, $T_{CO_{MEAN}}$ is the average outlet collector temperature, $\sigma_{T_{CO}}$ is standard deviation of outlet collector temperature and $U_{T_{CO}}$ is the uncertainty error of outlet temperature in percentage.

Table 3. Specifications of measuring instruments

Measuring instruments	Measured parameters	Model of measuring instruments	Accuracy	Ranges
Infrared thermometer	Temperature	MT-5	$\pm 2^\circ\text{C}/\pm 2\%$	40~550°C
Hygrometer	Humidity	KUSAM-MECO, KM919	Humidity: $\pm 2.5\%$ for 30-80%, $\pm 3.0\%$ for 20-29.9% and 80.1-89.9, $\pm 5\%$ for 1-19.9 and 90.1-99.9	Temperature: 20.0 ~ +60.0°C Humidity: 1.0 ~ 99.9% RH.
Lux meter	Solar radiation	KUSAM-MECO-LUX 99	$\pm 5\%$	2000 to 50000 Lux
Anemometer	Velocity	KUSAM-MECO, KM-909	$\pm 3\%$ FS	Temperature: 20.0° ~ +60.0°C Wind velocity: 0.0-30.0 m/s

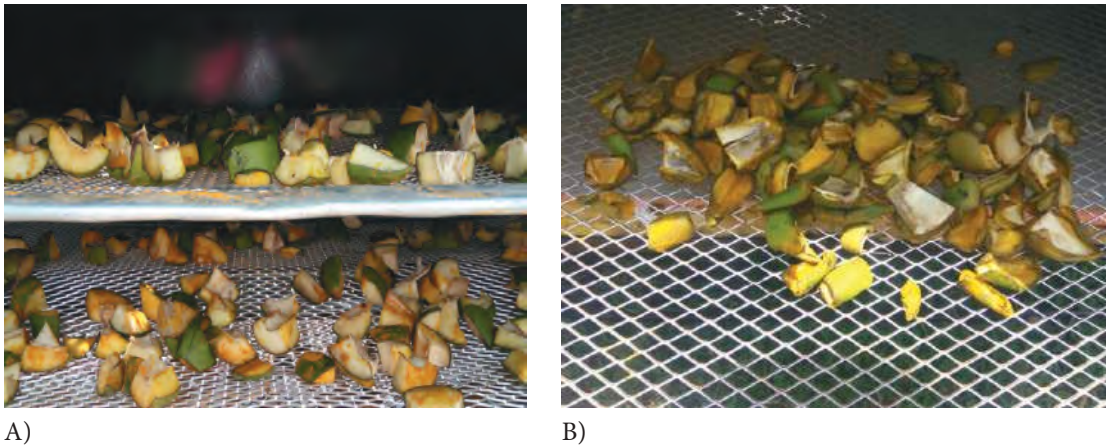


Figure 4. Drying of mangoes; A. Before drying; B. After drying.

Experimental Procedure for Drying Mangoes

The portable solar dryer is initially placed in an area that would provide for maximum exposure to solar radiation. The experiments were carried out during June to test five kilograms of green mangoes for pickle preparation. The mango slices are placed inside the drying chamber after it has been preheated for thirty minutes. Three days are required to complete the tests considering 6 hours of drying time per day. After the experiment is completed, the food products initial and final weights are determined for each day. To compare the results, an additional tray of mango slices is set outside the dryer for open sun drying. The drying of mangoes before and after drying is shown in Fig. 4. When comparing the hybrid type to other traditional solar drying techniques, the quality parameters, including taste, aroma, colour, and nutritional content, are higher. By reducing the moisture content in fruits and vegetables, solar drying improves their nutrient content by preventing microbe growth and deterioration. Dried food products therefore can be preserved fresher for a longer period in a hybrid solar dryer.

Performance Analysis of Solar Dryer

The dryer performance depends on the parameters such as inside temperatures of the dryer, relative humidity, solar radiation, and hot air velocity. The parameters such as pay-back period, cost-benefit ratio, moisture content on a dry basis, drying efficiency, drying rate, and collector efficiency are to be determined to assess the performance of the solar dryer by using Equations (3-5) [16] and Equation (6) [17].

The efficiency of the collector,

$$\eta_c = \dot{m}_a C_p \left(\frac{T_{co} - T_{ci}}{A_c \times I_T} \right) \quad (3)$$

Where, $\dot{m}_a$ is mass flow rate of air, C_p is specific heat at constant pressure, T_{co} and T_{ci} are collector outlet and inlet

temperature. Maximum value of $\dot{m}_a$ is calculated at 1:00 PM as 0.0272 kg/s and C_p is taken as 1007 J/kgK.

The drying efficiency of a system determines its overall efficiency, which is measured based on the instantaneous flux incidence (I_T), collector surface area (A_c), moisture removal (m_{wt}), and latent heat of evaporation (L_v). It is the energy required to use heated air to remove moisture from food products.

$$\text{Drying efficiency} = \eta_{\text{drying}} = \frac{m_{wt} \times L_v}{A_c \times I_T} \quad (4)$$

As evident from the Equation 4, the drying efficiency depends on many parameters such as quantity of moisture removal, latent heat of evaporation, collector surface area and instantaneous flux incidence. It can be improved with the higher moisture removal and latent heat of evaporation of moisture. It is calculated by taking collector surface area and latent heat of evaporation as 1 m² and 2260 kJ/kg respectively.

The ratio of the time interval and the difference between two successive moisture contents is known as the food product's drying rate (DR).

$$DR = \frac{MC_{db(t+dt)} - MC_{db(t)}}{dt} \quad (5)$$

Where, $MC_{db(t+dt)}$ is moisture content of food products at time $t + dt$ and $MC_{db(t)}$ moisture content of food products at time period t

The initial and final moisture contents are used to determine the moisture content ratio on a dry basis (m_{dr}) using Equation 6. The initial and final moisture content are taken as 74% and 10% respectively for inlet and outlet of collector [18].

$$m_{dr} = \frac{MC_{dbi} - MC_{dbf}}{MC_{dbf}} \quad (6)$$

Performance indices such as payback period and cost-benefit ratio have been discussed below using Equations (7-8) [17]. A solar drying system's payback period (PBP) is a significant economic factor. It is the amount of time required to recover the savings from the project's total investment during that time. It is calculated as explained below.

$$\text{Payback period} = \frac{C}{M_{\text{dry product}} P_{\text{dry product}} - M_{\text{fresh product}} P_{\text{fresh product}} - M_{\text{dry product}} X} \quad (7)$$

Where, C = Dryer's total capital cost, which is the sum of its labour and material costs.

X = Drying cost = Total capital cost per year / total amount of dried products produced annually

$M_{\text{dry product}}$ = Production of dry products annually

$M_{\text{fresh product}}$ = Amount of fresh products

As seen from Equation (7), the payback period depends on the drying cost, total capital cost and annual amount of fresh and dry product. Its value can be improved with the increase in the drying cost and amount of annual fresh and dry products. Less payback period is desired because the total investment cost on a solar dryer will be recovered during less period of time.

The cost-benefit ratio of a developed system is the ratio of its total annual benefit to its total capital cost. And is given as;

$$\text{Cost-benefit ratio} = \frac{\text{Annual benefit received}}{\text{Entire capital cost of the developed system}} \quad (8)$$

Using the above equations, the different performance indicators have been calculated and presented in Table 4.

Any system must be economically viable for the successful adoption by the market. The hybrid solar dryers are evaluated economically using various economic indicators.

Table 4. Performance parameters

Performance parameters	Values
Maximum collector efficiency with and without exhaust hot air recirculation	74.1% and 66.96%
Drying efficiency	32.89%
Drying rate	0.95 kg/hr
Moisture content on a dry basis	2.55
Payback period	1.439 years
Cost-benefit ratio	2.008

Table 5. Analysis of cost of solar dryer

Items	Cost/year (INR)	Cost/year (USD)
Material cost includes Aluminium Sheet, Glass plate, glass wool, Hinges, square bar, flat, angle, screw, drill bit, electro rod, riveting	15,736	196.977
Cost of electrical components includes solar panel, battery, charge controller, inverter, switches, wire, DC fan, electric coil	15,000	187.765
Total capital cost = material cost+cost of electrical components + labour cost	30,736	384.743
Annual cost = Total capital cost + maintenance cost (assuming 1% capital cost)	31,043	388.67
Drying cost (X)	75	0.94
Annual price of fresh product, $P_{\text{fresh product}}$	55000	688.55
Annual price of dry product, $P_{\text{dry product}}$	64,500	807.47

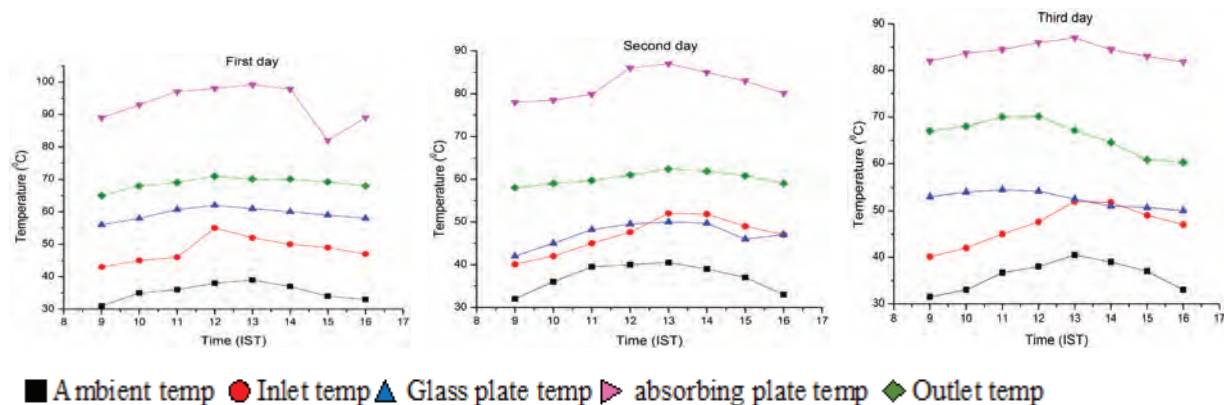


Figure 5. Temperature variations in solar collector.

The various cost parameters are presented in Table 5. The hybrid solar dryer's total capital cost is estimated at 30,736 INR. The annual cost is calculated by adding the total capital cost and maintenance cost. As an average estimation, the present hybrid solar dryer can dry 1000 kg of fresh product annually, producing 430 kg of dry products and the drying cost per kg of dry product is calculated as 75 INR as presented in Table 5. If a conventional solar dryer is used for the same amount of fresh product, 350 kg of dry product will be produced with a drying cost of 87.81 INR per kg. It is observed that the drying cost per kg is less in case of hybrid solar dryer because of more annual production of dry product.

The PBP value of 1.439 years signifies that the total investment in the hybrid solar dryer can be recovered within 1.5 years. The cost-benefit ratio 2.008 reveals a sound investment against the total benefit received over the period. Therefore, the developed system is economically viable and feasible for drying various vegetables.

RESULTS AND DISCUSSION

Temperature Variation in Solar Collector and Drying Chamber

To investigate the drying rate and drying efficiency of the hybrid solar dryer, the temperature variation in the drying chamber and solar collector needs to be measured on regular intervals. The tests are carried out throughout three days, with a 7-hour drying period every day from 9:00 AM to 4:00 PM.

The temperatures of the glass plate, absorbing plate, collector inlet, and collector outlet are constantly monitored in order to determine the temperature variation in the solar collector. It is found that when the temperature within the collector increases and the humidity decreases so that the ability of the air to carry moisture increases. The temperatures for the atmosphere, collector inlet, glass plate, absorbing plate, and collector outlet are 39, 52, 61, 99.2, and 71°C respectively on the first day at 1:00 PM. As shown in Fig. 5, similar tests conducted on the second and third days indicate that the highest temperatures are at 40.5, 52, 50, 87, 62.4°C and 40.5, 52, 52.5, 87, and 67.1°C, respectively. It has been noted that the highest temperature occurs between 12:00 and 1:00 PM at the various above-mentioned locations. The standard deviation and average temperature of the outlet collector at 1.00 PM are determined by repeated measurement five times. The standard deviation and uncertainty error of collector outlet temperature are calculated as 1.2 and 1.63%, respectively, by using Equations (1-2). The experimental measurement data is considered reliable as the uncertainty error is less than 10%.

The residual plot of the outlet collector temperature during three consecutive days with respect to time as shown in Fig. 6. A nonlinear regression analysis using MATLAB R2024a has been performed on the experimental

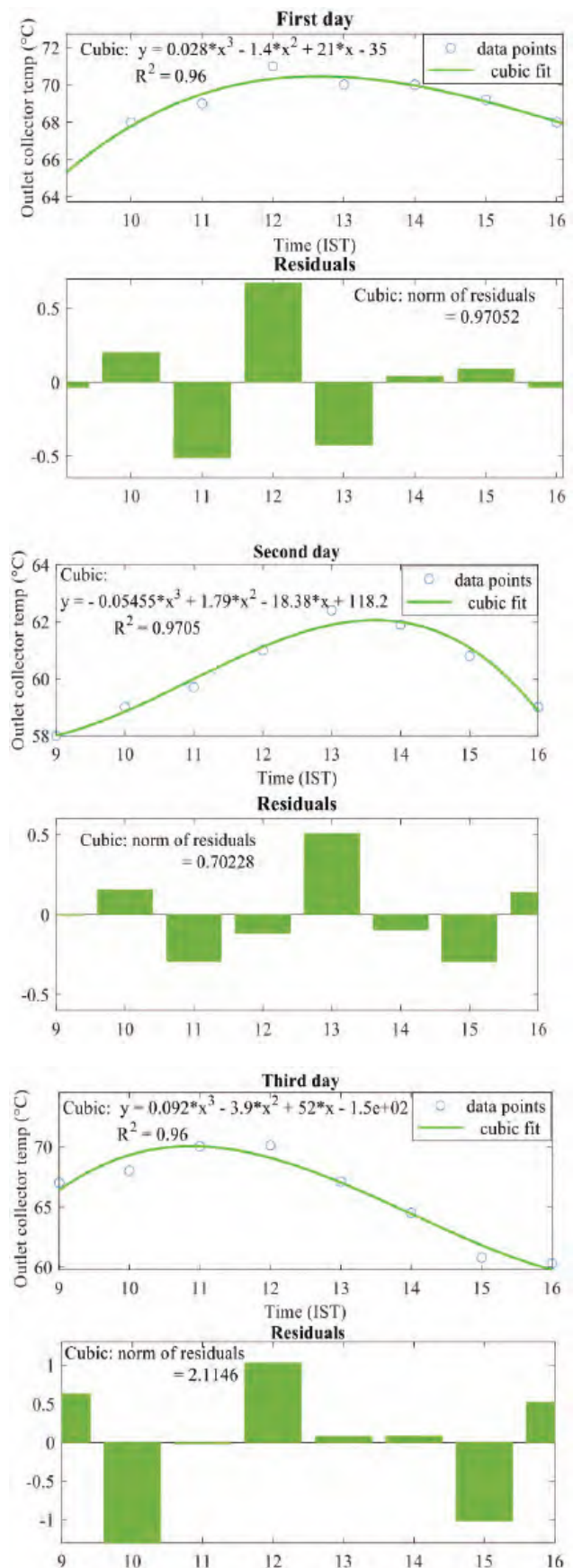


Figure 6. Error analysis plot for outlet collector temperature.

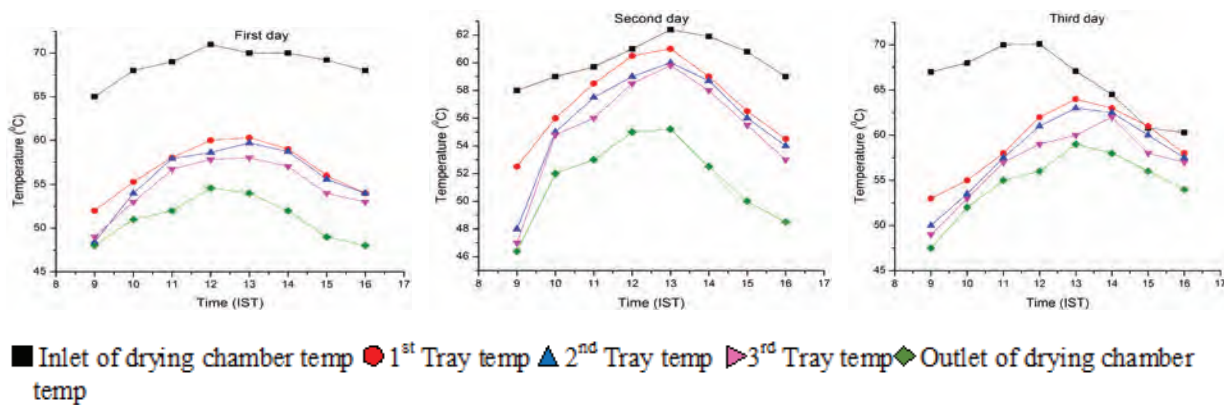


Figure 7. Temperature variation of the drying chamber.

data to provide a best-fit curve represented by a third-degree polynomial. The coefficient of determination (R^2) is used to analyze the relationship between variations of outlet collector temperature with respect to time period. The significance of the relationship between these two factors can be determined by this coefficient. The results show that there is a perfect correlation between the parameters, with R^2 values of 96%, 97%, and 96% for the first-, second-, and third-day observations, respectively. The norm of residuals for the first, second, and third day are found as 0.97, 0.7, and 2.11, respectively, representing a good relation between the parameters.

A similar procedure is used to regulate the drying chamber's temperature. The drying chamber consists of three trays, with the first tray positioned at the bottom. It is found that the temperature at the bottom tray is high and slightly decreases in trays placed above. The maximum temperatures of the drying chamber at 1:00 PM on the first day were recorded at 70°C at the inlet, 60.3, 59.7, 58, and 54°C at the first, second, and third trays, and the drying chamber outlet respectively. The maximum temperatures on the second and third days are 62.4, 61, 60, 59.8, 55.2, and 67.1, 64, 63, 60, and 59°C, respectively. From this data for three days, it is observed that the inlet drying chamber temperature shows maximum because it is almost the same at the outlet of the solar collector. This temperature gradually decreases when the hot air flows from the bottom towards the top of the tray. The heated and low-humidity air circulating in the first tray causes moisture content of the mango to evaporate and rise to the top of the drying chamber. The temperature further decreases in the second and third trays. This is because the first tray is positioned at the bottom of the drying chamber and collector outlet; the vaporized moisture will move upward from the first tray toward the outlet of the drying chamber, thus reducing the temperature in the second and third trays placed above. This results in a difference in temperature between the first tray and the drying chamber's outlet. A temperature gradient is established due to hot air flow from the bottom to the top of the drying

chamber. The drying rate and food quality are uniformly maintained due to the even distribution of hot air as the perforated trays are evenly spaced inside the drying chambers. The uncertainty error of drying chamber is obtained as 3.7% using Equation (2). The temperature variations of different trays have been shown in Fig.7.

Variation of relative humidity inside drying chamber

The dryer's relative humidity (RH) of air is not uniform; it varies with time during the drying period. As shown in Fig. 8, the RH of the exhaust moist air is compared for three days with that of the heated and ambient air within the drying chamber. At 1:00 PM on the first day, the heated and unsaturated air inside the drying chamber has a RH of 38%. In contrast, atmospheric air has a RH of 51%. As the temperature of the drying chamber increases, the RH gradually reduces and the amount of moisture removed from the food products increases. Drying chamber's hot air is found to have a lower RH than the outside air, while the moist air that exits the chamber has a higher RH. At 1:00 PM on the second day, the ambient air humidity is 50% and the RH of the first, second, and third trays, as well as the drying chamber outlet, are measured to be 38.4, 38.5, 38.7, and 38.9%, respectively. Similarly, the corresponding values on the third day are 35.7, 35.6, 35.7, and 36%.

The RH of drying chamber is found to be significantly different from that of the surrounding air. This is because the drying chamber's hot and dry air has a higher capacity to carry moisture than the surrounding air. The RH of air in the inner part of the drying chamber gradually decreases from 10:00 AM to 1:00 PM due to increased temperature. Because of the significant moisture removal of food products, there is a relatively slight difference in the humidity of the drying chamber and exhaust air during the same duration. On the second and third days, the RH of the exhaust air and the drying chamber remain unchanged and it remains constant, and the RH is gradually reduced in the inner part of the drying chamber as compared to the first day. Thus, the moisture removal rate from food products is

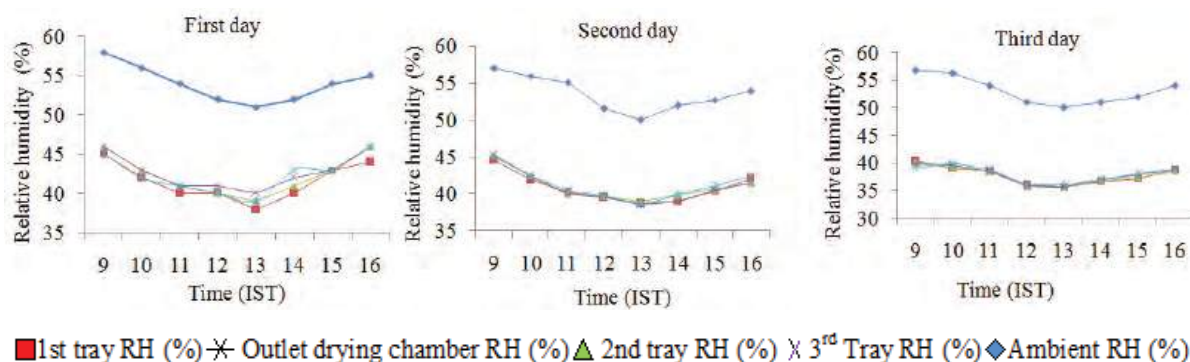


Figure 8. Variation of relative humidity of drying chamber and ambient air.

gradually reduced from the first to the third day. The uncertainty error of relative humidity is obtained as 3.147% using Equation (2).

Temperature of drying chamber during the night time

An additional heating device, such as an electric heating coil, is used to dry because solar radiation is unavailable in cloudy and night conditions. The electric heater is powered by a battery that solar panel charges during the day time and provides 4 to 5 hours of backup power at night. An electric coil has been placed in the drying chamber so that it can run at night time. The electric heating coil and inverter are powered by a battery. The experimental setup for the night use is shown in Fig. 9.

The drying chamber is preheated for thirty minutes before the drying process starts. For making bread toast, the temperature is constantly monitored between 6.30 and 7.30 PM. Table 6 shows that the drying chamber's maximum temperature is 66.7°C. The drying rate is evaluated as 0.85 kg/hr, resulting in a satisfactory drying effect and overall efficiency of the dryer.

Variation of collector outlet temperature, collector efficiency and solar radiation with time

The outlet temperature of the collector, solar radiation impacting the collector surface, and collector efficiency depend on the day time. Tests have been conducted on the variation in solar radiation for three days, considering the effect of collector efficiency and collector outlet temperature. At the maximum incident solar radiation of 670 W/m², the maximum collector outlet temperature, and its corresponding efficiency were determined as 72°C and 65.74%, respectively, using Equation (3) on the first day. The efficiency and collector outlet temperature gradually increases from 10:00 AM to 1:00 PM and then reduced after 1:00 PM as presented in Fig. 10. Similarly, at the maximum incident solar radiation of 657 W/m² and 643 W/m² on the second and third days, the corresponding maximum collector outlet temperature and efficiency were found as 72.5°C, 66.96%, and 72°C, 63% respectively as shown in Figs. 11 and 12, respectively.

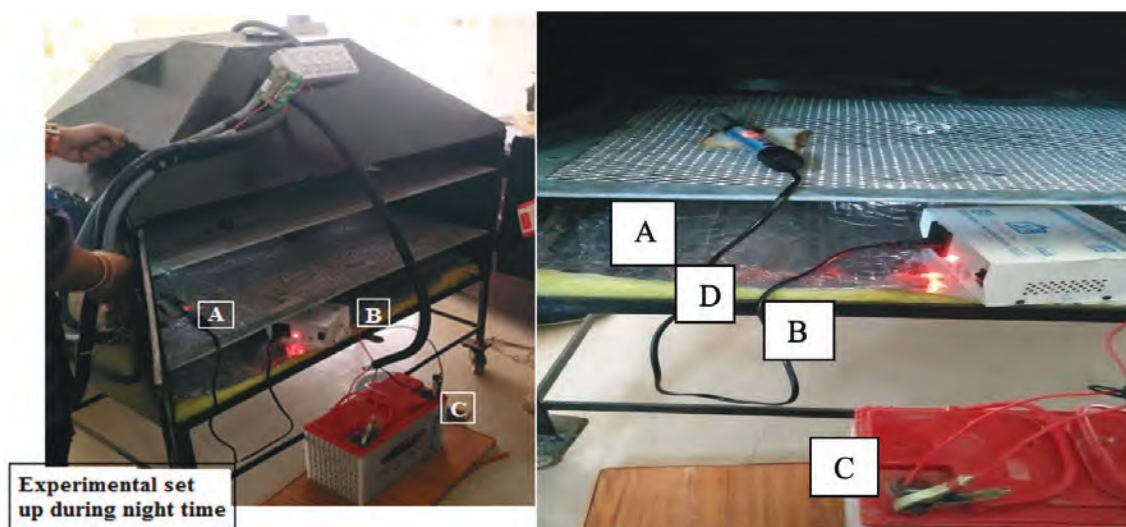


Figure 9. Experimental setup during night time; A. Electric heating coil; B. Inverter; C. Battery; D. Glass wool

Table 6. Temperature indication during night time

S. N.	Time (PM)	Drying chamber temp (°C)
1	6:30	43.5
2	6:40	50.8
3	6:50	54.9
4	7:00	58.6
5	7:10	62.4
6	7:20	64.7
7	7:30	66.7

Comparison Study of Hybrid Solar Dryer with Traditional Methods Of Drying

Initial and final mass of the sample, drying time, and percentage of mass reduction are compared for various solar drying methods. As shown in Fig. 13, five readings

of initial and final masses recorded every hour have been assessed for both hybrid and open solar drying systems. The maximum drying rate of a hybrid solar dryer is 0.85 kg/hr, which is higher than the rates of natural convection and open solar drying (0.6 kg/hr and 0.48 kg/hr, respectively). Drying of food products using open and natural convection solar drying techniques will slowly become dehydrated and have higher moisture content because they take longer to dry. Moreover, a hybrid solar dryer preserves dried products for a longer period of time.

Comparative Analysis of Experimental Results with Similar Study

Parameters including drying chamber temperature, collector efficiency, and drying efficiency have all been compared with the experimental findings from previous studies conducted under similar testing conditions as presented in Table 7. The drying efficiency found in the present study is quite good by the use of hot exhaust air recirculation.

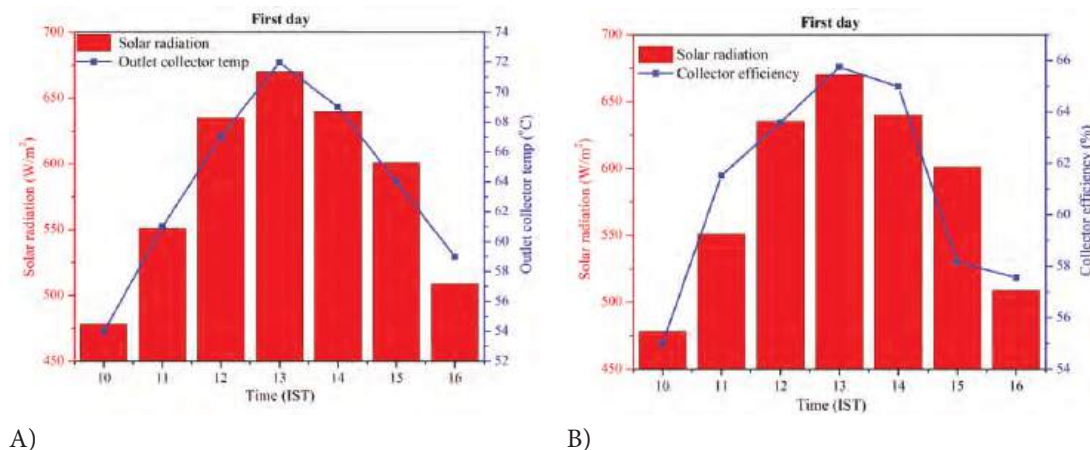


Figure 10. A. Variation of solar radiation and outlet collector temperature with time. B. Variation of solar radiation and collector efficiency with time.

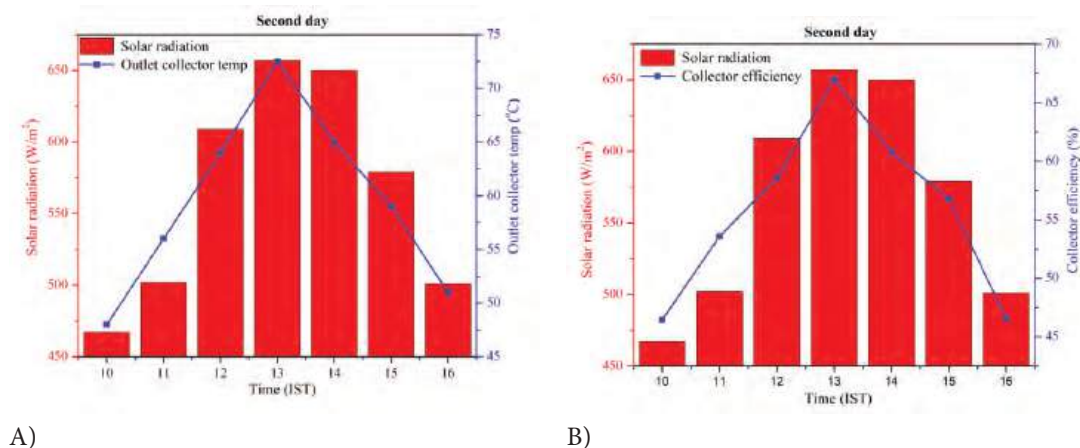


Figure 11. A. Variation of outlet collector temperature with incident solar radiation. B. Variation of efficiency incident solar radiation.

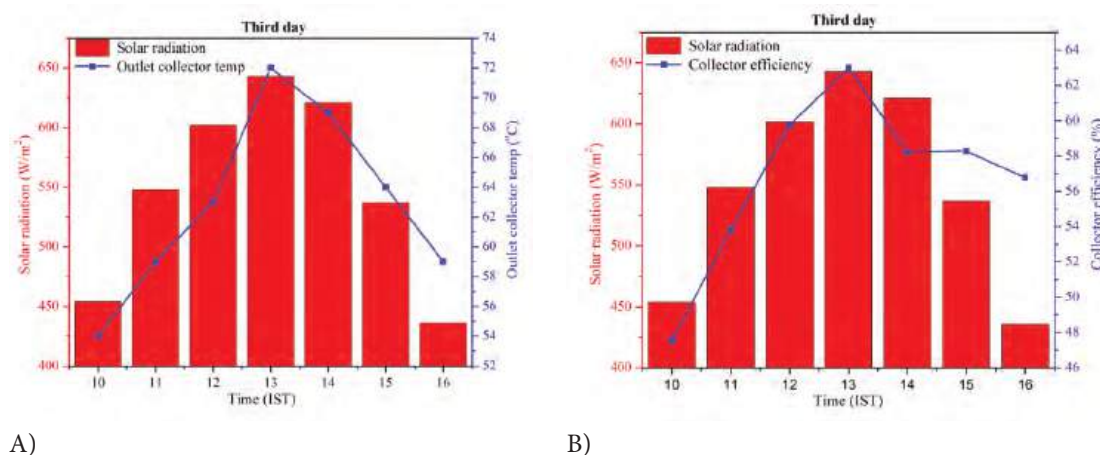


Figure 12. A. Variation of outlet collector temperature with incident solar radiation. B. Variation of efficiency incident solar radiation.

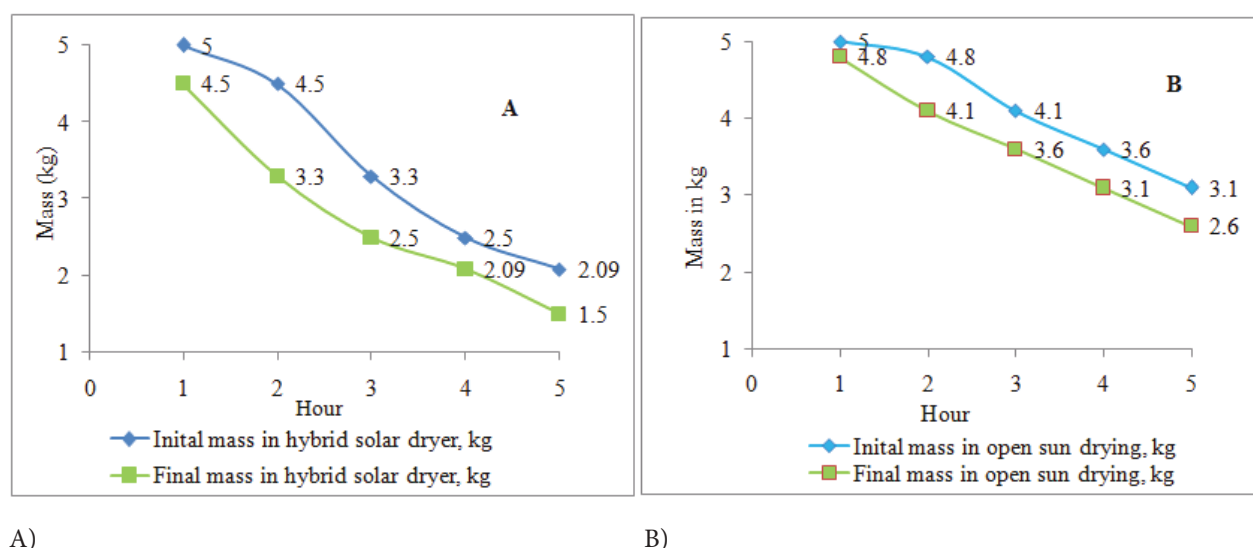


Figure 13. Initial and final mass of mangoes for five hours during the drying process. A. hybrid solar drying B. Open sun drying.

CONCLUSION

A hybrid solar dryer has been developed for day and night operation for drying mango slices and preparing bread toasts. The present work has been investigated with a new approach with the recirculation of exhaust hot air and turbulence generation through a baffle. Because of these provisions, the drying rate (0.95 kg/hr), drying efficiency (32.89%), collector efficiency (74.1%), and maximum drying temperature (69°C) are found quite improved over the previous studies. Economic indicators such as pay-back period (1.4394 years) and cost-benefit ratio (2.0008) have been evaluated. The above values being more than 1.0 signifies that the designed hybrid model is viable and feasible. The hybrid solar dryer can be utilized during the night

period using an electric heating coil powered by a battery, which is charged by a solar panel during the day time. It can be utilized for industrial applications such as food processing industries, agro-industries, and domestic purposes. The drying process in case of hybrid solar dryers is faster and more effective even in the adverse weather conditions.

Future Recommendations

The present work can be extended for future research by using different heat storage materials for drying purposes during night time and cloudy weather. Experiments can be performed inside the laboratory at a constant source of light intensity incident on a solar collector.

Table 7. Comparison of current findings with previous studies

Reference	Testing condition	Modification	Drying chamber temp (°C)	Collector efficiency (%)	Drying efficiency (%)
Present study	Specimen: Green Mangoes; Mass: 5 kg; Duration: 24 h	Using exhaust hot air recirculation	69	74.1	32.89
Hao et al. [19]	Specimen: Lemon slices; Mass: 4 kg; Duration: 27 h	Flat plate collector with dual function	48.97	44.6	9.5
Nwakuba et al. [5]	Specimen: sliced red pepper; Mass: 0.5 kg; Duration: 24 h	Solar dryer integrated with electric heater	60	40.6	24.4
Asnaz et al. [20]	Specimen: Mushroom slices; Duration: 9 h	Forced convection solar dryer	43.8	67.85	67.66
Wang et al. [21]	Specimen: Mango; Mass: 24 kg; Duration: 24 h	Indirect type forced convection solar dryer	52	49.06	30.9
Mugi, and Chandramohan [22]	Specimen: Guava slices; Mass: 800 gm; Duration: 14 h	Indirect type forced convection solar dryer	47.5	65.37	6.84
Gilago and Chandramohan [23]	Specimen: ivy gourd; Mass: 800 gm; Duration: 13 h	Indirect type forced convection solar dryer	32–62	77.2	7.8

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article

REFERENCES

- [1] Şirin C, Selimefendigil F, Öztöpe HF. Performance analysis and identification of an indirect photovoltaic thermal dryer with aluminum oxide nano-embedded thermal energy storage modification. *Sustainability* 2023;15:2422. [\[CrossRef\]](#)
- [2] Veeramanipriya E, Sundari AU. Performance evaluation of hybrid photovoltaic thermal (PVT) solar dryer for drying of cassava. *Solar Energy* 2021;215:240–51. [\[CrossRef\]](#)
- [3] Singh P, Gaur MK. Environmental and economic analysis of novel hybrid active greenhouse solar dryer with evacuated tube solar collector. *Sustainable Energy Technol Assess* 2021;47:101428. [\[CrossRef\]](#)
- [4] Singh P, Gaur MK. Heat transfer analysis of hybrid active greenhouse solar dryer attached with evacuated tube solar collector. *Solar Energy* 2021;224:1178–92. [\[CrossRef\]](#)
- [5] Nwakuba NR, Ndukwe MC, Asonye GU, Asoegwu SN, Nwandikom GI. Environmental sustainability analysis of a hybrid heat source dryer. *Polytechnica* 2020;3:99–114. [\[CrossRef\]](#)
- [6] Suherman S, Hadiyanto H, Susanto EE, Utami IAP, Ningrum T. Hybrid solar dryer for sugar-palm vermicelli drying. *J Food Process Eng* 2020;43:e13471. [\[CrossRef\]](#)

- [7] Srithanyakorn S, Bunchan S, Krittacom B, Luampon R. Comparison of mixed-mode forced-convection solar dryer with and without stainless wire mesh in solar collector. *Clean Energy* 2023;7:1316–29. [\[CrossRef\]](#)
- [8] Deef M, Samy Helal H, El-Sebaee I, Nadimi M, Paliwal J, Ibrahim A. Harnessing solar energy: A novel hybrid solar dryer for efficient fish waste processing. *AgriEngineering* 2023;5:2439–57. [\[CrossRef\]](#)
- [9] Salve S, Fulambarkar AM. Experimental studies on drying characteristics of green chilies in a solar dryer. *J Therm Eng* 2022;8:587–94. [\[CrossRef\]](#)
- [10] Srimanickam B, Kumar S. Drying investigation of coriander seeds in a photovoltaic thermal collector with solar dryer. *Int J Mech Eng Technol* 2021;14:659–68. [\[CrossRef\]](#)
- [11] Shrivastava A, Gaur MK, Singh P. Mango leather (Aam Papad) drying in hybrid greenhouse solar dryer with evacuated tube collector and finned drying tray: Drying behavior and economic analysis. *Energy Sources Part A Recover Util Environ Eff* 2025;47:2029974. [\[CrossRef\]](#)
- [12] Behera DD, Mohanty RC, Mohanty AM. Experimental investigation of a hybrid solar dryer for vegetable drying with and without phase change material. *J Braz Soc Mech Sci Eng* 2024;46:303. [\[CrossRef\]](#)
- [13] Heydari A, Forati M, Khatam S. Thermal performance investigation of a hybrid solar air heater applied in a solar dryer using thermodynamic modeling. *J Therm Eng* 2021;7:715–30. [\[CrossRef\]](#)
- [14] Awasthi A, Kallioğlu MA, Sharma A, Mohan A, Chauhan R, Singh T. Solar collector tilt angle optimization for solar power plant setup-able sites at Western Himalaya and correlation formulation. *J Therm Anal Calorim* 2022;147:11417–31. [\[CrossRef\]](#)
- [15] Padhi MR, Ghose P. Performance prediction of a model rotary air preheater through porous media approach. *J Therm Sci Eng Appl* 2023;15:071008. [\[CrossRef\]](#)
- [16] Behera DD, Mohanty RC, Mohanty AM. Thermal performance of a hybrid solar dryer through experimental and CFD investigation. *J Food Process Eng* 2023;46:e14386. [\[CrossRef\]](#)
- [17] Behera DD, Mohanty RC, Mohanty AM. Performance evaluation of indirect type forced convection solar mango dryer: A sustainable way of food preservation. *Therm Sci* 2023;27:1659–72. [\[CrossRef\]](#)
- [18] Ampah J, Dzisi KA, Addo A, Bart-Plange A. Drying kinetics and chemical properties of mango. *Int J Food Sci* 2022;2022:6243228. [\[CrossRef\]](#)
- [19] Hao W, Liu S, Mi B, Lai Y. Mathematical modeling and performance analysis of a new hybrid solar dryer of lemon slices for controlling drying temperature. *Energies* 2020;13:350. [\[CrossRef\]](#)
- [20] Asnaz MSK, Dolcek AO. Comparative performance study of different types of solar dryers towards sustainable agriculture. *Energy Rep* 2021;7:6107–18. [\[CrossRef\]](#)
- [21] Wang W, Li M, Hassanien RHE, Wang Y, Yang L. Thermal performance of indirect forced convection solar dryer and kinetics analysis of mango. *Appl Therm Eng* 2018;134:310–21. [\[CrossRef\]](#)
- [22] Mugi VR, Chandramohan VP. Comparison of drying kinetics, thermal and performance parameters during drying guava slices in natural and forced convection indirect solar dryers. *Solar Energy* 2022;234:319–29. [\[CrossRef\]](#)
- [23] Gilago MC, Chandramohan VP. Performance evaluation of natural and forced convection indirect type solar dryers during drying ivy gourd: An experimental study. *Renew Energy* 2022;182:934–45. [\[CrossRef\]](#)



Research Article

Experimental investigations on performance improvement of multi point sequential fuel injection engine by reducing back pressure

Chandrakumar PARDHI¹, Anshul GANGELE¹, Sanjay CHHALOTRE², C.P. JAWAHAR^{3,*}

¹Department of Mechanical Engineering, Amity University Madhya Pradesh, Gwalior, 474005, India

²Department of Mechanical Engineering, Sagar Institute of Science and Technology, Bhopal, 462036, India

³Department of Mechanical Engineering, School of Engineering and Technology, Alard University Pune, 411057, India

ARTICLE INFO

Article history

Received: 14 September 2024

Revised: 13 January 2025

Accepted: 14 January 2025

Keywords:

Exhaust Pipe; Backpressure;
Volumetric Efficiency; Exhaust
Emission; Throttle Position

ABSTRACT

Substantial developments in automotive technology have led to the recent explosion in the availability of high-performance vehicles. One of the most important components of internal combustion engines for reducing fuel consumption and exhaust emissions. Exhaust back pressure, a critical engine characteristic regulates the amount of internal exhaust gas trapped in the cylinder, affecting emissions and engine performance. The present work is focused on improving the performance of a multipoint sequential fuel injection engine by reducing the back pressure. The performance tests were carried out on a Hyundai i10 engine (1200 cc, multi-cylinder, 3000 rpm maximum speed). This study includes introducing fresh jet air at various flow rates into the exhaust system to reduce the concentration of exhaust gas contaminants and provide the desired backpressure while accounting for changes in engine speed. The effect of backpressure on various engine performance parameters at different engine speed conditions has been analyzed. It is inferred from the present study that at 2000 engine rpm, the temperature of exhaust gas drops between 50°C and 90°C and there is a marginal increase in the volumetric efficiency from 4.5 to 6.25%. The engine power increased about 6 to 8.2% at this operating condition.

Cite this article as: Pardhi C, Gangele A, Chhalotre S, Jawahar CP. Experimental investigations on performance improvement of multi point sequential fuel injection engine by reducing back pressure. J Ther Eng 2025;11(5):1497–1506.

INTRODUCTION

Exhaust manifolds, often called headers, collect exhaust gases from various internal combustion (IC) engine cylinders and direct them into a single pipe. The exhaust manifold's correct sizing is crucial since it influences fuel economy and emission efficiency [1]. The exhaust pipe

establishes a connection between this header and the cylinders. It plays a crucial role in multi-cylinder engines, consolidating numerous exhaust streams into a single pipe and connecting to the engine downstream [2].

The primary function of the exhaust system in automobiles is to reduce engine noise, but now, it is designed to

*Corresponding author.

*E-mail address: cpjawahar@gmail.com, cpjawahar@alarduniversity.edu.in

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkılıç



reduce noise as well as increase the reaction time between exhaust gas and the composition of the catalytic converter so that nitrogen oxides separate into nitrogen and oxygen [3]. Also, an exhaust system is essentially a lengthy tube with a fully developed boundary layer that slows the flow by rubbing against the walls; backpressure is an inevitable consequence. This results in an increase in the exhaust backpressure (EBP). Excessive back pressure can restrict the amount of fresh air entering the combustion chamber, reducing oxygen availability and potentially leading to incomplete combustion.

When the back pressure is high, the engine's performance decreases due to inefficient exhaust gas discharge. Loss of power and torque, increased fuel consumption, and increased emissions are typical outcomes of increased backpressure [4]. Due to back pressure, all the burnt gases are not discharged during the exhaust stroke and some amount of burnt gases are trapped in the cylinder. So, to get the exhaust gases out during the exhaust stroke, the piston must work harder [5]. Therefore, during the exhaust stroke, the piston must exert more force to expel the exhaust gases. Thus, the crankshaft's total power output decreases. Therefore, the pumping losses during the exhaust stroke increases proportionally to the increase in back pressure. This will increase the total power output [6].

To counteract the reduced airflow due to higher back pressure, turbocharging or supercharging could be one option. These options force additional fresh air into the combustion chamber, helping maintain a high airflow rate, even with moderate back pressure [7]. But as a concern with spark ignition engine, it increases the fuel evaporation rate which further leads to increasing excessive knocking (detonation), thermal losses and poor fuel efficiency.

In a turbocharged engine, a rise in back pressure leads to a decrease in the pressure drop across the turbine, resulting in a reduction in the speed of the turbocharger shaft. Consequently, the intake boost pressure and the mass flow of intake air also decrease [8]. The back pressure had a detrimental effect on the naturally aspirated engine. Increasing the diameter of the exhaust system can alleviate back pressure, but the confined space in passenger vehicles prevents the installation of a bigger exhaust system. Optimal engine performance requires the presence of backpressure, and eliminating it might result in a decrease in power and torque [9].

Excessive back pressure has significant effects on the operation of internal combustion engines, including reduced power production, increased fuel consumption, and heightened smoke emissions from the exhaust [10]. Therefore, minimizing the flow constraints in exhaust system design is crucial for mitigating the adverse effects of exhaust gases-induced back pressure. However, adhering to stringent pollution control regulations makes it exceedingly challenging to keep the necessary level of back pressure [11]. To maximize engine efficiency, minimize total emissions and keep ambient sound energy within acceptable

limits, back pressure must be optimized through careful design of the exhaust system, especially the muffler and exhaust manifold [12].

Methods to reduce back pressure:

Optimizing the exhaust system:

- Bigger pipes and smoother elbows: Reduce restrictions and improve flow.
- High-flow catalysts: Designed with less restrictive materials and structures.
- Performance exhaust systems: Larger diameter pipes and less restrictive mufflers are often used, but these can compromise noise levels.

Considering Engine Modifications:

- Camshaft modifications: Intake and exhaust opening overlap can be optimized for more efficient exhaust clearance.
- Light valves and springs: Reduce inertia and improve valve opening/closing speed to facilitate exhaust flow.
- Turbochargers and superchargers: Push more air into the engine, which may require reduced exhaust back pressure for optimal performance.

The review of literature indicates that the back pressure fluctuates with the injected air-flow rate, thus aiding in maintaining the engine's performance at an optimal level during increased engine revolutions per minute (rpm). Conversely, in some cases, increased backpressure effectively reduces the emission of nitrogen oxides (NO_x) in the exhaust gas due to higher temperatures [13]. Therefore, minimizing exhaust backpressure is crucial for improving the performance of any internal combustion engine, provided that the emission of nitrogen oxides (NO_x) remains unaffected. This study involves introducing fresh jet air at various flow rates into the exhaust system to reduce the concentration of exhaust gas contaminants and provide the desired backpressure while accounting for changes in engine speed.

LITERATURE REVIEW

Effects of Exhaust Backpressure on Efficiency, Power, and Fuel Use in Internal Combustion Engines [13].

Since the back pressure prevents the exhaust gas moving from the cylinder to the outside of the cylinder, the pistons in a spark ignition (SI) or compression ignition (CI) engine must use more energy to remove the exhaust gas. This causes the engine to lose some power and durability. The EBP study aims to establish the correlation between power and the power output of spark ignition and compression ignition engines [2]. The engine loses power due to back pressure, which increases the power to remove gas from the cylinder. For every 1 kPa decrease in exhaust pressure, engine power increases from 0.22 kW to 0.45 kW. Every 10 kPa decrease in back pressure reduces fuel consumption by approximately 1.5% to 3%. When the piston moves in

opposition to the exhaust gas, it can lead to exhaust blockage issues. When the catalyst back pressure decreases by 40%, then the torque increases by 5.5 Nm. The optimized exhaust silencer can recover 90% of the engine's lost torque by reducing the back pressure [13].

Effect of Exhaust Backpressure on Internal Combustion Engine Emission Determinants [14].

Exhaust backpressure typically results in a greater concentration of harmful pollutants, including carbon monoxide (CO), carbon dioxide (CO₂), nitrous oxides (NO_x), hydrocarbons (HC), and particulate matter (PM) [15]. Nitrogen oxide production normally peaks in cold weather, and nitrogen oxide emissions from diesel engines are generally higher. Most motor vehicles install exhaust gas recirculation (EGR) in their CI engines to reduce nitrogen oxides (NO_x). This increases the temperature of the exhaust gas [16]. Further, studies have shown that EGR enhances energy recovery, leading to longer combustion times and an increase in HC emissions. Improving EGR can reduce NO_x production in the internal combustion engine but will also lead to long combustion times, increased PM and HC emissions, and other adverse effects [17].

It is known that the fuel remaining in the cylinder will slow down the flame and therefore change its combustion properties. The breakdown that occurs when fuel residue lowers the temperature of the cylinder leads to reduced NO_x emissions. However, this process also increases carbon emissions, such as THC and CO, and increases engine power due to engine instability at high speeds (such as idle) [18]. The remaining gas in a cylinder slows the flame, which changes the combustion characteristics, as is well established. In addition, this process makes the car body vibrate more when the engine is not stable, which happens often during idling and other low-speed rotational situations [19].

This experimental study is focused on investigating the effect of lowering the back pressure change in response to fresh air addition in the exhaust pipe system on the performance of the petrol engine. It also focuses on optimizing the exhaust manifold design to reduce engine back pressure and enhance engine performance. Researchers have conducted numerous experiments on the parts of the exhaust pipe system, including the catalytic converter and silencer, to understand how backpressure impacts the performance and pollution levels of internal combustion engines.

Previous investigations have played a significant role in exploring further research through experimental analysis, engine simulations, and computational fluid dynamics (CFD) models [16]. Given its convergent geometry, carry out an experiment to ascertain the impact of a change in return force on the exhaust gas during the idling of both exhaust systems. The findings indicate that the shape of the junctions in the H-type exhaust pipe system maximizes the discharge pipe pressure. This impact leads to THC receiving the highest score in the emission results. However,

because of the influence of the connection geometry, the X-type exhaust system demonstrates the worst performance. Murali R. et al. [13] concluded their study on the impact of high back pressure on the performance of internal combustion engines. In general, increasing backpressure was determined to reduce engine performance and reduce the return of 40% of the torque of approximately 5.5 Nm. When back pressure increases, RGF increases and combustion efficiency decreases. The increase in EBP significantly impacted the small valve overlap angle. Further increase in EBP has affected residual exhaust gas, as the decrease in the valve overlap angle leads to an increase in residual exhaust gas [20].

Gülmez Y. et al. [18] carried out studies to investigate the negative effects of exhaust gas, pollution, and emissions of diesel engines. The results showed that the rear pressure increase could delay the combustion stage up to a 4° crank angle, reduce the average pressure, and reduce the cylinder pressure of the maximum engine load from 78.36 to 70.7 bar. When the backpressure increases to 24.66 kPa, brake specific fuel consumption (BSFC) increases by 3.29%. Leman AM. et al. [9] conducted a review of research comparing the performance of a catalytic converter with back pressure and cold start in internal combustion engines. One of the key concerns regarding a vehicle's engine combustion operation is a cold start. To minimize the amount of polluting gas exposed to the atmosphere, particularly during the cold-start phase, catalytic converter modification is an essential factor that must be taken into consideration. Additionally, we must handle the emission gas fluxes in the catalytic converter more carefully to prevent fuel waste and engine power losses.

Lee J. et al. [8] conducted a thorough investigation of the use of a turbocharger system to expand the working limits of a hydrogen spark ignition engine under heavy loads. The results indicate that higher boosts from the turbocharger helped high-load situations last longer. However, the high exhaust pressure stopped high-load situations from getting better at high speeds. Bhure S. [21] investigated the impact of back pressure on the efficiency and emissions of diesel engines equipped with electronic fuel injection. During the investigation, he manually controlled back pressure control valves (BPCVs) at three different levels: 100%, 87.5%, and 75% BPCV lifts. According to the data, there is a 15% decrease in NO_x and a notable decrease in HC and CO. However, exhaust smoke and BSFC levels have increased.

Sivaram A R. et al. [19] examined the impact of exhaust pipe length on a single-cylinder diesel engine's fuel economy and volumetric efficiency. The results of the study demonstrated that longer exhaust pipes, which raise backpressure, can result in higher fuel consumption and lower volumetric efficiency. Sapra H. et al. [22] studied a medium-speed diesel engine equipped at different speeds and loads compared to different values of static back pressure. The performance of a turbocharged engine, both pulsed and constant pressure, was evaluated in a validated model

simulation under two distinct valve overlap levels and a one-meter water column (mWC) high back pressure.

Bhandari R. et al. [6] identified which factors were most important in changing back pressure and how they were related. He determined the diameter of the punched holes, the distance between the holes, and the total number of holes. He uses a typical damper model to test the setup and understand how the different boundary conditions of the Fluent CFD analysis affect the results. K. S. Umesh et al. [12] conducted tests on the exhaust manifold of a four-stroke, four-cylinder engine at 1500 rpm. Improving fuel consumption is the primary goal of this effort, which employs both experimental and computational methods to decrease backpressure in the exhaust manifold. From a business perspective, the engine's total performance can be improved by modifying its geometry to use lengthy bends, changing its exit position (from center exit to reducer), and so on.

From the review of the literature, it is inferred that the researchers have primarily studied the effect of back pressure on internal combustion engine performance and its emissions. In general, exhaust back pressure negatively affects the efficiency of either an SI or CI engine. However, exhaust back pressure is not completely avoidable due to the muffler's intake and transmission losses, which are necessary to reduce the amount of sound energy released into the surrounding air. In addition, all harmful emissions of exhaust gases, except NO_x, increase under the influence of back pressure. There is a need to optimize the back pressure through careful design of the exhaust system, especially the muffler and exhaust manifold, to maximize engine efficiency, minimize exhaust emissions. Experiments were carried out to ascertain the impact of adding fresh air to the exhaust system on back pressure and its impact on automobile spark ignition engine performance. Furthermore,

emission characteristics related to engine power performance to assess the impact of decreased back pressure were examined.

EXPERIMENTAL SETUP

A 4-cylinder SI engine (Make: Hyundai Motor) was used for the experimental studies. Table 1 presents the engine specifications. To develop an experimental setup to inject fresh air into the exhaust pipe of a car, an air injection system, which includes an air pump or blower, air hoses and fittings, airflow control valves, and an air filter, has been used. For the exhaust system modifications, a fitting at a chosen point downstream from the engine, ideally near the catalytic converter, was welded into the exhaust pipe to serve as the air injection port. The nozzle was fixed from one end of the hose inserted into the exhaust pipe. The connector is provided at the other end of the hose to the blower outlet end. The steady fresh air flow rate is ensured by using a digital rotating vane anemometer. The maximum error rate of the device for mass flow rate measurement was less than $\pm 2\%$ for air. For measuring the back pressure, the easiest method is to install the back pressure gauge ($\pm 1\%$) to the oxygen sensor port by unscrewing it and then install the adapter provided in the kit by screwing the adapter into the oxygen port. The K-type thermocouples measured the temperature at the desired point in the exhaust pipe. The uncertainty of a K-type thermocouple is usually a maximum of $\pm 0.4^\circ\text{C}$. Launch C Reader VI engine scanner was used to measure engine rpm. The blower was connected to the variable speed controller to regulate the fresh air flow rate.

After that, pressure transducer, temperature measurement system and oxygen sensors were fixed at both upstream and downstream of the injection point to measure back pressure.

Table 1. Engine specifications

Item	Specification
Engine type	1.2 Kappa 4CYL Inline Naturally Aspirated MPSFIE
Total displacement	1197
Compression ratio	10.5:1
Number of cylinders	4
Number of valves per cylinder	4
Fuel supply system	MPFI
Maximum power at 6000 RPM	78.9 BHP
Maximum torque at 4000 RPM	111.7 N-m
Super Charge	Not Available
Bore & Stroke	71 x 75.6 mm
Firing order	1-3 - 4-2
Fuel	Gasoline

EXPERIMENTAL PROCEDURE

Experiment was carried out by varying the engine speed from 800 to 3000 rpm at some regular intervals. The measurements were noted firstly at the normally designed engine setup, and later on, the readings are taken with the injection of fresh air at the exhaust system. A blower is used to create the vacuum in exhaust pipes, hence facilitating the easier escape of exhaust gases from the cylinder. This exhaust fan creates a pressure difference by reducing the pressure outside the engine's cylinders. The forced intake of new air creates a zone of negative pressure within the engine's exhaust system. As a result, the combustion

chamber directs its exhaust gases toward the area of lower pressure and swiftly releases them into the atmosphere.

Measurements such as air pressure, atmospheric air temperature, throttle valve opening percentages, intake air mass flow rate, volumetric efficiency, brake torque, and engine power, while running the engine without air injection were taken at first. Then, we replaced the setup and injected fresh air at the desired flow rates of 2 m³/min, 3 m³/min and 4 m³/min. For one data point 4 set of readings are measured at different rpm. All the measurement values are very close to each other. The average value of each point is used to show the result. These specific airflow rates allow for an analysis of the system's efficiency at each

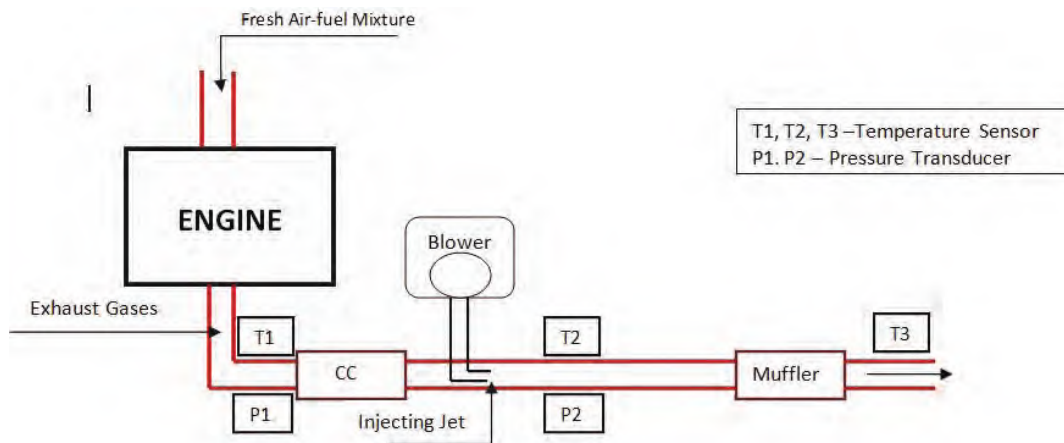


Figure 1. Schematic diagram illustrates the process of injecting fresh air into the exhaust system.



Figure 2. Pictorial view of exhaust system of experimental setup.

rate, thereby helping to identify optimal flow conditions for the desired performance outcomes. The back pressure was recorded while varying the engine speeds and airflow rates. The results of the back pressure measurements confirmed that there was a difference in back pressure that corresponded to the change in the flow rate of injecting fresh air into the exhaust pipe. This study examined the impact of changing the flow rate of injected fresh air on temperature and pressure by installing a pressure transducer and temperature sensors in an exhaust system. Figure 1 shows the schematic diagram of the setup of injecting fresh air into the exhaust system. Figure 2 represents the pictorial view of the exhaust system of the experimental setup with the arrangement of injecting fresh air.

RESULTS AND DISCUSSION

In conventional automobile engines, the exhaust valve opens prior to the completion of the power stroke and continues to remain open even after the commencement of the intake stroke. At the conclusion of the power stroke, the pressure within the cylinder exceeds the external pressure, facilitating the expulsion of exhaust gases from the cylinder. In this study, an exhaust fan to generate the necessary pressure differential and minimize the counteracting force has been employed so that the exhaust valve may remain closed for more time during the entire power stroke. This approach enabled us to harness the additional power that would otherwise be lost when the exhaust valve is opened prematurely to facilitate the expulsion of exhaust gases.

The variation of EBP with engine rpm for different flow rates of fresh air in the exhaust system is depicted in Figure

3. At lower engine speeds, the impact of pumping losses is insignificant. However, at higher speeds, the piston must exert a substantial amount of effort to expel the exhaust gases. Therefore, the increase in back pressure within the engine's exhaust system reduces the power output at the crankshaft.

Figure 4 illustrate the enhancement in volumetric efficiency resulting from a decrease in exhaust back pressure. In this situation, the enhancement in volumetric efficiency is a result of the gradual decrease in the amount of residual gases trapped in the combustion area. During the experiment we found that at flow of rates $2 \text{ m}^3/\text{min}$, at moderate speed of 2000 rpm, the system experiences reduced pressure losses and less frictional resistance, which tends to enhance volumetric efficiency. This enables the cylinders to draw in a greater amount of fresh air during the intake stroke, thereby enhancing the combustion process. An increase in the volumetric efficiency from 4.5 to 6.25% at 2000 rpm was observed during the study.

The impact of decreasing back pressure on the opening of the throttle valve is depicted in Figure 5. To evaluate the driver's desired acceleration relative to the complete depressed position of the accelerator pedal, the throttle valve opening is expressed as its percentage. The throttle position sensor on the accelerator control detects the driver's desired acceleration. In essence, reducing back pressure facilitates the expulsion of exhaust gases from the cylinder. This decreases the amount of force needed for the engine to pump, facilitating the downward movement of the piston during the exhaust stroke. As a result, the engine experiences less resistance, leading to a higher generation of power at the same throttle opening compared to situations

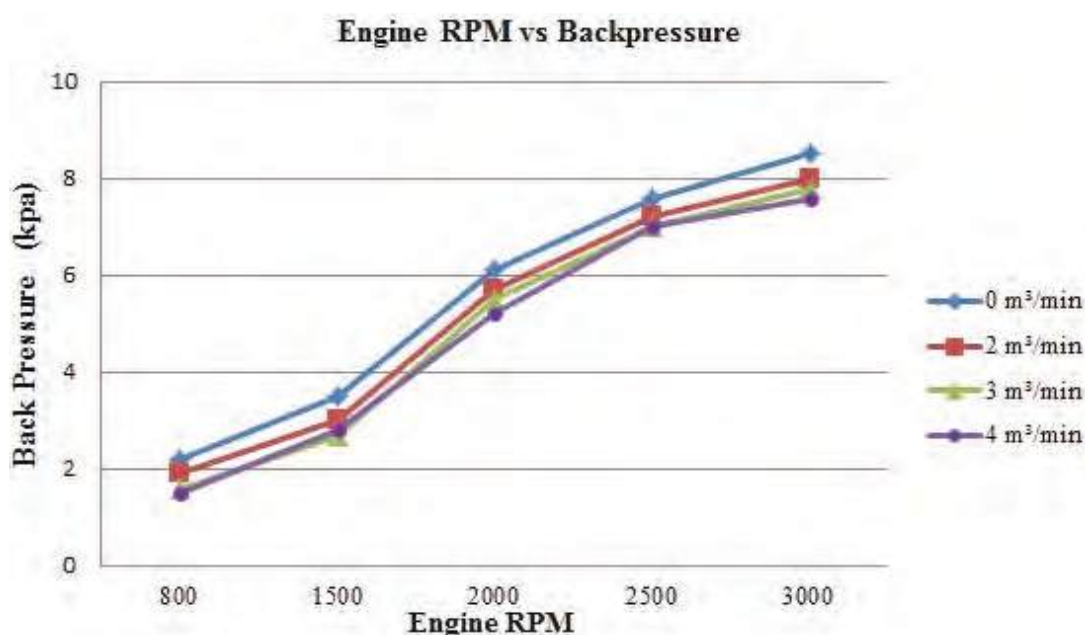


Figure 3. Variation of EBP with engine rpm for different flow rates of fresh air in the exhaust system.

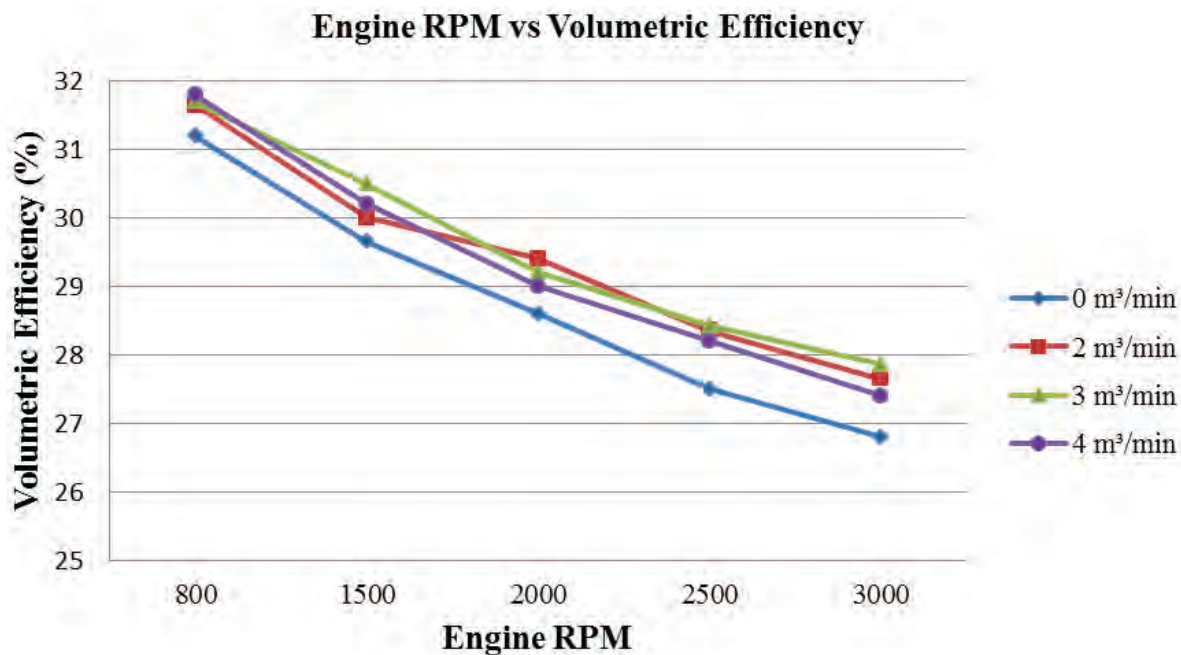


Figure 4. Variation of volumetric efficiency with engine rpm for different flow rates of fresh air in the exhaust system.

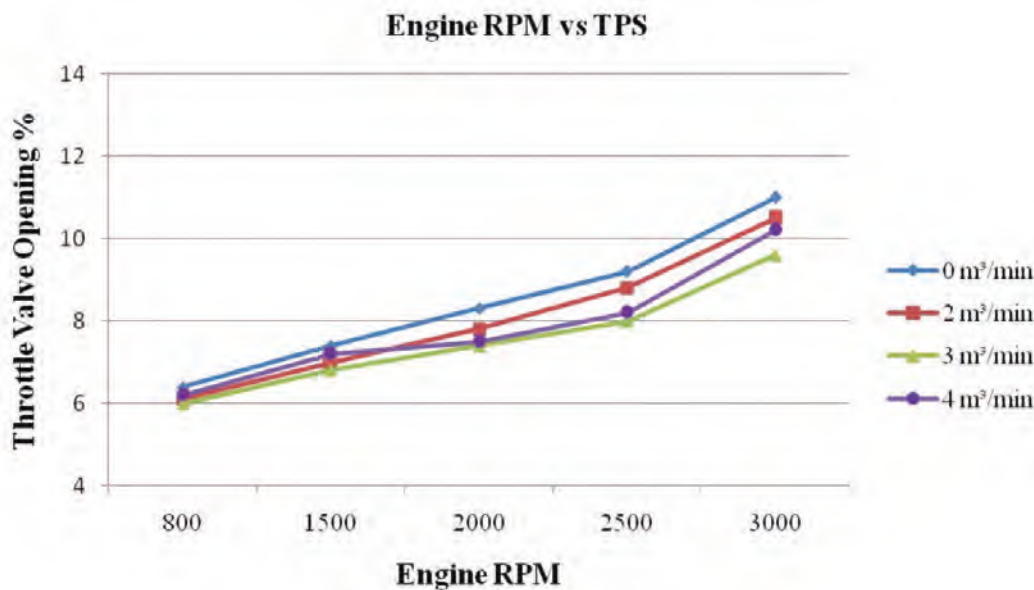


Figure 5. Variation of TPS % with engine rpm for different flow rates of fresh air in the exhaust system.

where the back pressure is greater. Therefore, if the target power output remains constant, the throttle may open to a slightly lesser extent in order to obtain that power as a result of the enhanced efficiency.

Figure 6 shows that reducing the back pressure in a SI engine generally leads to a decrease in the exhaust gas temperature. This is a result of an enhanced scavenging process. Reducing pressure in the lower back region enhances

the process of “scavenging” the cylinder, creating space for the intake stroke to accommodate the fresh air-fuel mixture. This extra amount of fresh air fuel mixture lowers the amount of heat absorption from the cylinder walls, which lowers exhaust temperatures. The results indicate a decrease in exhaust gas temperature between 50°C and 90°C.

Figure 7 depicts the variation of engine power with engine rpm at various airflow rates. As a result, the

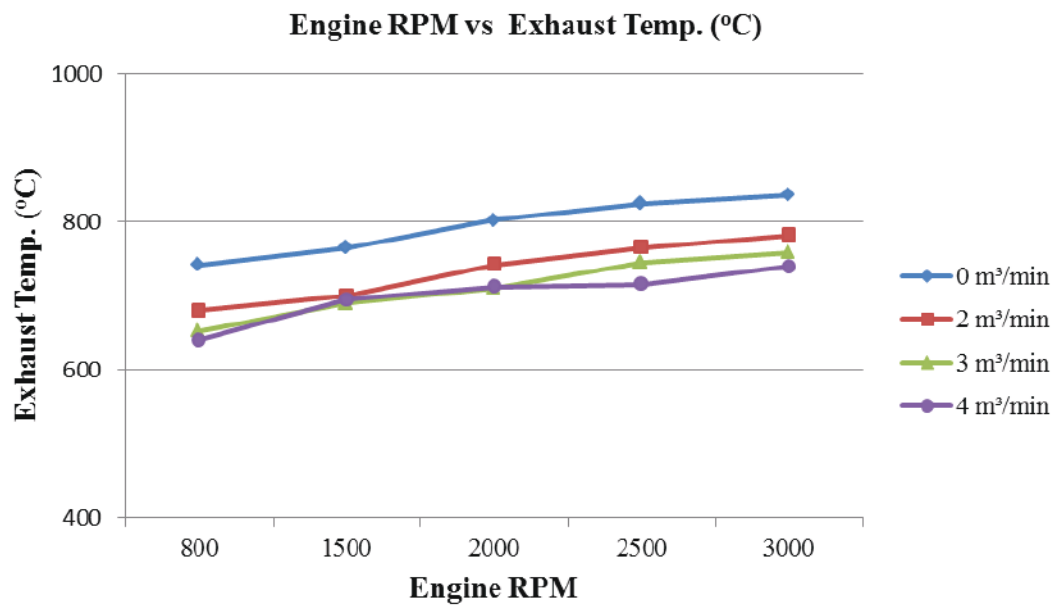


Figure 6. Variation of exhaust temperature with engine rpm for different flow rates of fresh air in the exhaust system.

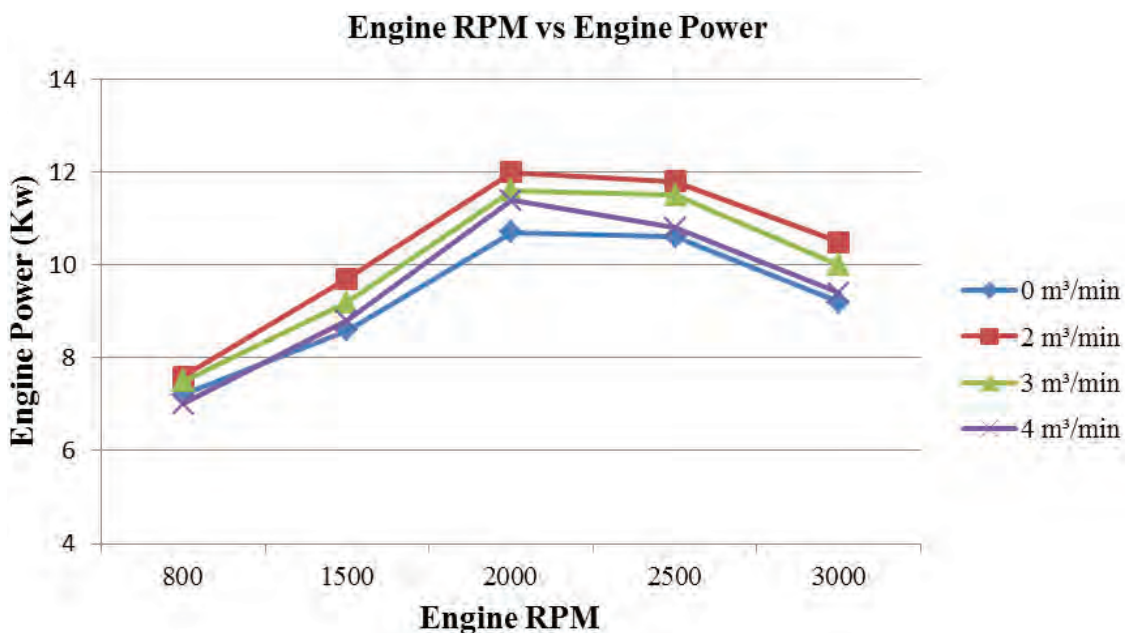


Figure 7. Variation of engine power with engine rpm for different flow rate of fresh air in exhaust system.

crankshaft's power output drops significantly at higher engine speeds. The engine scavenging efficiency could be enhanced by lowering the exhaust back pressure. As a result, the combustion chamber discharged exhaust gases more effectively during the exhaust stroke, making room for a greater quantity of fresh air-fuel mixture during the intake stroke. This enhanced breathing results in a boost in engine power. Lowering the back pressure indicates the engine power increases from 6 to 8.2%.

CONCLUSION

Experimental studies were carried out in the present work to ascertain the impact of reducing EBP on engine performance parameters in a naturally aspirated MPSFIE spark ignition engine. Excessive back pressure has several negative effects on internal combustion engine performance, including decreased power output, increased specific fuel consumption, and increased exhaust smoke. The

effects of backpressure reduction on the engine's exhaust emissions, volumetric efficiency, throttle valve position, and in-cylinder pressure characteristics were analyzed. The uncertainties associated with the measuring devices used in this study are considered. The results obtained are within an acceptable margin of error, supporting the accuracy and reliability of the results. The following conclusions are drawn from the present work.

1. As the engine's backpressure decreased, its specific fuel consumption also diminished. The reduction in leftover gases in the combustion chamber facilitates the intake stroke's ability to take in additional fresh air, thereby improving the combustion process and resulting in marginally higher volumetric efficiency from 4.5 to 6.25% at an air injection flow rate 2 m³/min.
2. Compared to the conditions of elevated back pressure, the engine experiences reduced resistance and can generate greater power at the same throttle position. Due to enhanced efficiency, the throttle may require a reduced opening to generate the same power output if the desired power level remains unchanged.
3. Reduced backpressure facilitates improved scavenging in the cylinder. Decreased back pressure enables a more complete expulsion of these heated gases, allowing room for the new air-fuel mixture to enter during the intake stroke. This reduces heat absorption from the cylinder walls, thus lowering exhaust temperatures.
4. Reducing exhaust back pressure improves the engine's scavenging efficiency. Consequently, the combustion chamber expels exhaust gases more efficiently during the exhaust stroke, hence accommodating a larger volume of fresh air-fuel combination during the intake stroke. This improvement leads to an increase in engine output from 6 to 8.2%.

The current study has some limitations; the power consumed by the air blower is not considered. However, in the study the power required to drive the blower can be taken from the battery. Further investigation can be conducted to optimize the engine operational parameters considering the power consumption of the blower.

AUTHORSHIP CONTRIBUTIONS

The authors have contributed equally to this work.

DATA AVAILABILITY STATEMENT

The authors have confirmed that the data supporting the findings of this study are available within the article. Raw data that support the findings of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

REFERENCES

- [1] Bober B, Andrych-Zalewska M, Boguś P. Influence of exhaust manifold modification on engine power. *Combust Engines* 2024;196:54–65. [\[CrossRef\]](#)
- [2] Venkatesan SP, Ganesan S, Devaraj R, Hemanandh J. Design and analysis of exhaust manifold of the spark ignition engine for emission reduction. *Int J Ambient Energy* 2020;41:659–664. [\[CrossRef\]](#)
- [3] Shekhar R, Singh Dhugga P, Malik K. CFD analysis of back pressure due to bend in exhaust pipe of 4 stroke petrol engine. *Int J Aerosp Mech Eng* 2016;3:3–5.
- [4] Deng X, Chen Z, Wang X, Zhen H, Xie R. Exhaust noise, performance and emission characteristics of spark ignition engine fuelled with pure gasoline and hydrous ethanol gasoline blends. *Case Stud Therm Eng* 2018;12:55–63. [\[CrossRef\]](#)
- [5] Kritsanaviparkporn E, Baena-Moreno FM, Reina TR. Catalytic converters for vehicle exhaust: fundamental aspects and technology overview for newcomers to the field. *Chemistry* 2021;3:630–646. [\[CrossRef\]](#)
- [6] Bhandari R, Joshi P, Ghimire A, Tamrakar A. Study and optimization of muffler to reduce back pressure of exhaust system. *SSRN Electron J* 2023. Preprint. doi: 10.2139/ssrn.4615719 [\[CrossRef\]](#)
- [7] Thirumurugaveerakumar S. Design and optimization of muffler back pressure. *AIP Conf Proc* 2020;2270. [\[CrossRef\]](#)
- [8] Lee J, Park C, Kim Y, Choi Y, Bae J, Lim B. Effect of turbocharger on performance and thermal efficiency of hydrogen-fueled spark ignition engine. *Int J Hydrogen Energy* 2019;44:4350–4360. [\[CrossRef\]](#)
- [9] Leman AM, Rahman F, Jajuli A, Zakaria S, Feriyanto D. Emission treatment towards cold start and back pressure in internal combustion engine against performance of catalytic converter: a review. *MATEC Web Conf* 2016;87:02021. [\[CrossRef\]](#)
- [10] Fernoaga V, Sandu V, Balan T. Artificial intelligence for the prediction of exhaust back pressure effect on the performance of diesel engines. *Appl Sci* 2020;10:7370. [\[CrossRef\]](#)
- [11] Liu H, Li Z, Zhang M, Xu H, Ma X, Shuai S. Exhaust non-volatile particle filtration characteristics of three-way catalyst and influencing factors in a gasoline direct injection engine compared to gasoline particulate filter. *Fuel* 2021;290:120065. [\[CrossRef\]](#)
- [12] Umesh KS, Pravin V, Rajagopal K. Experimental investigation and CFD analysis of a multi-cylinder four stroke SI engine exhaust manifold for optimal geometry to reduce backpressure and to improve fuel efficiency. *Int J Automob Eng* 2014;4:13–20.

-
- [13] Murali R, Shahrman AB, Razlan ZM, Ahmad WKW, Azizul AI, Rojan MA, et al. A review on the correlation between exhaust backpressure and the performance of IC engine. *J Phys Conf Ser* 2021;2051:012044. [\[CrossRef\]](#)
- [14] Murali R, Shahrman AB, Razlan ZM, Rojan MA, Azizul AI, Rani MFH, et al. A study on the significance of exhaust manifold's bending angle to the brake torque of 115cc SI engine. *J Phys Conf Ser* 2023;2643:012016. [\[CrossRef\]](#)
- [15] Choi Y, Lee J, Jang J, Park S. Effects of fuel-injection systems on particle emission characteristics of gasoline vehicles. *Atmos Environ* 2019;217:116941. [\[CrossRef\]](#)
- [16] Kang IS, Yang SM. The effect of back pressure change on exhaust emissions according to the confluence geometry of a dual exhaust system in idling. *Appl Sci* 2022;12:1855. [\[CrossRef\]](#)
- [17] Jang J, Lee J, Choi Y, Park S. Reduction of particle emissions from gasoline vehicles with direct fuel injection systems using a gasoline particulate filter. *Sci Total Environ* 2018;644:1418–1428. [\[CrossRef\]](#)
- [18] Gülmez Y, Özmen G. Effects of exhaust backpressure increment on the performance and exhaust emissions of a single cylinder diesel engine. *J Eta Marit Sci* 2021;9:177–191. [\[CrossRef\]](#)
- [19] Sivaram AR, Rajavel R, Jayakumar N, Vinothkumar M. Exhaust back pressure effect on the performance features of a diesel engine. *ARPJ Eng Appl Sci* 2017;12:5353–5356.
- [20] Huang L, Liu J, Liu R, Wang Y, Liu L. Experimental investigation on combustion and performance of diesel engine under high exhaust back pressure. *Machines* 2022;10:919. [\[CrossRef\]](#)
- [21] Bhure S. Effect of exhaust back pressure on performance and emission characteristics of diesel engine equipped with diesel oxidation catalyst and exhaust gas recirculation. *Int J Veh Struct Syst* 2018;10:199–203. [\[CrossRef\]](#)
- [22] Sapra H, Godjeva M, Visser K, Stapersma D, Dijkstra C. Experimental and simulation-based investigations of marine diesel engine performance against static back pressure. *Appl Energy* 2017;204:78–92. [\[CrossRef\]](#)



Research Article

Technical and economic feasibility analysis of 1kW Rooftop Solar Photovoltaic System for Mysuru, Karnataka, India

Raghavendra Deva Das RAO^{1,*} , Bommegowda Sadashive GOWDA²

¹Department of Mechanical Engineering, Visvesvaraya Technological University, Karnataka, 590018, India

²Department of Mechanical Engineering, Vidyavardhaka College of Engineering, Karnataka, 570002, India

ARTICLE INFO

Article history

Received: 13 May 2025

Revised: 21 July 2025

Accepted: 31 July 2025

Keywords:

Carbon Dioxide Output;
Economic Analysis; Financial
Feasibility; Return on
Investment; Rooftop Solar
Photovoltaic (PV) Systems; Solar
Irradiation

ABSTRACT

Rooftop solar photovoltaic (PV) systems have emerged as a key decentralized energy solution in response to rising electricity demand, climate change concerns, and the global shift toward low-carbon power generation. Despite favourable policies and falling technology costs, adoption in many regions remains suboptimal. This study investigates the techno-economic feasibility of 1 kW rooftop solar PV systems in Mysuru, Karnataka, India, an urban area with high solar potential, receiving an average Global Horizontal Irradiance (GHI) of approximately 5.5 kWh/m²/day across over 300 sunny days annually. A system configuration comprising monocrystalline or polycrystalline PV modules, a 1 kW inverter. The system is projected to generate 1,500–1,800 kWh annually, resulting in savings of up to ₹8,850 per year at the current residential tariff of ₹5.90/kWh. Installation costs range between ₹45,000 and ₹50,000, with government subsidies such as those under the PM Surya Ghar: Muft Bijli Yojana, potentially covering up to ₹78,000. The financial analysis indicates a payback period of 4–6 years and a return on investment of 15–20% over a 25-year operational lifespan, alongside annual carbon dioxide emission reductions of approximately 1 ton. While the system proves technically and economically viable, challenges persist, including upfront capital costs, limited public awareness, and spatial constraints. The study underscores the importance of policy support, public engagement, and community solar initiatives in scaling rooftop PV adoption and advancing India's renewable energy transition

Cite this article as: Rao RDD, Gowda BS. Technical and economic feasibility analysis of 1kW Rooftop Solar Photovoltaic System for Mysuru, Karnataka, India. J Ther Eng 2025;11(5):1507–1519.

INTRODUCTION

The use of renewable energy has been increasing dramatically in the past few years, and among the available technologies, the solar photovoltaic (PV) system has become one of the most attractive solutions

for home and commercial energy services. Most particularly, rooftop PV solar systems are emerging as an inexpensive and environmentally friendly power source [1,2]. Several studies have examined the technical and economic feasibility of such systems in various

*Corresponding author.

*E-mail address: raghavddasrao@gmail.com

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkılıç



geographical areas and have made significant contributions to their adoption [3–5].

In India, grid-connected solar PV plant techno-economic viability has been thoroughly investigated, and it has been demonstrated that such systems are scalable and financially effective [6–8]. Similarly, the performance of rooftop solar PV systems has been evaluated in regions like Eastern India, where they show high energy production and operational efficiency [9]. Reviews of solar PV systems have noted opportunities and challenges for scaling this technology, highlighting growth potential in both urban and rural settings [10–12].

Studies in Southeast Asia, particularly Malaysia, have evaluated grid-connected PV systems, focusing on economic advantages and system performance under tropical conditions [13,14]. In particular, techno-economic analyses in Malaysia have shed light on their potential and feasibility of grid-connection [15]. Global reviews of solar PV power generation have noted the increasing adoption of solar energy as a result of advancements in technology and cost reductions [16–18]. With the rapid increase of solar PV systems, research has provided data on operational performance, challenges, and environmental aspects [19–21].

Techno-economic studies of small-sized rooftop solar panels have been carried out, where case studies demonstrate cost and environmental benefits in India [22–24]. Efforts to optimize hybrid renewable energy systems, integrating PV with other renewables such as wind, have been conducted to improve efficiency and reliability [25,26]. In the Indian context, building-integrated PV systems have been explored for energy efficiency, showing the potential of solar energy to be incorporated into the built environment [27]. Reviews of solar thermal collectors and economic analyses of solar systems emphasize the importance of these approaches in improving overall energy efficiency [28,29].

Performance and cost studies have provided insights into the design of grid-scale PV applications, showing their feasibility for large-scale deployment [30–32]. Broader social, economic, and environmental impacts of renewable energy systems have also been evaluated, highlighting their contribution to sustainable development [33]. Advanced software tools for simulating and optimizing hybrid renewable energy systems have been developed, supporting the design of more efficient and cost-effective PV installations [34–36].

Challenges of integrating PV into existing energy infrastructure, including technical, financial, and policy bottlenecks, have been discussed extensively [37]. Rural electrification studies have emphasized solar PV as a viable option for decentralized, off-grid power supply, alleviating global energy poverty [38]. Environmental factors such as dust, humidity, and air speed significantly affect PV performance, and strategies have been proposed to mitigate these effects [39]. Recent technological improvements, including high-efficiency solar cells and modules, have further reduced costs and enhanced energy generation [40]. New approaches such as phase-change material-based cooling and machine learning-driven performance

evaluation have also been investigated to improve the operational efficiency of PV systems [41].

The objective of this paper is to assess the technical and economic viability of a 1 kW rooftop solar PV system in Mysuru, Karnataka, by integrating results from a wide range of studies on solar PV performance, cost, and integration. Based on energy generation, cost-effectiveness, and ecological viability, the research provides useful information to estimate the potential of rooftop solar PV installations in the area. This analysis also offers guidance for implementing solar energy solutions in India and other developing countries.

Novelty and Research Gap

While numerous studies have explored the techno-economic viability of rooftop solar PV systems in various parts of India and Southeast Asia, most focus on either large-scale installations, generalized models, or major metropolitan regions [42]. However, localized assessments that incorporate micro-climatic data, site-specific irradiance conditions, policy incentives, and dynamic performance modeling for small-scale (1 kW) rooftop PV systems remain limited in the literature, especially for Tier-2 Indian cities like Mysuru. Existing studies often overlook the integration of region-specific irradiance profiles, real-world consumption patterns, and government subsidy impacts under the latest schemes, such as the *PM Surya Ghar: Muft Bijli Yojana*. Furthermore, few studies evaluate inverter performance, energy losses, solar fraction, and lifecycle carbon mitigation alongside economic metrics such as Levelized Cost of Energy (LCOE), ROI, and payback period in a unified framework [43].

To address these gaps, this study presents a detailed technical and financial analysis of a 1 kW grid-connected rooftop solar PV system for Mysuru by incorporating realistic simulation data (using Meteonorm 8.2), system loss factors, financial incentives, and CO₂ offset calculations. The work is novel in its region-specific application, its integrated environmental–economic evaluation, and in proposing scalable policy recommendations for broader adoption in similar urban settings across developing countries. This study contributes to a replicable methodology for decentralized renewable energy deployment, bridging the gap between technical feasibility studies and on-ground implementation strategies [44].

SYSTEM UNDER STUDY

Site Details

The project is based in Mysuru, Karnataka, India, located at latitude 12.31° N and longitude 76.65° E, with an altitude of 737 meters above sea level. The site operates in the UTC+5.5-time zone. The weather data used for this project is synthetic and sourced from Meteonorm 8.2, covering the years 1996–2015 with 100% satellite data (Sat=100%). The albedo value, which represents the reflection coefficient of the surface, is set at 0.20.

Table 1. Site geographical parameters

Parameter	Details
Geographical Site	Mysuru, India
Latitude	12.31° N
Longitude	76.65° E
Altitude	737 m
Time Zone	UTC +5.5
Weather Data Source	Meteonorm 8.2 (1996–2015), Synthetic, Sat=100%
Project Settings	Albedo: 0.20

Table 2. Site geographical parameters

Category	Details
System Type	Grid-Connected System
Orientation	Fixed Plane
Tilt/Azimuth	20° / 0°
3D Scene Definition	No 3D scene defined, no shading
Near Shadings	No shadings
User's Needs	Fixed constant load: 1000 W
Global Energy	8760 kWh/year
PV Array Information	
- Single module Power rating	100W
- Number of Modules	10 units
- Nominal Total Power	1000 Wp
Inverter Information	
- Number of Units	1 unit
- Nominal Total Power	1200 W

System Configuration

Here is the provided information formatted in tabular form:

This system is a grid-connected photovoltaic (PV) system with a fixed-plane orientation. The tilt angle of the solar panels is set at 20°, with an azimuth of 0°, aligning the panels for optimal sunlight capture. There is no 3D shading model or nearby obstructions causing shading, ensuring unimpeded solar energy generation. The system meets a fixed constant load demand of 1000 W and generates an annual energy output of 8760 kWh. The PV array consists of 10 modules with a combined nominal power output of 1000 Wp. Energy conversion is handled by a single inverter rated at 1200 W, with a performance ratio of 0.833, indicating efficient system operation.

Single Line Diagram

The diagram illustrates the flow of energy in a photovoltaic (PV) system. Below are the components and their roles:

1. Solar Modules (1 x ASE-100-DG-UR/mono, 5 Strings):
The system consists of a single array of solar panels labelled ASE-100-DG-UR/mono. It is configured in 5 strings, which are parallel connections of multiple

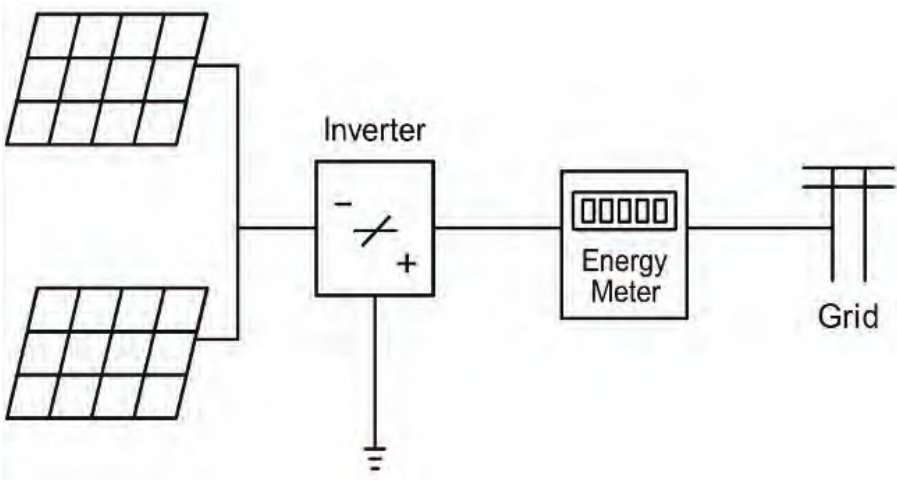


Figure 1. Single line diagram of a 1kW grid-connected Solar PV system.

Table 3. Power generation and performance ratio

Parameter	Value
Produced Energy	1492.6 kWh/year
Used Energy	8760.0 kWh/year
Specific Production	1493 kWh/kWp/year
Performance Ratio (PR)	77.83 %
Solar Fraction (SF)	17.04 %

panels. The array supports 2 Maximum Power Point Tracking (MPPT) mechanisms to optimize the power output from the panels.

2. Inverter (1 kVA): The direct current (DC) generated by the solar modules is sent to an inverter. The inverter has a power rating of 1 kVA and is equipped with 2 MPPTs to efficiently convert DC into alternating current (AC).
3. Grid: The AC electricity from the inverter is transmitted to an injection point for utilization to the grid. The energy is measured in kilowatt-hours (kWh) at this point, which typically represents the grid connection or the end-user consumption point.

Power Generation and Performance Ratio

The table 3 provides a summary of the energy performance of a solar system. The produced energy represents the total amount of electricity generated by the system in a year, which amounts to 1492.6 kWh. Meanwhile, the used energy is significantly higher, standing at 8760.0 kWh/year, indicating that the energy demand exceeds the supply from the solar system.

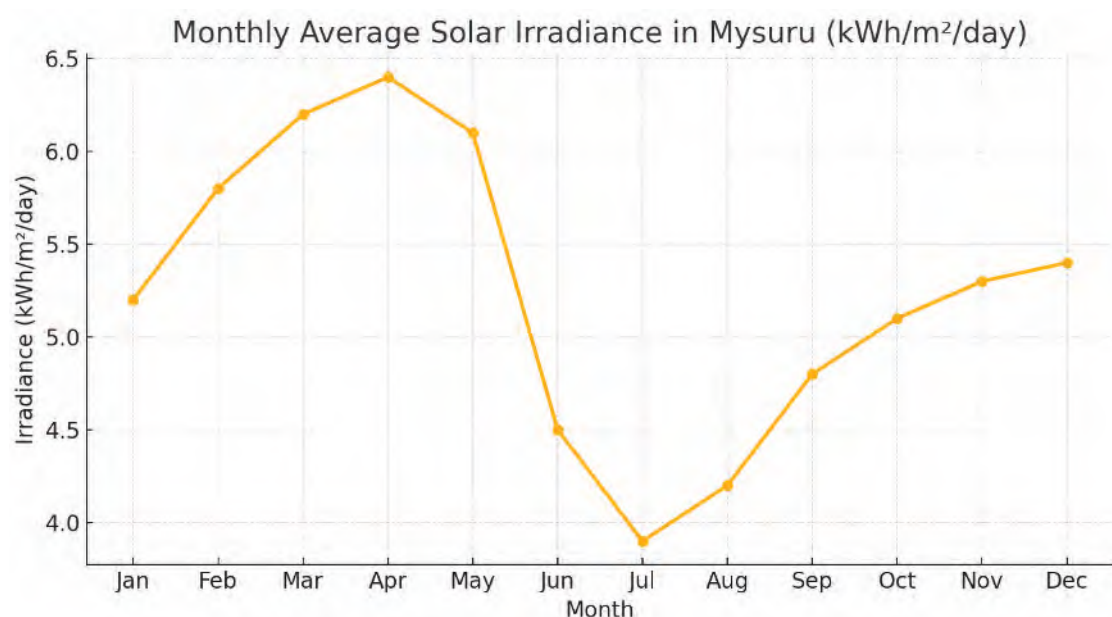
The specific production value of 1493 kWh/kWp/year reflects the efficiency of the system in converting solar energy into usable electricity. The performance ratio (PR), which is 77.83%, is a measure of the system's efficiency after accounting for losses due to environmental and system factors. Finally, the solar fraction (SF), at 17.04%, shows the percentage of the total energy consumption that is met by the solar system.

The annual energy generation of 1,492 kWh from the 1 kW rooftop solar PV system in Mysuru aligns well with performance outcomes reported in comparable Indian cities with similar solar irradiance. For example, rooftop systems in Bangalore—located within the same climatic zone and receiving comparable average Global Horizontal Irradiance (GHI) of around 5.5–5.6 kWh/m²/day—have shown specific yields in the range of 1,450 to 1,520 kWh/kWp/year. Similarly, in Hyderabad, which benefits from slightly higher solar exposure (GHI ~5.9–6.2 kWh/m²/day), studies have reported annual yields between 1,550 and 1,600 kWh/kWp. These comparisons confirm that the system performance in Mysuru is consistent with regional expectations, validating the simulation-based analysis and underscoring the city's suitability for rooftop solar PV deployment under typical urban conditions.

Other Simulated Results

Irradiance profile

The Figure 2 illustrates the monthly average solar irradiance in Mysuru, measured in kilowatt-hours per square meter per day (kWh/m²/day), highlighting the seasonal variation in solar energy availability throughout the year. The data shows that solar irradiance is highest between

**Figure 2.** Solar Irradiance profile.

February and May, peaking in April at approximately 6.4 kWh/m²/day, indicating a strong potential for solar energy generation during these months. Conversely, the lowest irradiance occurs in July, dropping to around 3.9 kWh/m²/day, which coincides with the region's monsoon season, when cloud cover and rainfall reduce solar exposure. After July, the irradiance values gradually increase, reaching around 5.4 kWh/m²/day by December. This trend confirms that Mysuru experiences substantial solar radiation for most of the year, making it a suitable location for rooftop solar photovoltaic (PV) installations. Understanding this monthly variation is crucial for optimizing system design, energy yield estimation, and financial planning, particularly for grid-connected or hybrid solar systems intended to supply consistent power throughout the year.

Power handling characteristics

The Figure 3 illustrates the power handling characteristics of a 1 kW rooftop solar PV system over a typical day, emphasizing the influence of ambient temperature on energy output. The nominal energy at STC (Standard Test Conditions), shown by the black dashed line, peaks at approximately 6.35 kWh around 12:30 PM, representing the theoretical maximum energy under ideal conditions. The virtual energy at MPP, depicted by the blue dotted line, accounts for actual environmental conditions and peaks slightly lower, at about 5.9 kWh. The inverter output energy, indicated by the red solid line, reflects the usable

energy after losses and reaches a maximum of around 5.1 kWh. The graph shows energy generation beginning around 6:00 AM, increasing steadily, and tapering off by 6:30 PM, following the sun's path. The ambient temperature, represented by the green dashed line (right y-axis), rises from approximately 5°C in the early morning to a peak of 26°C around noon, before decreasing in the evening. The observed drop in inverter output energy in the afternoon, despite continued high irradiance, highlights the impact of elevated temperatures on PV performance. This quantification confirms that thermal effects, along with inverter and system losses, reduce actual output compared to theoretical expectations, emphasizing the importance of temperature management in PV system design and evaluation.

Energy exchange characteristics

Figure 4 illustrates the energy exchange characteristics of the 1 kW rooftop solar PV system over a 24-hour period, highlighting the interaction between solar energy generation and grid dependency. The green curve represents the available solar energy, which begins around 6:00 AM, peaks at approximately 870 W near 12:00 noon, and drops back to zero by 6:30 PM, following the solar irradiance pattern. The red curve shows the energy drawn from the grid, which is highest during early morning and late evening hours—reaching 1,000 W when solar generation is absent—and reaches a minimum of approximately 170 W during peak solar hours. The blue line indicates the total energy supplied

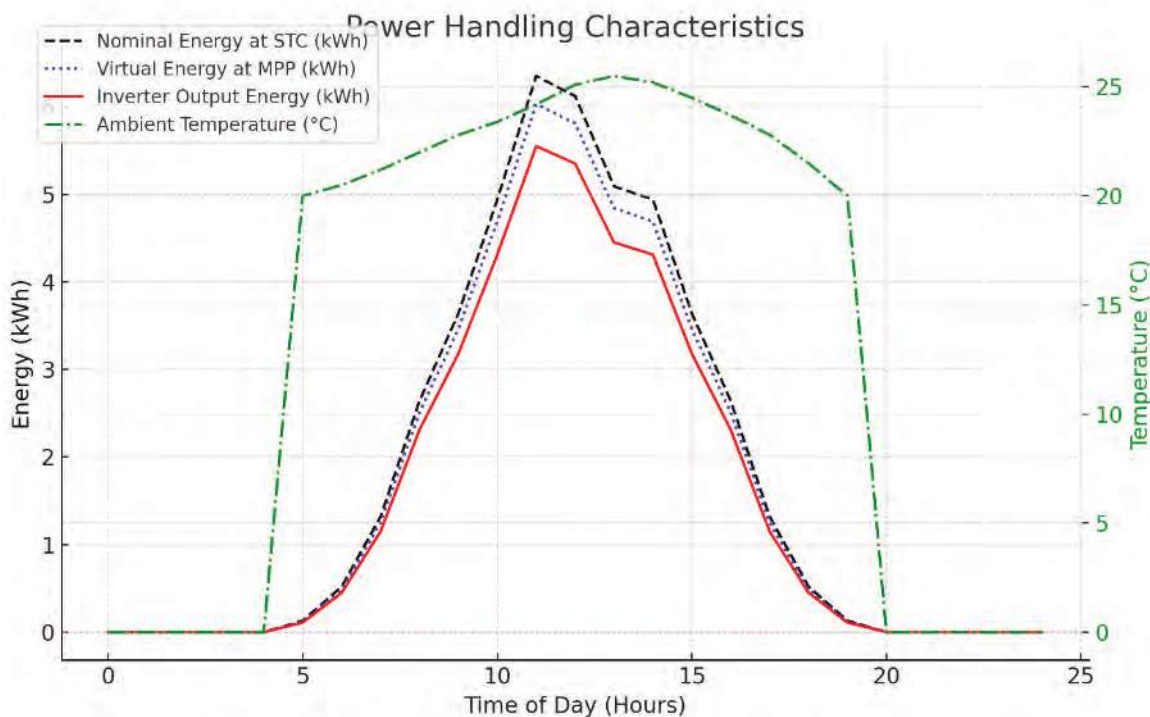


Figure 3. Power handling characteristics representing array virtual energy at MPP, inverter output as a function of ambient temperature.

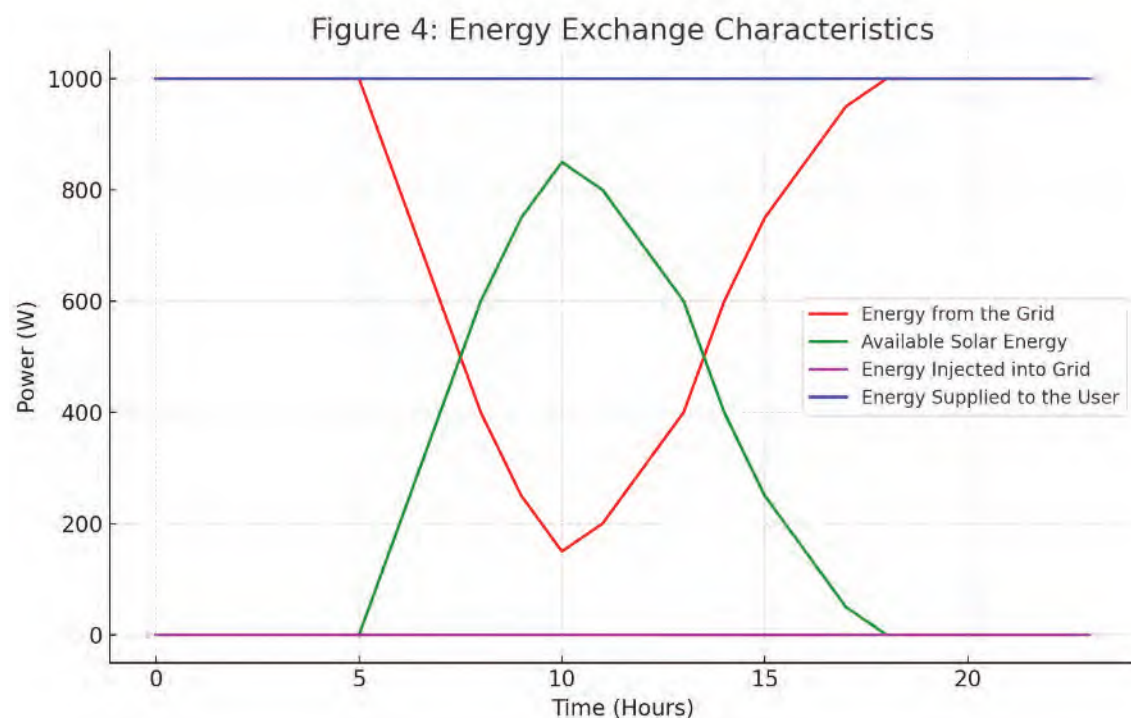


Figure 4. Energy exchange characteristics.

to the user, which remains constant at 1,000 W throughout the day, reflecting a fixed load demand. Notably, the purple line—representing energy injected into the grid—remains flat at zero, indicating that all solar energy is consumed locally and no excess power is exported. This pattern highlights the system's role in reducing grid dependency during daylight hours, but also emphasizes the need for grid support during non-solar periods. The absence of grid export suggests the system is either not connected under a net metering arrangement or is undersized relative to user demand.

Comparison of inverter output energy vs. nominal energy at stc

Figure 5 shows a comparison between the nominal energy output at Standard Test Conditions (STC) and the actual inverter output energy of a 1 kW solar PV system over a 24-hour period. The black dashed line represents the nominal energy output at STC, which peaks at approximately 6.35 kWh around 12:30 PM, indicating the theoretical maximum energy generation under ideal conditions. In contrast, the red solid line illustrates the actual inverter output energy, which accounts for real-world losses such as temperature effects, system inefficiencies, and inverter losses.

The inverter output starts rising from around 6:00 AM, peaks at 5.09 kWh by 11:00 AM, and then gradually declines, reaching nearly zero by 7:00 PM. The noticeable difference between the nominal and actual output during

midday, particularly the gap between 6.35 kWh (nominal) and 5.09 kWh (actual) highlights the impact of performance losses. This includes temperature-induced efficiency drops, which become more significant during peak sunlight hours. Overall, the graph clearly illustrates how real-world conditions reduce the energy output below theoretical maximum levels, underscoring the importance of factoring in system losses for accurate performance and financial modeling.

Solar PV voltage current characteristics

The figure shows the Voltage-Current (V-I) characteristic curve of a solar PV module. As the voltage increases from 0 to 40 V, the current decreases non-linearly from about 8 A to 0 A, illustrating the typical behaviour of a PV cell. The curve highlights how current output drops with rising voltage, and it helps identify the maximum power point (MPP), the optimal operating point for maximum energy output.

Loss diagram

The loss diagram summarizes the energy flow and losses in the 1 kW rooftop solar PV system, from solar irradiation to usable AC output. Starting with a global horizontal irradiation of 1941 kWh/m², about 2.6% is lost due to tilt and orientation (IAM factor), resulting in 1903.7 kWh of nominal array energy. Further losses due to temperature (12.7%), module quality, mismatch, and wiring reduce the energy to 1603.2 kWh at the Maximum Power Point (MPP). Inverter-related losses, such as efficiency losses and power threshold

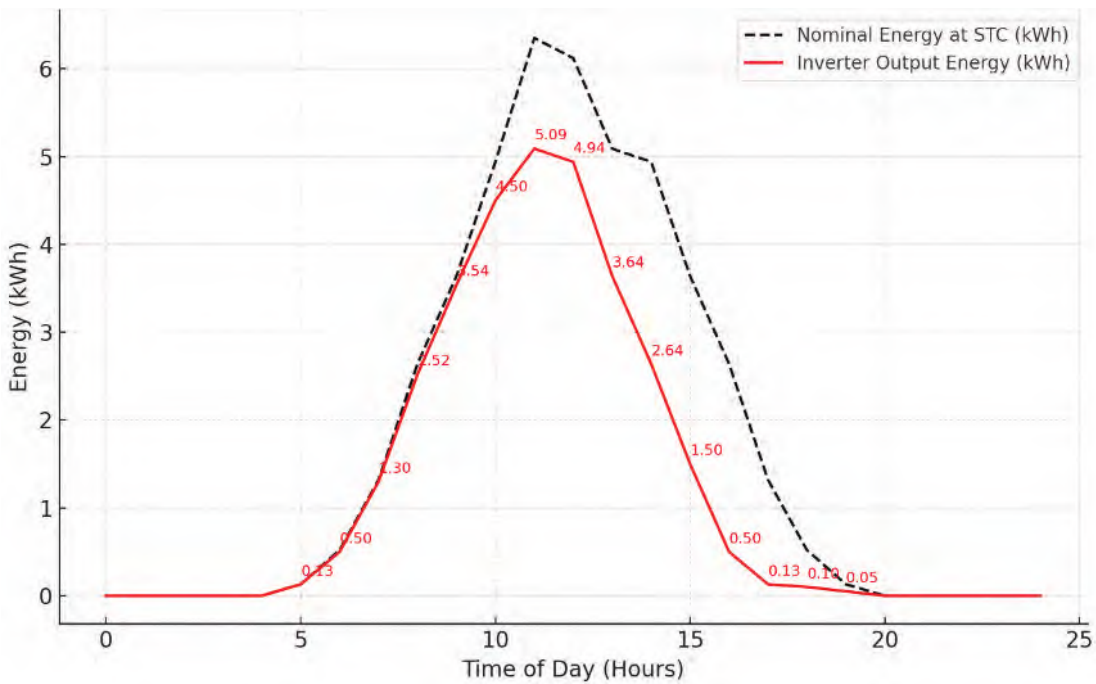


Figure 5. Comparison of Inverter Output Energy vs. Nominal Energy at STC.

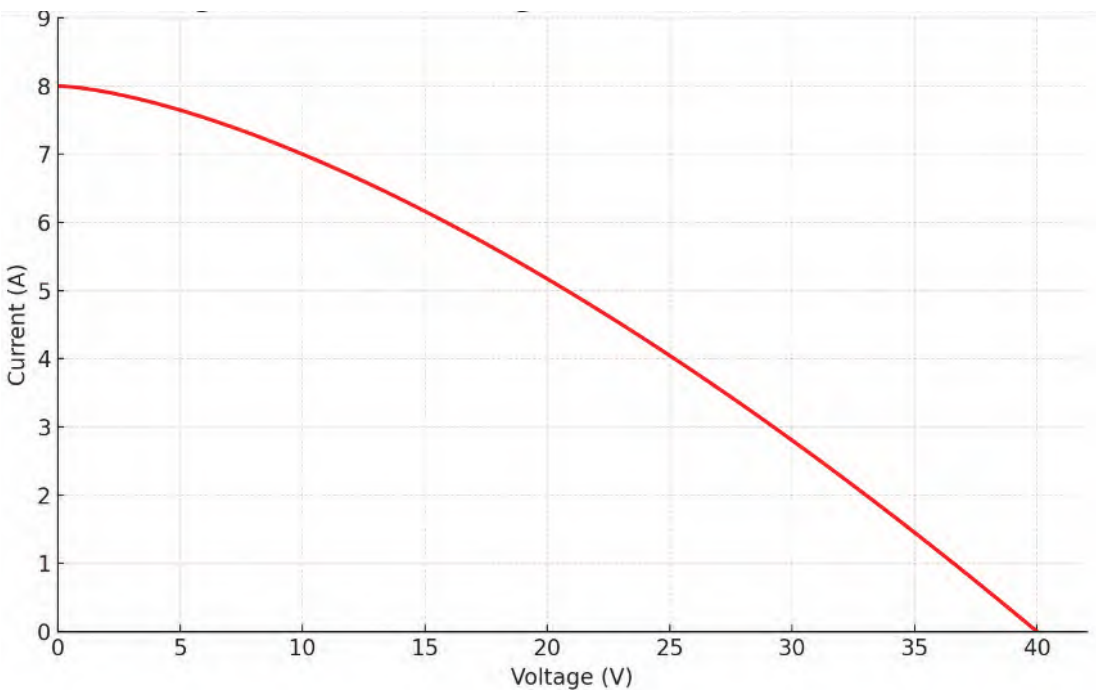


Figure 6. Voltage Current Characteristics.

effects, account for 6.9%, yielding 1492.6 kWh as the final available energy at the inverter output. This is the energy supplied to the user, while no energy is injected into the grid. The diagram clearly shows that while the system is efficient, about 23% of total potential energy is lost, mainly due to temperature and inverter inefficiencies.

Energy Balance Statistics

Across different indicators such as energy production, system efficiency, and yearly pattern of use, the information shown in Table 4 offers a thorough summary of solar energy performance. Looking at these monthly and yearly patterns will help us to learn much about the efficiency of the system

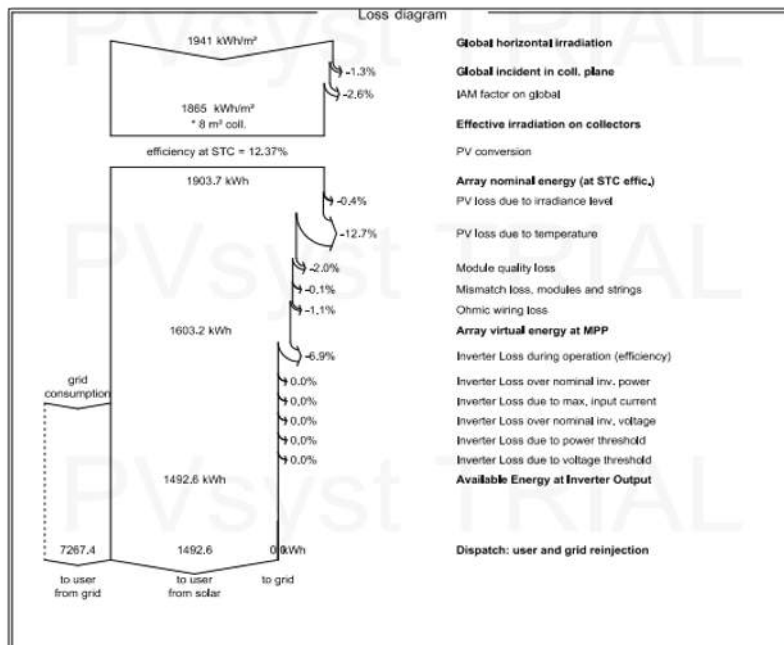


Figure 7. Loss diagram.

Table 4. Energy balance statistics

Month	GlobHor (kWh/m ²)	DiffHor (kWh/m ²)	T_Amb (°C)	GlobInc (kWh/m ²)	GlobEff (kWh/m ²)	EArray (kWh)	EUser (kWh)	ESolar (kWh)	EGrid (kWh)	EFGrid (kWh)
January	171.1	51.39	22.17	197.2	192.9	164.7	744.0	155.4	0.00	588.6
February	168.7	55.00	24.48	181.8	177.9	149.9	672.0	141.4	0.00	530.6
March	195.6	69.25	26.89	195.4	191.0	159.5	744.0	148.0	0.00	594.2
April	188.4	72.03	27.57	176.4	171.9	144.4	720.0	135.1	0.00	584.9
May	187.1	81.41	28.46	162.6	157.5	135.7	744.0	125.9	0.00	618.1
June	154.7	82.88	24.94	129.5	125.3	110.4	744.0	103.1	0.00	640.9
July	135.7	83.83	24.08	116.2	112.1	100.1	744.0	96.0	0.00	633.4
August	153.3	76.19	23.77	144.7	140.7	122.0	744.0	113.5	0.00	630.5
September	153.9	76.11	23.99	155.1	151.1	130.3	744.0	121.1	0.00	622.9
October	150.9	71.11	23.89	155.1	150.7	130.3	744.0	121.1	0.00	622.9
November	138.2	56.01	22.48	152.0	148.3	127.4	720.0	118.8	0.00	601.2
December	148.0	54.88	21.87	170.8	166.7	143.5	744.0	134.5	0.00	609.5
Year	1940.7	837.02	24.37	1915.3	1864.7	1603.2	8760.0	1492.6	0.00	7267.4

and possible opportunities for development. The information gives rise to a thorough examination below.

By examining this solar energy installation, one can see several main patterns and get some ideas on its general year-long efficiency and performance. The information begins with Global Horizontal Irradiance (GlobHor), which has an annual total of 1940.7 kWh/m²; the data indicates that March, April, and May represent the top solar energy output months. Ideal for solar power harvesting are these months

since they have more irradiance and longer daylight hours. Likewise, reflecting good solar panel angle and tracking, Global Incident Irradiance (GlobInc) adds up to 1915.3 kilowatt hours per square meter annually and follows a parallel pattern. The correlation of these parameters indicates that the system is well-designed to optimize energy capture.

The annual system's Global Efficiency (GlobEff) of 1864.7 kWh/m² is very close to the GlobInc values, suggesting little energy loss throughout the capture procedure.

This shows the consistent performance and great reliability of the solar system. As for energy output, the system produces 1603.2 kWh yearly, with peak manufacturing from March to May. But the production falls a bit in June and July, which might be due to reduced sunlight or the effect of greater temperatures, since heat lowers the effectiveness of solar panels.

On the usage side, the **EUser** data shows a consistent energy demand of 744.0 kWh every month, adding up to 8760.0 kWh annually. Solar energy contributes approximately 1492.6 kWh annually (**ESolar**), covering about 17% of the total energy needs. This makes a significant impact on reducing reliance on external power sources. Notably, the system does not draw any energy from the grid (**EGrid** = 0.00 kWh), demonstrating its ability to meet real-time energy demands entirely through solar power. Furthermore, the system offsets 7267.4 kWh of fossil fuel-based energy (**EFGrid**) annually, showcasing its contribution to reducing carbon emissions and supporting environmental sustainability.

The annual average ambient temperature of 24.37°C also affects the functioning of the system, together with the above-mentioned factors. Since lower temperatures reduce heat-related energy losses, cooler months like January, February, and December correspond with marginally better efficiency. Particularly in hotter seasons, this drives home the need to manage heat to preserve ideal system performance.

From these observations, several possibilities for betterment present themselves. From February to May, the system runs very smoothly, implying that extra energy produced during these months could be saved for less productive times like June, July, and December. Further lowering dependence on supplementary energy sources could come from investing in energy storage systems or increasing the capacity of the system. Implementing policies to control heat in warmer months, like air conditioning, could also help to raise general efficiency. In essence, the solar energy system works dependably and offers a significant contribution to decreasing reliance on traditional energy sources. By

balancing major fossil fuel consumption, it effectively produces clean energy year-round. This review underlines its advantages and presents down-to-earth ideas for improvement; hence, it is a useful tool for knowledge and enhancement of solar energy facilities in parallel situations.

Economic Feasibility Analysis

The presented analysis offers a comprehensive evaluation of the solar energy system's cost structure, operational efficiency, and energy production. By examining installation expenses, yearly maintenance costs, and energy output, the data provides valuable insights into the system's economic feasibility and long-term sustainability. This assessment highlights the balance between affordability and performance, showcasing the potential of solar energy as a reliable and environmentally friendly power source.

Installation costs

This analysis looks at the cost structure, operational costs, and energy production of the solar energy system. By looking at installation costs, annual maintenance costs,

Table 6. Operating Cost

Item	Total (INR/year)
Maintenance	12,000.00
Provision for Inverter Replacement	30,000.00(if required)
Cleaning	10000
Total (OPEX)	50000

Table 7. System cost summery

Parameter	Value
Total Installation Cost	50,000.00 INR
Operating Costs	50000 INR/year
Produced Energy	1493 kWh/year
Cost of Produced Energy	2.0099 INR/kWh

Table 5. Installation cost

Item	Quantity (Units)	Cost (INR)	Total (INR)
PV Modules (ASE-100-DG-UR/mono)	10	200.00	2,000.00
Supports for Modules	10	1,000.00	10,000.00
Inverters (EZH1)	1	30,000.00	30,000.00
Wiring	1	2,000.00	2,000.00
Combiner Box	1	2,000.00	2,000.00
Monitoring System, Display Screen	1	2,000.00	2,000.00
Measurement System, Pyrometer	1	2,000.00	2,000.00
Total Depreciable Asset			42,000.00
Total Installation Cost			50,000.00

and energy output, the data gives us an idea of the system's viability and long-term sustainability. This balances affordability and performance and shows solar energy as a reliable and eco-friendly power source.

This analysis gives a clear picture of the cost structure, operational expenses, and energy production efficiency of the solar energy system. Installation costs total 50,000 INR, which includes depreciable assets and auxiliary components. The largest investment is in the inverter (30,000 INR) as it is the heart of the system that converts and manages energy from the photovoltaic modules. The PV modules cost 2,000 INR, and the remaining components – supports, wiring, combiner box, monitoring system, and measurement equipment – cost 18,000 INR. These components ensure system durability, accurate performance monitoring, and overall functionality.

In terms of operational costs, the annual expenses are very low at 2,000 INR, out of which 1,000 INR is for maintenance, 1,000 INR for cleaning, and a provision for inverter replacement. These low operational costs make the system affordable and manageable over time.

The system summary shows key performance metrics. The system generates 1493 kWh annually, the levelized cost of energy (LCOE) is 2.0099 INR/kWh. This cost is competitive with conventional power sources and is a good option to reduce long term energy expenses. Also the system is very efficient so most of the produced energy is utilized and minimal wastage.

From this, we can see the system is designed for cost and reliability. But there is scope to optimize energy production by adding more PV modules or improving energy storage solutions to utilize excess energy. This can also reduce the LCOE and add more value to the system. Overall, the system balances installation cost, operational cost, and energy output and is a sustainable and eco-friendly energy solution.

Government tariffs

In the context of Mysuru, Karnataka, the residential electricity tariff is mentioned as ₹5.90/kWh. This tariff serves as the baseline for calculating energy savings when offsetting grid electricity with solar PV generation.

- Savings from Solar: The system generates around 1,500–1,800 kWh annually, resulting in a potential annual saving of up to ₹8,850, calculated as $1500 \text{ kWh} \times ₹5.90$.
- Cost of Produced Energy (LCOE): ₹2.0099/kWh, which is significantly lower than the residential tariff, highlighting the economic attractiveness of rooftop solar under current tariff structures.

Grid buy-back rates

The manuscript indicates that no energy is exported back to the grid (i.e., $E_{Grid} = 0.00 \text{ kWh/day}$), suggesting that the system is either:

- Operating in self-consumption mode only, or
- Connected without an active net metering agreement that allows feed-in.

In Karnataka, grid buy-back rates for rooftop solar systems have historically ranged between ₹2.50 to ₹4.00/kWh depending on system size and DISCOM regulations. However, this benefit is not realized in the current configuration in your study due to zero export.

Net metering benefits and potential

Although not fully implemented in the analyzed system, net metering can provide significant additional benefits:

- Under net metering, excess solar power fed to the grid earns credits, which can offset electricity usage during non-solar hours (e.g., nighttime).
- If implemented, this would:
 - Improve solar fraction (currently 17.04%) by allowing for better utilization of excess daytime energy.
 - Reduce grid dependency (currently supplying 18.20 kWh/day) without necessarily requiring energy storage.
 - Enhance financial returns by monetizing otherwise curtailed or unused generation.

Given Karnataka's support for rooftop solar under the PM Surya Ghar: Muft Bijli Yojana, and similar schemes, a properly structured net metering policy would:

- Increase the annual return on investment (currently estimated at 15–20%).
- Reduce the payback period (4–6 years) further.
- Align with national renewable energy targets by promoting decentralized, clean power generation.

Carbon Emission Balance

This section evaluates the carbon emission reduction potential of a 1 kW rooftop solar photovoltaic (PV) system installed in Mysuru, Karnataka. The assessment is based on updated emission factors, system generation data, and life cycle emission estimates.

Annual emissions avoided

Based on the estimated annual energy generation of 1,500 kWh, and using a grid emission factor of 0.8 kg CO₂/kWh for India as per the Central Electricity Authority (CEA), the system can offset:

$$\text{Annual Avoided Emissions} = 1,500 \text{ kWh} \times 0.8 \text{ kg CO}_2/\text{kWh} = 1,200 \text{ kg CO}_2 \text{ or } 1.2 \text{ metric tons CO}_2$$

Emissions over 25-year lifespan

Assuming consistent output and no major degradation losses beyond standard assumptions, the system will offset approximately:

$$1.2 \text{ metric tons CO}_2/\text{year} \times 25 \text{ years} = 30 \text{ metric tons CO}_2 \text{ over its lifetime}$$

Life cycle emissions of system components

To determine the net carbon impact, emissions from manufacturing and installing the solar system must also be considered. Life cycle emissions data for a 1 kW PV system are approximately:

$$\text{PV Modules: } 1,700 \text{ kg CO}_2$$

- Mounting Structure: 600 kg CO₂
- Inverter: 600 kg CO₂
- Balance of System: 200 kg CO₂
- Total Lifecycle Emissions: ~3,100 kg CO₂ (3.1 metric tons)

7.4 Net Emission Reduction

Net CO₂ Offset = Emissions Avoided - Lifecycle Emissions

Net CO₂ Offset = 30 tons - 3.1 tons = 26.9 metric tons CO₂

This analysis demonstrates the significant environmental benefit of adopting rooftop solar, with the 1 kW system yielding a net reduction of nearly 27 metric tons of CO₂ over its useful life.

CONCLUSION

This study comprehensively assessed the technical, economic, and environmental feasibility of a 1 kW rooftop solar photovoltaic (PV) system in Mysuru, Karnataka—a Tier-2 Indian city with substantial solar potential. Simulation results based on Meteonorm data and PVSyst modeling confirm that the system can generate approximately 1,492.6 kWh annually, with a performance ratio of 77.83% and a solar fraction of 17.04%, aligning well with observed yields in comparable urban regions such as Bangalore and Hyderabad. The levelized cost of energy (LCOE) is estimated at ₹2.01/kWh, significantly lower than the local residential grid tariff of ₹5.90/kWh, leading to potential annual savings of up to ₹8,850.

The financial analysis reveals a payback period of 4–6 years and a return on investment ranging between 15–20%, especially when factoring in central subsidies under schemes like PM Surya Ghar: Muft Bijli Yojana. Environmentally, the system is projected to achieve a net carbon offset of 26.9 metric tons of CO₂ over a 25-year lifespan, even after accounting for lifecycle emissions of the system components. Although the system does not currently export energy to the grid, introducing net metering and grid buy-back mechanisms could further improve energy utilization and economic returns.

Overall, the study validates that small-scale rooftop PV systems are both technically viable and financially attractive in urban Indian contexts. Nonetheless, broader adoption will depend on improved public awareness, financing mechanisms to reduce upfront costs, integration of energy storage, and policy measures that enable energy exchange with the grid. The methodology and results from this case study offer a scalable template for similar cities aiming to transition toward decentralized and sustainable energy solutions.

Limitations of the System

1. Limited Scale of Study:

The analysis is based on a 1kW system, which may not represent the performance or cost dynamics of larger or commercial-scale systems.

2. Site-Specific Assumptions:

The findings are tailored to Mysuru's climatic, solar irradiance, and electricity tariff conditions, limiting generalizability to other regions.

3. Static Financial Parameters:

Assumptions like inflation, interest rates, and panel costs are taken as fixed for simplicity, which may not reflect future market variability.

4. Exclusion of Degradation and Maintenance Dynamics:

The system performance over time may degrade due to aging of PV panels, inverter failures, or soiling, which are not dynamically modeled.

5. No Battery Storage Considered:

The study does not account for energy storage systems, which could significantly alter technical and economic feasibility.

6. Grid Reliability and Policy Dependence:

Feasibility is highly influenced by current net metering policies and grid infrastructure, which may change in the future.

Future Scope of the System

1. Scaling to Higher Capacities:

Future work can extend the analysis to 5kW, 10kW, or higher capacity systems to evaluate economies of scale and commercial viability.

2. Integration of Battery Storage:

Adding energy storage systems like lithium-ion batteries could improve reliability and allow for greater self-consumption, especially in grid-unstable regions.

3. Dynamic Performance Modeling:

Incorporating seasonal variability, long-term degradation, and real-time meteorological data can yield more accurate performance predictions.

4. Techno-Economic Optimization:

Optimization techniques (e.g., HOMER, PVSyst simulations) can be used to determine the most cost-effective system configurations under varying conditions.

5. Policy and Incentive Impact Assessment:

Simulating different tariff structures, government subsidies, and policy scenarios could inform better decision-making for stakeholders.

6. Environmental Impact Analysis:

Future research can include a Life Cycle Assessment (LCA) to quantify the environmental benefits in terms of CO₂ reduction and resource use.

7. Smart Grid and IoT Integration:

Investigating the integration of rooftop solar systems with IoT-based monitoring and smart grids for better energy management.

NOMENCLATURE

PV	Photovoltaic
GHI	Global Horizontal Irradiance (kWh/m ² /day)
PR	Performance Ratio (%)

SF	Solar Fraction (%)
STC	Standard Test Conditions
MPPT	Maximum Power Point Tracking
DC	Direct Current
AC	Alternating Current
kWp	Kilowatt peak (rated power of PV module/system)
kWh	Kilowatt-hour (unit of electrical energy)
LCOE	Levelized Cost of Energy (₹/kWh)
ROI	Return on Investment (%)
INR	Indian Rupee (₹)
CO ₂	Carbon Dioxide
T _{amb}	Ambient Temperature (°C)
E _{Array}	Energy generated by the PV array (kWh)
E _{User}	Total user energy demand (kWh)
E _{Solar}	Energy supplied by solar system (kWh)
E _{Grid}	Energy drawn from grid (kWh)
E _{FGrid}	Fossil fuel energy offset from grid (kWh)
GlobHor	Global Horizontal Radiation (kWh/m ²)
DiffHor	Diffuse Horizontal Radiation (kWh/m ²)
GlobInc	Global Incident Radiation on PV plane (kWh/m ²)
GlobEff	Effective Global Radiation after losses (kWh/m ²)
LCE	Lifecycle Emissions (kgCO ₂ or tCO ₂)

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Windarta J, Saptadi S, Denis, Satrio DA, Silaen JS. Economic feasibility analysis of rooftop solar power plant design with household-scale on-grid system in Semarang city, E3S Web of Conferences, 2020;202:09002. [\[Crossref\]](#)
- [2] Ahsan S, Javed K, Rana AS, Zeeshan M. Design and cost analysis of 1 kW photovoltaic system based on actual performance in Indian scenario. *Int J Eng Res Appl* 2016;6:17-22. [\[Crossref\]](#)
- [3] Kumar A, Sudhakar K. Techno-economic analysis of 1 MWp grid-connected solar PV plant in India. *Int J Ambient Energy* 2016;37:1-6.
- [4] Sharma R, Goel S. Performance analysis of an 11.2 kWp rooftop grid-connected PV system in Eastern India. *Energy Reports* 2017;3:76-84. [\[Crossref\]](#)
- [5] Chandel SS, Naik MN, Chandel R. Review of solar photovoltaic systems in India: Opportunities and challenges. *Renew Sustain Energy Rev* 2014;40:431-441.
- [6] Kumar NM, Sudhakar K, Samykano M. Techno-economic analysis of 1 MWp grid-connected solar PV plant in Malaysia. *Int J Ambient Energy* 2019;40:434-443. [\[Crossref\]](#)
- [7] Khatib T, Mohamed A, Sopian K, Mahmoud M. Assessment of grid-connected photovoltaic systems in Malaysia. *Int J Photoenergy* 2012;2012:1-9. [\[Crossref\]](#)
- [8] Ali HM, Jangam A. Techno-economic feasibility analysis of solar photovoltaic power generation: A review. *Smart Science* 2017;5:171-180.
- [9] Kumar A, Kumar K. Current status and future aspects of solar photovoltaic power generation in India. *Int J Eng Res Technol* 2015;4:717-721.
- [10] Sinha S, Schandel SS. Review of recent trends in optimization techniques for solar photovoltaic-wind based hybrid energy systems. *Renew Sustain Energy Rev* 2015;50:755-769. [\[Crossref\]](#)
- [11] Kumar R, Sudhakar K. Performance evaluation of 10 MW grid-connected solar photovoltaic power plant in India. *Energy Reports* 2015;1:184-192. [\[Crossref\]](#)
- [12] Shukla AK, Sudhakar K, Baredar P. Design, simulation, and economic analysis of standalone rooftop solar PV system in India. *Solar Energy* 2016;136:437-449. [\[Crossref\]](#)
- [13] S. A. Kalogirou, "Solar thermal collectors and applications," *Prog. Energy Combust. Sci.*, vol. 30, no. 3, pp. 231-295, 2004. [\[Crossref\]](#)
- [14] Singh P, Singh N. Comprehensive study of grid-integrated solar photovoltaic systems. *Renew Sustain Energy Rev* 2016;54:1010-1023.
- [15] Kalogirou SA. Economic analysis of solar energy systems. *Renew Sustain Energy Rev* 2010;14:2843-2852.
- [16] Tiwari H, Pandey AK. Performance analysis of grid-connected rooftop solar PV systems in India. *Energy Sustain Dev* 2018;43:130-138.
- [17] Kumar R, Sudhakar K. Techno-economic analysis of a rooftop solar PV system: A case study of a hostel building. *Energy Reports* 2017;3:76-84.
- [18] Green MA. *Solar cells: Operating principles, technology, and system applications*. Englewood Cliffs, NJ: Prentice-Hall; 1982.

- [19] Parida B, Iniyan S, Goic R. A review of solar photovoltaic technologies. *Renew Sustain Energy Rev* 2011;15:1625-1636. [\[Crossref\]](#)
- [20] Akella AK, Saini RP, Sharma MP. Social, economic, and environmental impacts of renewable energy systems. *Renew Energy* 2009;34:390-396. [\[Crossref\]](#)
- [21] Sinha SS, Chandel SS. Review of software tools for hybrid renewable energy systems. *Renew Sustain Energy Rev* 2014;32:192-205. [\[Crossref\]](#)
- [22] Shukla AK, Sudhakar K, Baredar P. Renewable energy resources in South Asian countries: Challenges, policy and recommendations. *Resource-Efficient Technol* 2017;3:342-346. [\[Crossref\]](#)
- [23] Ismail MS, Moghavvemi M, Mahlia TMI. Techno-economic analysis of an optimized photovoltaic and diesel generator hybrid power system for remote houses in a tropical climate. *Energy Convers Manag* 2013;69:163-173. [\[Crossref\]](#)
- [24] Mekhilef S, Saidur R, Kamalisarvestani M. Effect of dust, humidity and air velocity on efficiency of photovoltaic cells. *Renew Sustain Energy Rev* 2012;16:2920-2925. [\[Crossref\]](#)
- [25] Green MA. Recent developments in photovoltaics. *Solar Energy* 2004;76:3-8. [\[Crossref\]](#)
- [26] Chaurey A, Kandpal TC. Assessment and evaluation of PV based decentralized rural electrification: An overview. *Renew Sustain Energy Rev* 2010;14:2266-2278. [\[Crossref\]](#)
- [27] Chandel SS, Nagaraju Naik M, Chandel R. Solar powered water pumping systems: A review. *Solar Energy* 2016;128:218-227
- [28] Abdallah SM. Techno-economic feasibility of hybrid solar-wind power generation in the Middle East region. *Energy Convers Manag* 2013;72:17-24.
- [29] Badran AHA. Optimization of solar-wind hybrid system for remote areas: Case study. *Renew Energy* 2012;38:1561-1567.
- [30] Li RSL, Mahmud HAA, Al-Lawati SSS. Hybrid wind and photovoltaic power systems for remote areas. *Energy* 2012;38:1144-1152
- [31] Hargreaves JL. Analysis of hybrid photovoltaic and wind energy systems. *Renew Energy* 2014;39:2651-2660.
- [32] Gupta GP, Sharma PK. Techno-economic evaluation of hybrid renewable energy systems: A review. *Renew Sustain Energy Rev* 2016;60:1079-1091.
- [33] Ali HM, Jangam A. Use of phase change materials for enhancing the efficiency of photovoltaic systems. *J Renew Sustain Energy* 2014;6:1-14.
- [34] Sohail MMTP, Islam MMS. Integration of photovoltaic-thermal systems: Design and economic analysis. *Energy Procedia* 2014;61:89-92.
- [35] Al-Qaraghuli MA, Siddiqui MSAH, Shamsheer ST. Techno-economic analysis of hybrid renewable energy systems for off-grid communities. *Energy* 2013;63:282-294.
- [36] Ali MZBH. Off-grid hybrid solar photovoltaic-wind power systems: A case study of a remote area. *Energy* 2015;83:1-11.
- [37] Beaudoin JMR. Techno-economic feasibility of solar energy systems in a variety of applications. *Renew Sustain Energy Rev* 2011;15:2990-3002.
- [38] Jain NK. Economic and environmental feasibility of rooftop solar photovoltaic systems in India. *Energy* 2016;98:1001-1008.
- [39] Anitha DD, Kumar SR. Optimization and simulation of hybrid renewable energy systems. *Int J Energy Environ Eng* 2014;5:102-112.
- [40] Lee DK. Assessment and optimization of hybrid power systems for rural electrification. *Renew Energy* 2008;33:1375-1383.
- [41] Wang W, Zhang C, Li X, Zhang H. Performance evaluation of solar photovoltaic systems using machine learning techniques. *Environ Res Commun* 2024;6:025002.
- [42] Yadav A, Pathak Y. A review on optimization of hybrid renewable energy systems using machine learning algorithms. In: Sharma D, Bhardwaj A, Singhal PK, editors. *Proc Int Conf Smart Syst Adv Comput* 2024:381-393.
- [43] Ahmad A, Nazir R, Shah SMA. Comparative analysis of rooftop PV systems under different climatic conditions using PVsyst. *ISWA J* 2024;15:200389.
- [44] Ünal Y, Öztürk HH. Energy and exergy analysis of a solar-assisted heat pump system for greenhouse heating. *J Therm Eng* 2024;10:1297575.



Research Article

Parametric study and performance evaluation of a multi-layered twin pcm-based vertical thermal energy storage system

Amit Kumar GHOSH^{1,*}, Pabitra HALDER¹

¹Indian Institute of Engineering Science and Technology, Shibpur, 711103, India

ARTICLE INFO

Article history

Received: 07 March 2024

Revised: 28 June 2024

Accepted: 01 July 2024

Keywords:

Alternating Arrangement; Multi-layer; Phase Change Materials (PCMs); Thermal Energy Storage (TES); Twin-PCM

ABSTRACT

This study illustrates an ingenious approach to thermal energy storage using multi-layer twin-phase change materials (PCMs) alternately arranged inside a vertical shell and tube energy storage system. A complex conjugate heat transfer and fluid flow problem is numerically solved, and the effects of different parameters on energy storage performance are evaluated. This study highlights the impact of the number of layers, radius ratio, and flow Reynolds number in the melting and solidification process of PCMs. It also includes the comparison between multi-layer twin PCMs over single-layer individual PCM. The melting and solidification rate improves approximately by 44.3% and 19%, respectively, as the number of layers (n) increases from 2 to 8. Additionally, an 8-layer twin PCM-based system accelerates energy storage and retrieval efficacy, attaining a 33.2% enhancement in stored energy and a 5.3% increase in retrieved energy at a fixed time compared to a 2-layer configuration. It is also observed that in the 8-layer-PCM-1&2 alternating arrangement, the melting rate improved by about 33.65% and 55.84% compared to the 1-layer-PCM-1 and 1-layer-PCM-2 respectively. Similarly, the solidification rate is also promoted in the multi-layer system as compared to the single-layer system. Moreover, it is evident that a lower R/r ratio ($R/r = 2$) significantly decreases both melting and solidification times by about 78.6% and 89%, respectively, compared to a higher ratio ($R/r = 4$). Similarly, an increase in the HTF flow Reynolds number correlates with reductions in melting and solidification times by approximately 5.8% and 1%, respectively. This study also facilitates the expedited advancement of both the melting and solidification processes of PCMs with distinct physical justifications that can enable a clear understanding of the improvement of thermal energy storage systems.

Cite this article as: Ghosh AK, Halder P. Parametric study and performance evaluation of a multi-layered twin pcm-based vertical thermal energy storage system. JTherEng2025;11(5):1520–1539.

INTRODUCTION

Energy is the cornerstone of the entire world as it meets all societal and industrial demands. We are now in a stage

where all non-renewable energy resources are almost depleted level. The motivation for the research is going towards the shifting of non-renewable resources to renewable resources. The utilization of some natural resources

*Corresponding author.

*E-mail address: 2021amp002.amit@students.iiests.ac.in

This paper was recommended for publication in revised form by Editor-in-Chief Ahmet Selim Dalkılıç



is less effective because it is sporadic. This sporadicity can be reduced by the utilization of an energy storage unit. The storing medium in this unit is a phase change material (PCM) that is kept inside a sealed, insulated container that can be charged and discharged through a diathermal wall. There are numerous applications, including but not restricted to the following: Latent Heat Thermal Energy Storage (LHTES) [1–4], solar power plants [5,6], solar collectors [7–11], air conditioning and ventilation [12–15], cooling of electronic equipment [16–20], heat recovery thermal management [21–23], and spacecraft [24,25].

In the realm of efficient thermal energy storage, the meticulous selection of phase change materials (PCMs) stands as an important factor, particularly in applications where precise temperature regulation and optimal energy management are imperative. Agyenim et al. [26] conducted a comprehensive review that emphasized the importance of PCM selection in particular applications. In the low to medium temperature range, RT-42 and RT-50 are common types of PCM that are utilized in thermal storage applications [27]. Chaabane et al. [28] investigated the thermal performance of an Integrated Collector Storage Solar Water Heater (ICSSWH) with phase change materials (PCM), such as RT42-graphite. Their findings show that RT42-graphite greatly minimizes night thermal losses, making it a suitable choice for applications for maintaining heat retention overnight is crucial to enhancing overall system performance and efficiency. Cano et al. [29] investigated a thermal storage system based on phase change materials (PCMs) in a homemade heat exchanger. Among the PCMs studied, RT-42 exhibited the highest energy accumulation, indicating higher thermal storage performance

and efficient energy transmission. Elbahjaoui and Qarnia [30] examined the thermal behavior of a latent heat storage system (LHSU) integrated with a flat-plate solar collector incorporating various phase change materials (PCMs), such as RT-42, RT-50, and RT-60. The results showed that the quantity of latent heat stored during the charging process was around 19.3 MJ for RT-42, 16.54 MJ for RT-50, and 12.79 MJ for RT-60, with RT-42 having the largest latent heat storage capacity of the three. Şimşek and Demirci [31] compared the melting times of RT-35 and RT-42 in a double-tube concentric heat exchanger using CFD. Their studies showed a significant reduction in melting time, with RT-42 experiencing up to an 86.33% decrease compared to the finless model, surpassing RT-35, which reduced melting time by up to 85%. There is a lot of research that has shown the use of PCM in various thermal energy storage systems. A few studies are listed in Table 1 to show the detailed applications based on different PCMs.

The advent of nano-enhanced phase change materials (nePCMs) has sparked a significant improvement in thermal management and storage systems. In the comprehensive review, Mebarek-Oudina and Chabani [38] highlighted how the integration of nanoparticles into traditional PCMs enhances thermal conductivity while reducing specific heat capacity. This combination produces high thermal diffusivity, which enables nePCMs to store and release heat more effectively. These enhancements considerably increase the performance of thermal storage systems [39], HVAC systems [40], solar systems [41], electrical components [42], and batteries [43]. Several studies that elucidated the advancements and applications of nePCMs in various energy storage systems provide valuable insights into

Table 1. Detail review of PCM used for the particular application

Ref.	PCM	Applications	Contributions
Trp [32]	RT-30	Vertical Shell and Tube LHTES	Experimental results support numerical desired outcomes by demonstrating non-isothermal paraffin melting and isothermal solidification.
Zhang et al. [33]	RT-35	Vertical Shell and Tube LHTES	Helical fins improve PCM melting in LHTES, resulting in superior thermal performance over traditional designs.
Feng et al. [34]	RT-40	Evacuated solar tube collector	RT-40 considerably improves solar thermal storage, maximizing efficiency and reliability in solar energy systems. remarkably systems that use PCM with a latent heat of 160 kJ/kg perform superior.
Alshihmani et al. [35]	RT-35 HC RT-42	PCB Cooling	The impact of RT-42 and RT-35 HC PCM on PCB cooling, with RT-42 surpassing RT-35 in terms of thermal performance.
Refaey et al. [36]	RT-31 RT-35 RT-42	Photovoltaic panels	PCM incorporation improves solar panel cooling with RT-42, which uses triangular fins to achieve a 24% temperature decrease while increasing efficiency by 7-8.4%.
Xu et al. [37]	RT-42 RT-50 RT-60	Horizontal Shell and Tube LHTES	Optimizes triplex-layer PCM melting in LHTES unit using RT-42, RT-50, and RT60 PCMs, considerably lowering melting time by up to 71%.

Table 2. Review of nePCM used for different applications

Ref.	Applications	Contributions
Algarni et al. [44]	Tube solar collector	Integrating 0.33 wt% of copper-enhanced phase change materials to an evacuated tube solar collector (ETSC) improves its efficiency by 32%.
Bashirpour-Bonab [45]	Multi-tube energy storage	Dispersing 3-7% copper-oxide nanoparticles (CuO) in paraffin RT82 phase-change material (PCM) reduces melting time by 8.07-22.18%.
Elarem et al. [46]	Tube solar collector	The addition of 1% Cu nanoparticles to paraffin in an ETSC with fins boosts HTF temperature by 2 K and optimizes PCM melting.
Rothan [47]	Cold storage unit	Adding nano-sized particles to water in a finned wavy tank reduced the freezing time by 19.93% for 30-40 nm particles while increasing the freezing rate by 41.16%.
Bahrami et al.[15]	Air conditioning ventilating system	Using nano-enhanced PCM wallboards in a room minimizes air-conditioning energy usage by 7.4%, improves energy storage by 4.37%, and reduces fluctuations in temperature by 52%.
Zahid et al. [48]	Electronics cooling	NePCM (RT-54HC/Al ₂ O ₃) containing 0.25 wt% Al ₂ O ₃ nanoparticles reduces base temperature by 36.2%, hence improving electronic device cooling.
Swamy et al. [49]	Battery thermal management	Adding 20% CuO and Al ₂ O ₃ nanoparticles to PCM raises the melting fraction substantially, increasing thermal uniformity.

enhancing thermal management and storage performance, as summarized in Table 2.

The heat energy storage and retrieval are carried out considering the melting and solidification of PCM in a cycle. PCM is melted and stored in the thermal energy followed by the retrieval of energy through the solidification processes. The shell-and-tube design stands as a widely preferred configuration employed in latent heat thermal energy storage (LHTES) systems, where PCMs are utilized to absorb and release heat energy. The storage performance of shell-and-tube LHTES systems can be improved through various design modifications. Cao and Faghri [50] studied the effect of geometrical factors viz. Length to diameter ratio (L/D), radius ratio (r_o/D) considering a horizontal type storage system. This numerical study demonstrated a slight decrease in energy storage density with the L/D ratio and a significant reduction with r_o/D . Ismail and Melo [51] investigated a vertical energy storage unit and established a correlation between heat transfer rate and solidification time. The findings revealed that melting time increased with an increase in H/r_o and R_o/r_o . Trp et al. [52] explored the impact of L/D and r_o/D on both melting and solidification considering a vertical storage. The findings showed that the selection of geometrical parameters had a substantial impact on the rate of heat transfer and the amount of energy stored or released. Tao et al. [53] studied on horizontal LHTES system that shows the time taken for melting increased by 16.3% when the outer radius expanded from 24.0 mm to 28.0 mm, whereas it decreased by 45.4% when the velocity increased within the range of 10 - 20 m/s. The effect of eccentricity concerning the inner and outer tube was explored by Zheng et al. [54] to analyze melting and melting-solidifying characteristics. It also highlighted the

consequences of the Rayleigh number on the optimum eccentricities. Shen et al. [55] studied two different configurations altering the shell radius while maintaining a fixed HTF-tube radius and varying the HTF-tube radius while keeping the outer PCM shell radius constant. The study revealed that the density of energy storage/retrieval was optimal at a radius ratio of 5.

It is essential to take into account the LHTES system's performance with different geometrical configurations. Vyshak et al. [56] compared the melting rates of PCM filled in cylindrical, rectangular, and cylindrical shell containers with equal heat transfer area and volume. It was observed that the melting duration was shortest for PCM stored in a cylindrical shell container, with a noticeable impact of the PCM mass increase on its geometry. Akgun et al. [57] evaluated the results of conical and non-conical storage in terms of melting time and came to the conclusion that conical storage reduced melting time by 20%. A comparative study between conical and cylindrical shell-type storage systems was conducted by Seddegh et al. [58]. It revealed that while the solidification time was nearly the same, the conical tube-shaped unit melted 12% faster than its cylindrical counterpart. Alaraji et al. [59] investigated the melting and solidification characteristics of RT-26 in an LHTES system with an elliptic inner tube. It was revealed that the time for complete melting deteriorated with an increment of the eccentricity of the inner tube. An annular tube comprising vertical and horizontal heat storage was the subject of an experimental study by Dukhan et al. [60]. This study shows how the orientation of storage units has a major effect on the charging of PCM. This was concluded that vertical-type heat exchangers can store more energy than horizontal. Zağlanmıř et al. [61] investigated numerically

a thermal storage system based on several pipe sequences developed for body-pipe-type heat exchanger models. The findings reveal that the duration of melting deteriorates with the increment of HTF inlet temperature. It was also observed that the sequence of the tube affects the melting rate and stored energy.

Another popular method for improving heat transfer is to place extended additional surfaces like fins inside the PCM and a heat source/sink. The main advantage of using a fin is it increases the surface area that resulting in uniform heat transfer which expedites the phase transition process. Paria et al. [62] experimented on a horizontal LHTES system with annular fins of 24 and 48 numbers. They showed that charging time decreased by 58% and 76% in two different cases. Joshi and Rathod [63] studied LHTES systems using fin. It was obtained that the optimal location and size of the fin reduce the melting time. They have used a fin design that enhances the thermal performance than conventional fin configuration at about 4.38%. Nakhchi and Esfahani [64] increased the efficacy of a vertical LHTES system by adopting a new design of upward and downward-stepped fins. It was observed that all stepped fin configurations outperformed straight fins in terms of thermal performance. To assess the impact of the length and fin number that are positioned radially on the functionality of a horizontally oriented cylindrical LHTES system, Nie et al. [65] showed a numerical study using a 2D axisymmetric domain. The melting was found to be enhanced more quickly by the straight fins than by the angled, lower, and upper fin configurations. Nóbrega et al. [66] explored the impacts of axial fins that are positioned vertically along a central pipe in a cylindrical LHTES system. It was observed that the entire solidification time decreased as the number and width of the fins increased. Tiari et al. [67] studied the numerical investigation of the LHTES system using six different annular fin configurations. The twenty-fin arrangement with a constant length was proven to be the most effective arrangement in terms of both melting and solidification time, which reduced the overall time by approximately 76.3%.

In recent years, the multi-layered arrangement is one of the novel approaches in which single or multiple PCMs are arranged in layers. Brousseau and Lacroix [68] studied the thermal performance in a multi-layered system that aimed to reduce the energy load during a peak demand period. The output heat load and the total energy stored were proposed to be correlated as a function of the design factors and operating conditions. Li et al. [69] investigated an LHTES unit in which three PCM were considered. They compared the melting rate of three PCMs and suggested the optimum length for three PCMs. Ezra et al. [70] examined different rows of PCMs arranged in cascaded manners. It was determined for the number of different materials what range of melting temperatures would result in the fastest melting. Beemkumar et al. [71] experimented to explore the heat transfer of cascaded energy storage considering

multiple types of fins. It was found that the copper-encapsulated annular finned balls with D-mannitol as the PCM had a faster rate of energy transfer both during charging and discharging. Prasad et al. [72] conducted a comparative study on three PCM-based cascaded LHTES units. The comparison of the combined model with the model solely based on conduction demonstrated that the pure conduction model underestimates the rate of heat transfer between the HTF and PCM at lower flow velocities of the HTF, where free convection significantly influences the phase change phenomena. Kareem et al. [73] investigated a multi-slab latent heat storage system in which nine slabs are filled with RT-25 and RT-21. The results demonstrated that the height of the channel, the height of the PCM slab, the melting temperature of PCM, and the length-to-width ratio strongly affect the melting and solidification performance.

This work introduces an advanced approach to thermal energy storage that uses a multi-layer twin PCM-based vertical thermal energy storage system with an alternating PCM arrangement. As per the authors' knowledge, previous studies have mainly explored the use of uniform PCM layers. This study investigates the effectiveness of using two PCMs with different melting temperatures arranged alternately, a configuration that has not yet been examined. Its unique configuration is intended to increase both energy storage density and retrieval speed. The number of layers, radius ratio, and Reynolds numbers of the HTF flow are investigated to determine their impact on system performance. By comparing single and multi-layer configurations, the study elucidates PCM melting and solidification processes and their impact on energy storage performances. The results demonstrate that the alternating PCM arrangement significantly improves the melting and solidification performances, leading to more efficient energy storage and faster energy recovery. These findings have promising implications for a wide range of thermal storage applications, including renewable energy systems, backup power solutions, thermal management in electronics, building temperature regulation, industrial waste heat recovery, and transportation systems like electric vehicles and aerospace.

PHYSICAL MODEL

A two-dimensional vertical axisymmetric model as depicted in Figure 1, has been considered in the present computation. The domain consists of a tube surrounded by a shell of multi-layered twin-phase change materials arranged in alternating fashion. Each PCM layer is separated by a thin thermally coupled wall. Each layer is divided in such a way that both PCMs have the same mass. The outer surface of the shell and the extended tube are maintained in thermally insulated condition. The tube radius (r), shell length (L_s) and tube length (L_t) are considered as 12.5 mm, 500 mm, and 700 mm respectively. The radius of the shell (R) has been varied based on the radius ratio (R/r) 2, 3, and 4. The number of layers (n) of PCM ranges

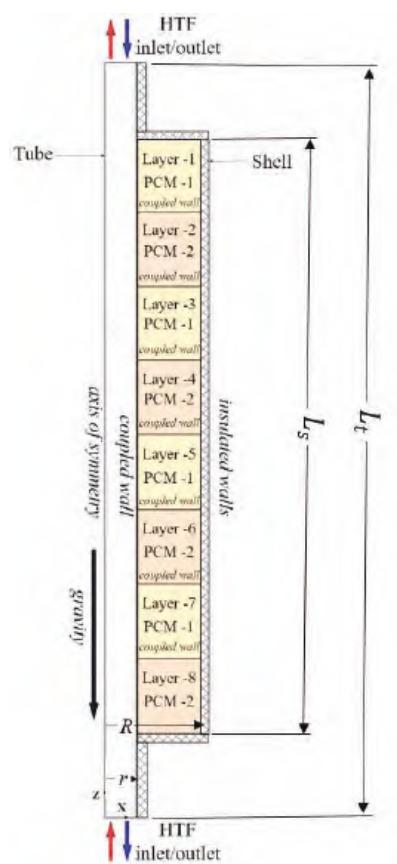
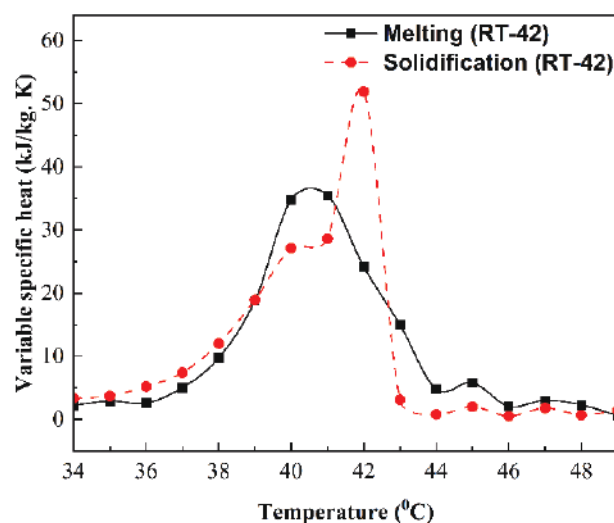


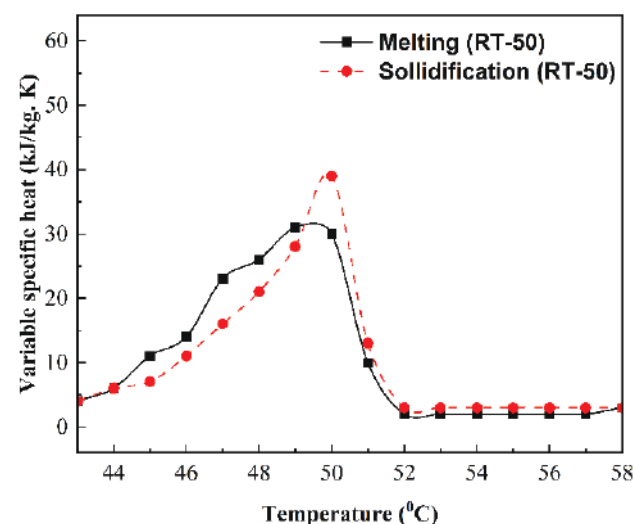
Figure 1. Schematic diagram of the physical model.

between 2, 4, and 8, with each layer having equal length. There are three velocity inlet conditions are selected based on Reynolds number (Re) of 5000, 10000, and 15000.

During the melting of PCMs, hot heat transfer fluid at 80°C flows in the upward direction and charges the PCMs. The initial temperature of PCMs is kept at 25°C . Whereas in solidification, the reverse process is executed. During the solidification, the cold HTF at 25°C flows in the downward direction and discharges the PCMs. The thermophysical



(a)



(b)

Figure 2. Variable specific heat with the temperature of (a) PCM-1 (RT-42) and (b) PCM-2 (RT-50).

properties of PCM-1, PCM-2, and HTF are tabularized in Table 3.

Table 3. Thermo-physical properties of PCM-1, PCM-2, and HTF [60] [74]

Description	PCM-1 (RT-42)	PCM-2 (RT-50)	HTF (water)
Density (solid/liquid), (kg/m^3)	880/760	880/760	998.2
Specific heat, ($\text{kJ}/\text{kg K}$)	variable (Fig. 2)	variable (Fig. 2)	4.182
Thermal Conductivity, ($\text{W}/\text{m K}$)	0.2	0.2	0.6
Dynamic viscosity, ($\text{kg}/\text{m s}$)	0.0030	0.0038	0.001003
Thermal expansion coefficient, ($1/\text{K}$)	0.0008	0.0006	-
Latent heat of fusion, (kJ/kg)	165	160	335
Solidous temperature, ($^\circ\text{C}$)	38	45	0
Liquidous temperature, ($^\circ\text{C}$)	42	50	0

MATHEMATICAL MODEL AND BOUNDARY CONDITION

A 2D model is considered to examine the transient heat transfer, fluid flow, and melting and solidification processes. For the simplicity of present numerical computation, a few assumptions have been made: (i) HTF and liquid PCMs are incompressible, (ii) viscous dissipation and radiation heat exchange are neglected, (iii) thickness of solid boundaries are neglected, (iv) PCMs volume expansions are neglected. The coupled non-linear differential equation has been solved for HTF as well as PCMs [55].

Mathematical Modeling for HTF

The conservation of mass is stated by

$$\rho_{HTF} (\nabla \cdot \vec{V}_{HTF}) = 0 \quad (1)$$

The conservation of momentum and energy is expressed below:

$$\rho_{HTF} \frac{\partial}{\partial t} (\vec{V}_{HTF}) + \rho_{HTF} \nabla \cdot (\vec{V}_{HTF} \vec{V}_{HTF}) = \nabla \cdot ((\mu_{HTF} + \mu_{t,HTF}) \nabla \vec{V}_{HTF}) + \rho_{HTF} g - \nabla p_{HTF} \quad (2)$$

$$\frac{\partial (\rho_{HTF} C_{p,HTF} T_{HTF})}{\partial t} + \nabla \cdot (\vec{V}_{HTF} (\rho_{HTF} C_{p,HTF} T_{HTF})) = \nabla \cdot ((K_{HTF} + K_{t,HTF}) \nabla T_{HTF}) \quad (3)$$

Where, $\mu_{t,HTF}$ and $K_{t,HTF}$ are viscosity and thermal conductivity due to turbulence are as follows:

$$\mu_{t,HTF} = C_{\mu} \rho_{HTF} \frac{k^2}{\varepsilon} \quad (4)$$

$$K_{t,HTF} = \frac{\mu_{t,HTF} C_{p,HTF}}{Pr_{t,HTF}} \quad (5)$$

Here, k and ε are the turbulent K.E and rate of dissipation respectively. k and ε can be evaluated considering the standard k - ε turbulent modeling which is governed by the following equation:

$$\frac{\partial (\rho_{HTF} k)}{\partial t} + \nabla \cdot (\rho_{HTF} \vec{V}_{HTF} k) = \nabla \cdot \left(\mu_{HTF} + \frac{\mu_{t,HTF}}{\sigma_k} \nabla k \right) + G_{k,HTF} + G_{b,HTF} - \rho_{HTF} \varepsilon \quad (6)$$

$$\frac{\partial (\rho_{HTF} \varepsilon)}{\partial t} + \nabla \cdot (\rho_{HTF} \vec{V}_{HTF} \varepsilon) = \nabla \cdot \left(\mu_{HTF} + \frac{\mu_{t,HTF}}{\sigma_\varepsilon} \nabla \varepsilon \right) + \frac{\varepsilon}{k} (C_1 G_{k,HTF} - C_2 \rho_{HTF} \varepsilon) \quad (7)$$

Where, $G_{k,HTF}$, $G_{b,HTF}$ is turbulent K.E. generation due to velocity gradient and buoyancy respectively. The values of C_{μ} , C_1 , C_2 , $Pr_{t,HTF}$, σ_k , σ_ε are 0.09, 1.44, 1.92, 0.85, 1.0, and 1.3 respectively.

Mathematical Modeling For Pcm

Enthalpy-porosity method is employed to model the solid-liquid phase transition process. The model consists of

the conservation of mass, momentum, and energy which are as follows:

$$\rho_{PCM} (\nabla \cdot \vec{V}_{PCM}) = 0 \quad (8)$$

$$\rho_{PCM} \left(\frac{\partial}{\partial t} (\vec{V}_{PCM}) + \nabla \cdot (\vec{V}_{PCM} \vec{V}_{PCM}) \right) = \nabla \cdot (\mu_{PCM} \nabla (\vec{V}_{PCM})) + \rho_{PCM} (1 - \beta(T - T_{ref})) g - \nabla p_{PCM} + \vec{S}_y \quad (9)$$

$$\frac{\partial (\rho_{PCM} h_{PCM})}{\partial t} + \nabla \cdot (\rho_{PCM} \vec{V}_{PCM} h_{PCM}) = \nabla \cdot (K_{PCM} \nabla (T_{PCM})) \quad (10)$$

Here, μ_{PCM} and p_{PCM} , $\vec{V}_{PCM}$ are PCM's dynamic viscosity, static pressure, and velocity respectively. The variation of density in the buoyant force part in the momentum equation is taken as the Boussinesq approximation to account for the free convection flow phenomenon. The extra source term $\vec{S}_y$ is the momentum sink term which is represented as follows

$$\vec{S}_y = C_{mush} \frac{(1 - \gamma)^2}{\gamma^3 + \delta} \vec{V}_{PCM} \quad (11)$$

$$\gamma = \begin{cases} 0, & T \leq T_{PCM,S} \\ \frac{T - T_{PCM,S}}{T_{PCM,L} - T_{PCM,S}}, & T_{PCM,S} < T < T_{PCM,L} \\ 1, & T \geq T_{PCM,L} \end{cases} \quad (12)$$

Where, C_{mush} is a constant, called the mushy zone constant. In the current study, C_{mush} is kept as a default value of $10^5 \text{ kg/m}^3 \cdot \text{s}$. Here, δ is a small constant considered to prevent zero division when the PCM is completely solid and the melting process is starting, i.e., liquid fraction, $\gamma = 0$.

The volumetric enthalpy, h_p the energy equation is stated below:

$$h_{PCM} = h_{ref} + \int_{T_{ref}}^T C_{p,v} dT + \gamma L \quad (13)$$

Where, h_{ref} is the reference PCM enthalpy estimated at T_{ref} and L is the latent heat of the fusion of PCM. The energy stored and recovered during charging and discharging is evaluated by

$$Q_T = \frac{Q(t)}{m_{PCM}} \quad (14)$$

$Q(t)$ and m_{PCM} are the energy stored at time instance t and the mass of PCM respectively. $Q(t) = H - H_{Ref}$, H_{Ref} is the reference enthalpy at the reference temperature, and H can be calculated as follows:

$$H = \int_0^V h_{PCM} \rho_{PCM} dV \quad (15)$$

Mathematical Modeling for Thermally Coupled Walls

$$\rho_{wall} \frac{\partial}{\partial t} (h_{wall}) = K_{wall} \nabla \cdot (\nabla T_{wall}) \quad (16)$$

Where, h_{wall} and K_{wall} are the enthalpy and thermal conductivity of the thermally coupled wall. h_{wall} is calculated by the following integral.

$$h_{wall} = \int_{T_{ref}}^{T_{wall}} C_{p,wall} dT \quad (17)$$

Boundary and Initial Conditions

Boundary and initial conditions are selected as follows:

HTF inlet: (i) Velocity inlet boundary condition, $V_x = 0$,

$V_z = V_{inlet}$,

(ii) $T_{HTF} = T_{inlet} = 80^\circ\text{C} / 25^\circ\text{C}$ (melting/solidification)

HTF outlet: Outflow boundary condition

PCM: $T_{PCM} = T_{int} = 25^\circ\text{C} / 80^\circ\text{C}$ (melting/solidification)

Thermally coupled wall : (i) $V_x = 0$, $V_z = 0$ (ii) wall adjacent to tube, $K_{wall} \frac{\partial T_{wall}}{\partial x} = K_{HTF} \frac{\partial T_{HTF}}{\partial x}$, (iii) wall adjacent to PCM, $K_{wall} \frac{\partial T_{wall}}{\partial x} = K_{PCM} \frac{\partial T_{PCM}}{\partial x}$, (iv) walls in-between layers,

$K_{wall} \frac{\partial T_{wall}}{\partial z} = K_{PCM} \frac{\partial T_{PCM}}{\partial z}$

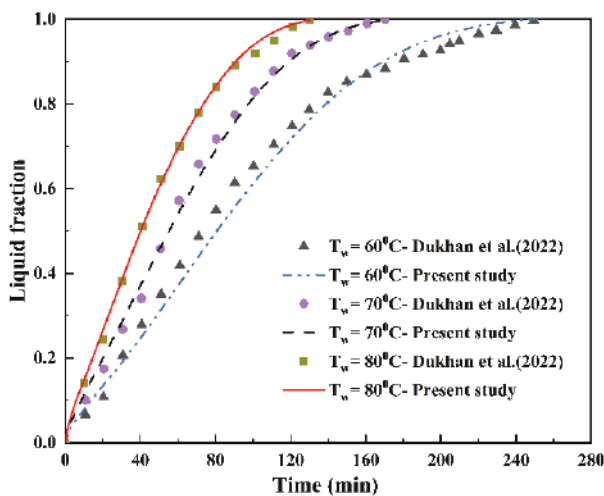
Insulated walls: (i) $V_x = 0$, $V_z = 0$ (ii) $\frac{\partial T}{\partial x} = 0$, $\frac{\partial T}{\partial z} = 0$

NUMERICAL METHODOLOGY AND VALIDATION

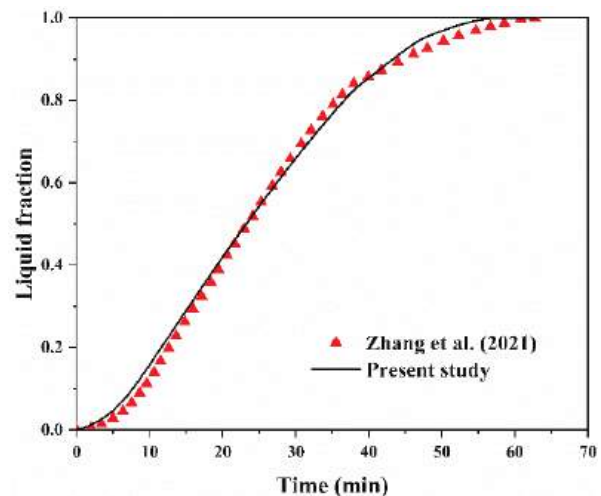
All the non-linear governing equations are numerically computed in the Finite Volume Method (FVM) based commercially available software package ANSYS Fluent 2019 R3. The advantages and flow chart of FVM are elucidated in the appendix section. The solid-liquid phase change process is modeled using the enthalpy-porosity technique [75]. The solver adopts an enthalpy-porosity formulation, considering the liquid-solid mushy zone as a porous medium

with porosity equal to the liquid fraction, and includes a momentum sink term in the momentum equations to account for pressure drop caused by solid material. The coupled algorithm is considered to combine the pressure correction and velocity for solving continuity and pressure field. Pressure correction is accomplished by the PRESTO scheme. The Second-Order Upwind discretization is selected to attain higher-order accuracy at cell face values in momentum, energy, turbulent kinetic energy, and dissipation rate equations. The gradient between two cell values is calculated by the Least Squares Cell-Based scheme. The transient formulations are accomplished by the First Order Implicit scheme. The under-relaxation factors for momentum, energy, liquid fraction, turbulent kinetic energy, and turbulent dissipation rate are 0.75, 1, 0.9, 0.8, and 0.8 respectively. The criteria for convergence are set at 10^{-6} for continuity, momentum, turbulent K.E, and turbulent dissipation rate whereas 10^{-9} is selected for the energy equation. The identical solution methodology is employed for both the melting and solidification processes due to their inherent inversion. The computation facility used for simulation is Intel® Xeon® CPU E5-2650 v3 @ 2.30GHz.

For checking the consistency of the present numerical model, computational outcomes are verified with experimental results. Figure 3(a), illustrates the variation of the liquid fraction with flow time at three different HTF inlet temperatures which is experimented with by Dukhan et al. [60]. The present results show a very good match with the experimental results. This validation model is replicated in all the simulations to achieve good accuracy and closer results to the experimental. A secondary validation is performed to ensure the accuracy of the present FVM code by comparing it with previously reported numerical findings. This verification is accomplished by comparing the current



(a)



(b)

Figure 3. Verification of present numerical results with (a) experimental data, (b) numerical simulation data.

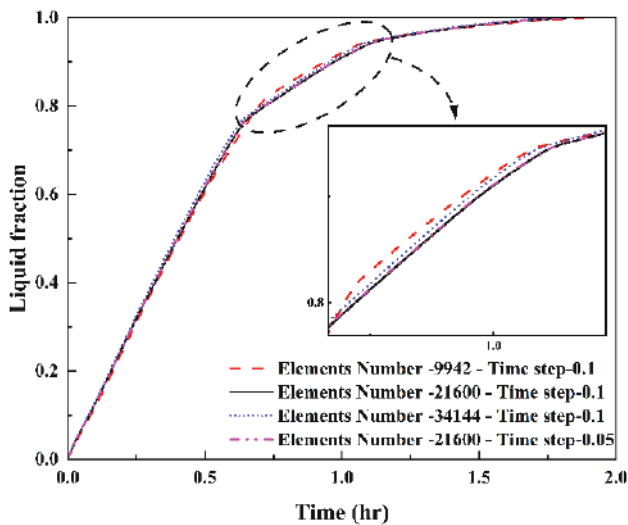


Figure 4. Grid and time independence test.

numerical predictions with the numerical results reported by Zhang et al. [76] in Figure 3(b).

GRID AND TIME INDEPENDENCE TEST

Grid independence test is performed by considering different element numbers as depicted in Figure 4. It is understood that results are almost constant beyond 21600 number elements. We fixed the number of elements to 21600 and checked for the time independence test. It is seen that beyond the elements 21600 and time step of 0.1s there is very negligible change in the liquid fraction. So, in all the simulations the number of elements is considered as 21600, and the time step of 0.1s. It is also to be noted here that the same number of elements and the time step are used in the validation study for achieving good accuracy with the experimental and numerical.

RESULTS AND DISCUSSION

During Melting of PCM

Figure 5a depicts the temporal variation of the liquid fraction with the number of layers of PCMs during the melting process. As the number of layers (n) varies from 2 to 8, the overall liquid fraction has a significant change. The time needed for fully melting is shortened by 44.3% if the number of layers (n) is increased from 2 to 8. When the number of layers is 2, PCM-1 and PCM-2 are separated by a single thermally coupled wall. Initially, both layers have an initial temperature lower than the melting point of both PCM. The melting process begins near the tube wall surface, spreading into PCM-1 and PCM-2. Figure 6 presents the contours of the liquid fraction and temperature. It is seen that PCM-1 melts faster than PCM-2. This is because the PCM-1 with a lower melting point will start to melt at

a lower temperature and use less heat energy to reach that temperature. As soon as the PCM-1 begins to melt, it may absorb heat energy more quickly and distribute it faster throughout its entire volume than the PCM-2. In Figures 7a and 7b, it can be observed that a clockwise vortex is formed in the melted pool of both PCM-1 and PCM-2. This represents the natural convection currents driven by temperature gradients within both PCMs. As the PCMs absorb heat, they change from solid to liquid, creating regions of varying density. The melted PCMs near the heat source become less dense and rise, while the cooler, denser liquid descends, resulting in convection currents. The intensity of the vortex near the melt front is greater in PCM-1 than in PCM-2. This indicates that PCM-1 has stronger convection currents than PCM-2, which increases heat transfer rates and accelerates the melting process. The stronger nature of the vortex in PCM-1 facilitates more efficient mixing and heat distribution throughout the melted zone, resulting in a faster-moving melt front. From the temperature contours, it is also observed that there exists a temperature gradient between layers that derives heat flow through the thermally coupled walls, both conductively and convectively. This promotes consistent heat distribution and speeds up the overall melting rate, emphasizing the synergistic effect of temperature gradients and thermally coupled walls in multi-layered PCM systems. It can also be seen from Figure 7d that the vortex near the melt front in PCM-1 disappears, but in Figure 7a, the vortex near the melt front in PCM-1 is evident. It is also obvious that more melting volume is seen inside the melt pool in the case of 8 layers than 2 layers. More melts mean a larger volume of fluid, which may be enough to disrupt the vortex and create a more stable flow pattern. When the number of layers is 8, PCM-1 and PCM-2 are alternately arranged and each of the layers is separated by a thermally coupled wall, a temperature gradient in between layers is created that leads to differences in density within the PCMs, driving natural convection currents where warmer, less dense fluid rises and cooler, denser fluid sinks. However, the presence of thermally coupled walls and alternating PCM materials introduces a more complex flow pattern. Specifically, at the junctions between different PCM layers, the interaction of opposing thermal and fluid flow fields from each layer leads to the formation of counter-rotating vortices. From Figure 7b and 7c, it can be seen those temperature-induced counter-rotating vortices formed near the conjunction of two PCM layers. Those vortices create a low-pressure region at the center. This can cause the melted PCMs to be drawn in from the surrounding region, increasing fluid mixing and promoting convective heat circulation. It can be noticed that the strength and intensity of the vortices increase as the number of layers increases. Figure 7c shows multi-counter rotating vortices formed near the conjunction between two layers. It is evident in proper mixing that accelerates the heat transfer which causes a high liquid fraction as flow time marched.

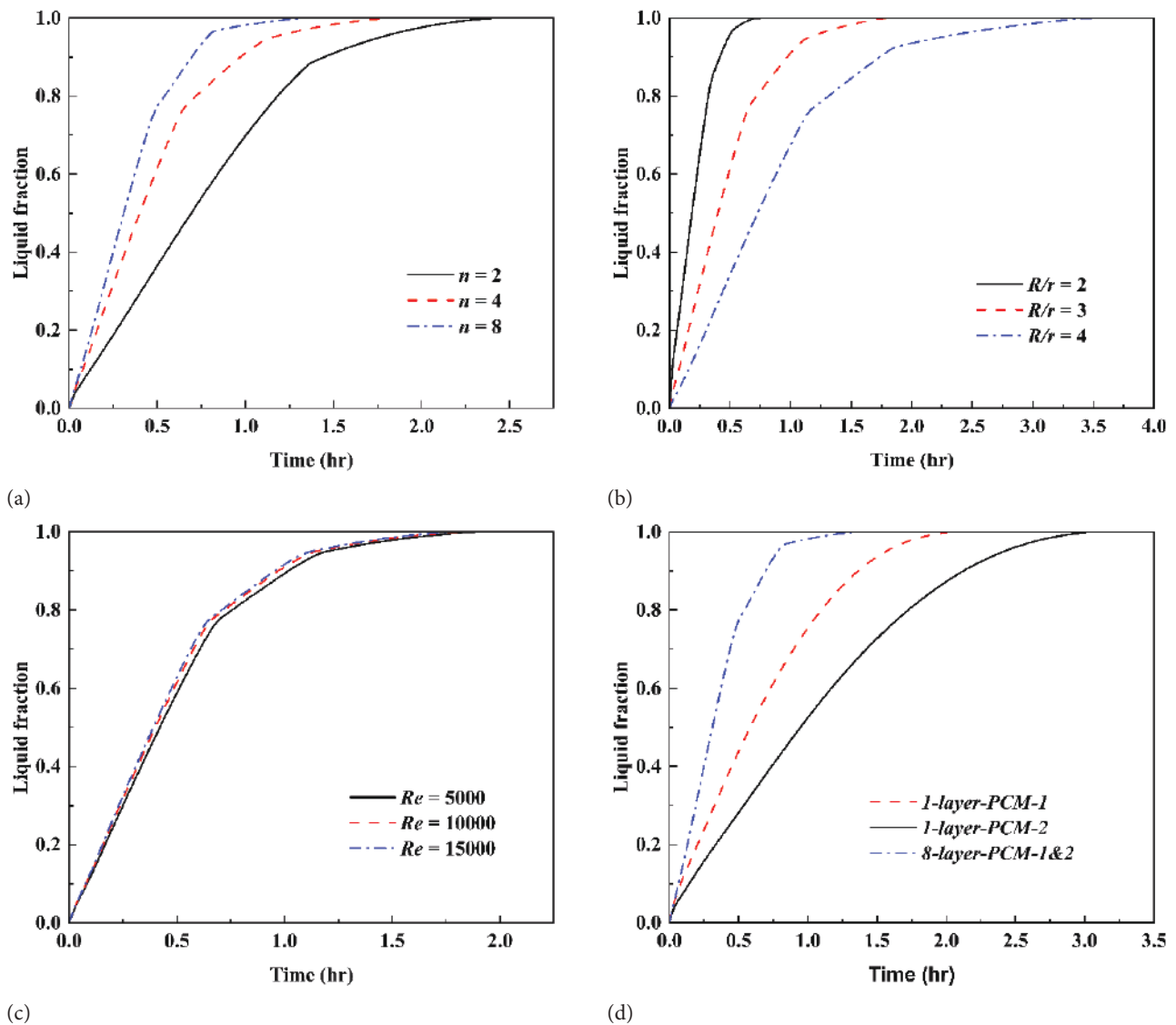


Figure 5. Temporal variation of liquid fraction with different parameters during melting of PCMs at (a) $Re = 10000$, $R/r = 3$, (b) $Re = 10000$, $n = 4$, (c) $n = 4$, $R/r = 3$, (d) $Re = 10000$, $R/r = 3$.

Figure 5b illustrates the variation of liquid fraction with the radius ratio of shell and tube. It can be observed that raising the R/r ratio and keeping the tube ratio constant leads to an increase in melting time. Specifically, lowering the R/r ratio from 4 to 2 decreases the time needed for complete melting by 78.6%. This phenomenon results from the interaction between thermal mass and thermal resistance. As the shell radius increases, so does the volume and mass of the PCMs within it, necessitating more thermal energy to reach the melting point and resulting in increased thermal inertia. This increased mass introduces more thermal resistance, decreasing total heat transfer and hindering the rate at which the PCM's average temperature rises. Furthermore, the increased shell radius necessitates heat to traverse a greater distance from the tube to the outer shell, thereby augmenting thermal resistance and prolonging the

melting duration. Conversely, a lower R/r ratio decreases the PCM volume, reducing both thermal mass and thermal resistance, thus allowing quicker heat transfer and accelerating the melting process. However, this also leads to a lower thermal storage capacity due to less material available for energy storage. Therefore, while designing a thermal energy storage system, the radius ratio of the shell to the tube must be taken into account, as it has an important impact on melting time and overall performance. To ensure a reasonable trade-off between quick melting and sufficient thermal storage capacity, balancing the R/r ratio is essential for maximizing system performance.

Figure 5c describes the variation of the liquid fraction at different HTF flow Reynolds numbers. It is seen that liquid fraction improves with HTF flow Reynolds number. Quantitatively rise in Reynolds number (Re) from

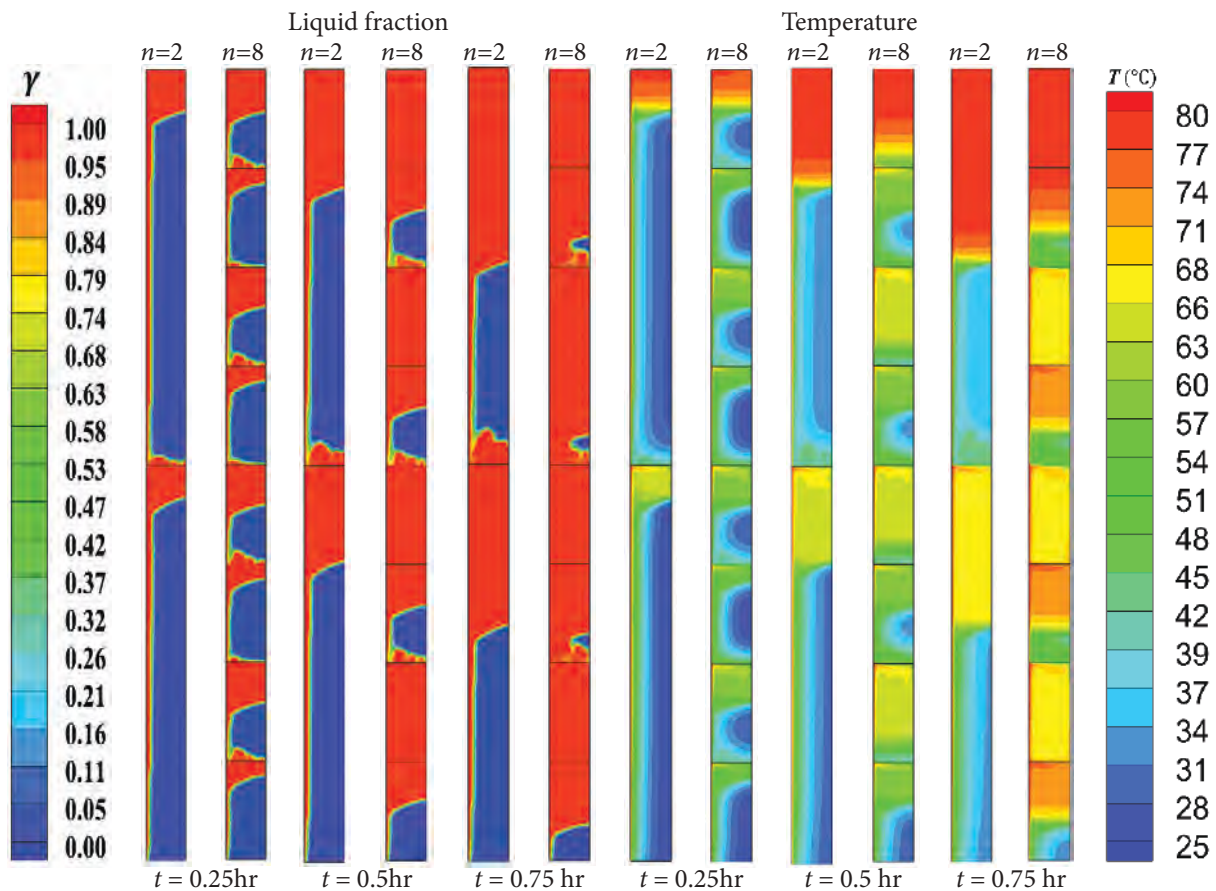


Figure 6. Contours of liquid fraction and temperature at different flow times and number of layers during melting of PCMs at $Re = 10000$, $R/r = 3$.

5000 to 15000 can reduce the melting time by about 5.8%. Convective heat transfer has been improved, which can account for the rise in the liquid fraction that occurs as the Reynolds number rises. The turbulent nature of the fluid flow that results from a rise in the Reynolds number might result in a greater heat transfer coefficient between the HTF and PCMs. The larger liquid fraction that emerges from the enhanced heat transfer coefficient can be attributed to the increased amount of heat that is transmitted from the HTF to the solid PCMs. However, the enhancement in liquid fraction with an increase in the Reynolds number is typically small because the heat is transferred from the HTF to the solid PCMs through a process called conduction. Nevertheless, compared to the rate at which heat may move through the HTF, the solid PCM normally conducts heat at a significantly slower rate. This is because PCMs are often significantly less thermally conductive than HTF, which makes them less effective in transferring heat. Hence, when PCMs are heated by the HTF, heat may accumulate inside the solid PCMs more quickly than it can be transmitted out from the heated region. This can lead to a temperature gradient within the PCMs, with the heated region having a higher temperature than the surrounding

PCMs. If the temperature of the heated region reaches the melting temperature of the PCMs, the PCMs will begin to melt. However, the rate at which the PCMs melt will be limited by the rate at which heat can be conducted away from the heated region. In other words, the solid PCMs will act as a thermal bottleneck that limits the overall melting rate. This bottleneck effect arises due to the thermal resistance encountered within solid PCMs, impeding the heat flow from hotter to cooler regions, thus delaying the phase change process. Therefore, even if the flow rate of the HTF is increased, the overall melting rate may not increase significantly if the solid PCMs act as a thermal bottleneck. In this case, the rate of heat transport through the PCMs will continue to limit the melting rate, regardless of the flow rate of the HTF.

Figure 5d explains the comparison of time variation of the liquid fraction with three different cases viz. *1-layer-PCM-1*, *1-layer-PCM-2*, and *8-layer-PCM-1&2* alternating arrangement. The finding itself provides a better understanding of the selection of the multi-layer configuration. The overall melting time is reduced in the eight-layered configuration by 33.65% and 55.84% as compared to the single-layer of PCM-1 and PCM-2 respectively. Firstly,

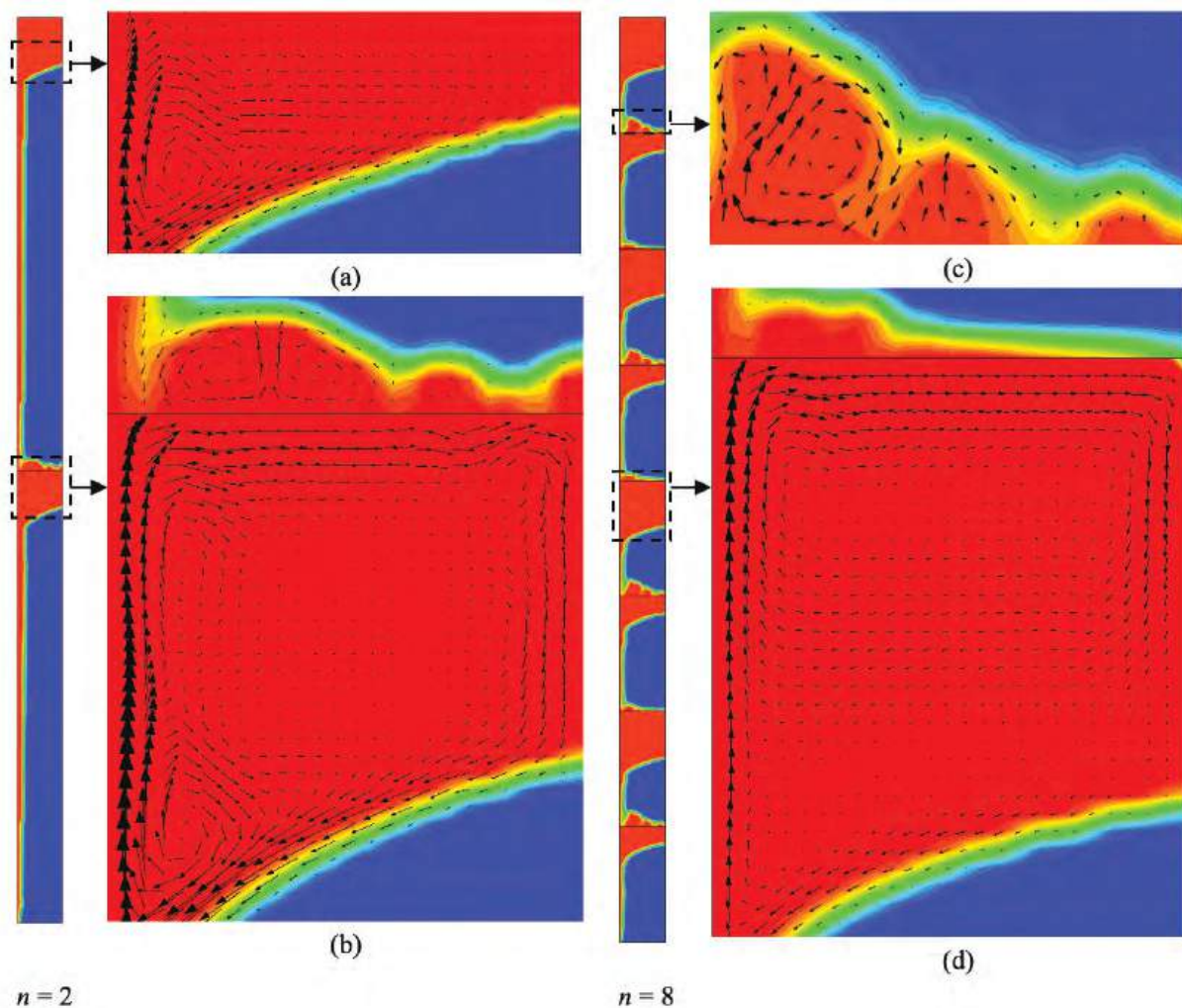


Figure 7. Velocity vector during melting of PCMs at $Re = 10000$, $R/r = 3$, $t = 0.25$ hr.

compared to single-layer PCMs, the 8-layer alternating PCM arrangement uses more layers, creating a higher surface area for heat transfer. Having more surface area can expedite heat transport and minimize the overall melting time. Secondly, in the multi-layer arrangement heat is transported not only from the HTF to the PCM but also between the layers themselves. As the HTF moves along the walls, heat is absorbed by all PCM layers, which initiates the melting process and creates a melt pool in each layer. These melt pools create temperature gradients at the boundaries of each consecutive pair of PCMs, causing inter-layer heat transfer from the bottom to the higher layer due to temperature differences. The alternating arrangement of PCM-1 and PCM-2 is very useful because their different thermophysical properties, generate large variances in heat absorption capacities. These variations provide higher temperature gradients between consecutive layers, which increases the overall heat transfer rate of the system. As heat moves from a lower layer's melt pool to the higher

layer, the temperature in the upper layer rises more quickly, facilitating faster melting. This inter-layer heat transfer process constantly generates favorable temperature gradients relative to the HTF, resulting in efficient heat distribution throughout the system. This ongoing process considerably increases the melting rate in multi-layered systems when compared to single-layer systems. As a result, the alternating PCM arrangement is superior to single-layer arrangements in terms of thermal storage performance, as it leverages the synergistic effects of different material properties to enhance heat absorption, storage, and heat transfer efficiency.

During Solidification of PCM

Figure 8a illustrates the time changes of the liquid fraction with the number of layers during the solidification process. It is observed that solidification becomes faster if the number of layers raises (n) from 2 to 8. It is seen that near the tube wall surface where the solidification process begins, spreading into PCM-1 and PCM-2. Figure 9

illustrates the liquid fraction and temperature contours that give a proper insight into the results. Notably, PCM-2 solidifies faster than PCM-1. The differences in solidification rates are due to a temperature gradient within the PCMs, which causes buoyancy-driven convection circulation within the melted PCMs. As seen in Figures 10a and 10b, velocity vectors reveal the formation of vortices within the PCMs. A single counter-rotating vortex forms within PCM-1, whereas multiple rotating vortices develop within PCM-2. The presence of multiple vortices in PCM-2 improves fluid mixing and promotes better convective heat circulation. As a result, PCM-2 dissipates heat more efficiently and achieves the solidus temperature faster than PCM-1. From Figure 9, it can be seen that at $n = 8$, each layer solidifies and the overall solid fraction is greater than $n = 2$. Firstly, with an increasing number of layers, there

could be a higher number of nucleation sites available for phase transition. The increased surface area provided by additional layers can promote better heat transfer, which can enhance the solidification process. Secondly, during solidification, temperature gradients inside the melted PCMs cause buoyancy-driven flow patterns, which may result in the formation of vortices. These vortices have an important role in redistributing liquid material and promoting mixing, hence accelerating the solidification process. The alternating arrangement of PCM-1 and PCM-2 accentuates these occurrences. The creation of clockwise and counter-clockwise vortices within PCM-2 is particularly noteworthy. This unique occurrence can be due to the alternating arrangement, which causes disruptions in flow dynamics, resulting in the formation of diverse vortex patterns. Furthermore, inter-layer heat transfer between

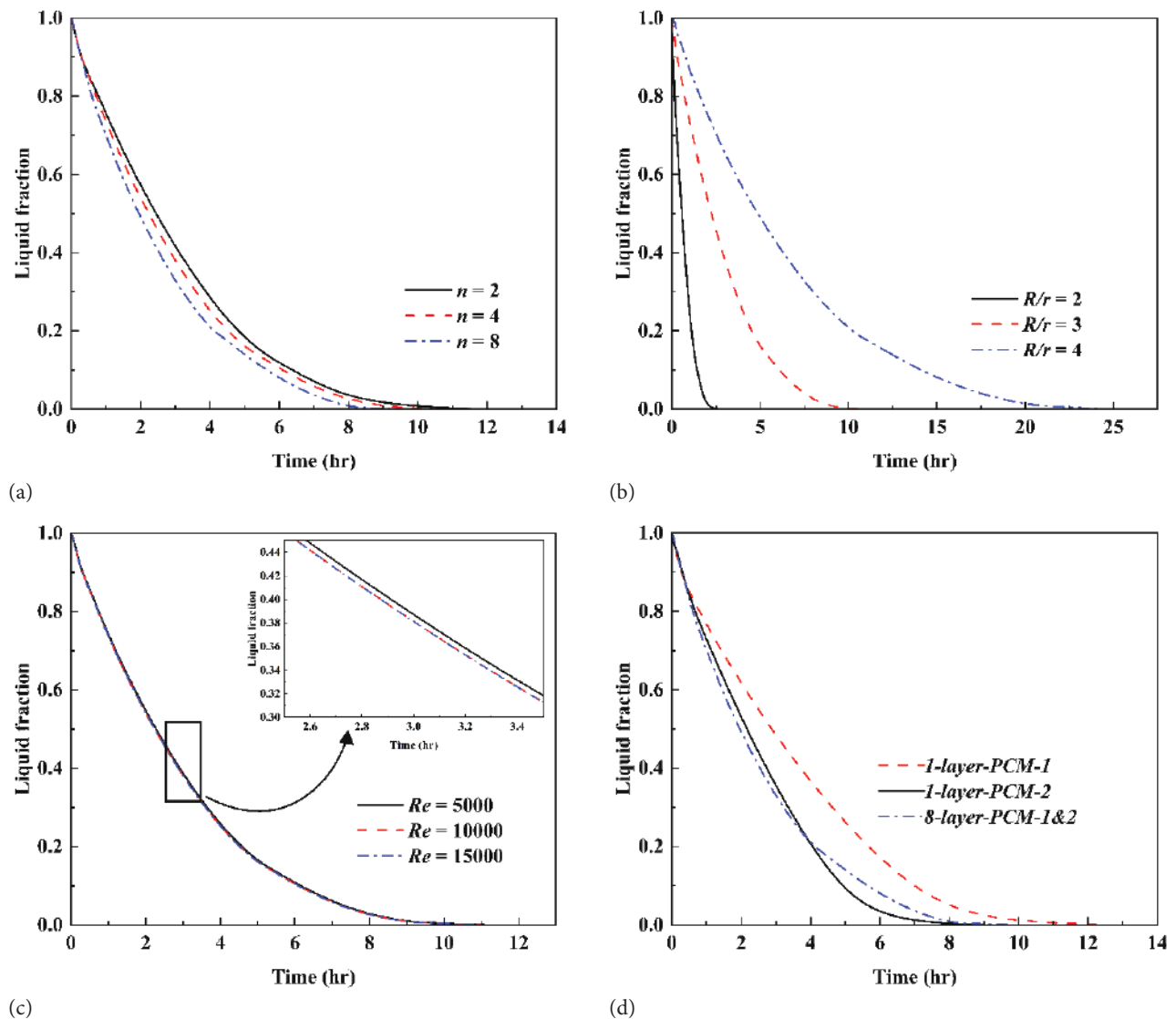


Figure 8. Temporal variation of liquid fraction with different parameters during solidification of PCMs at (a) $Re = 10000$, $R/r = 3$, (b) $Re = 10000$, $n = 4$, (c) $n = 4$, $R/r = 3$, (d) $Re = 10000$, $R/r = 3$.

the alternating layers of PCM-1 and PCM-2 aids in the generation of multiple vortices, as temperature gradients between successive layers cause buoyancy-driven flow patterns, promoting the emergence of complex flow structures. Figures 10c and 10d exhibit velocity vectors at $n = 8$, which depict flow mechanics in the PCM-1 and PCM-2 alternating arrangement. Notably, buoyancy-driven flow patterns are prominent in PCM-1, whereas PCM-2 leads to the formation of both clockwise and counter-clockwise vortices, which help to speed up solidification. The increase in the number of layers potentially enhances vortex formation as the liquid material traverses narrower interlayer spaces. This intensified flow creates more vortices, thereby further hastening the solidification process. This results in a reduction in the solidification time at about 19% if the number of layers increases from 2 to 8.

Figure 8b represents the time variation of the liquid fraction with the radius ratio of shell and tube. It is evident that increasing the R/r ratio and keeping the tube ratio constant leads to an increase in solidification time. The justification of such results can be explained in the context of thermal resistance, mass, and temperature gradient. The heat resistance of the liquid PCMs and the HTF increases

as the R/r ratio rises while the tube ratio remains constant. With a larger shell diameter, heat must travel a greater distance through the shell before being released to the HTF, resulting in delayed solidification. The mass of the PCM also has an important influence. As PCMs cool and solidify, the mass of the solid PCMs increases while the mass of the liquid PCMs decreases. As the R/r ratio grows, so does the mass of the liquid PCMs in contact with the tube's cold surface. The surface-to-volume ratio decreases as the R/r ratio increases, slowing the conductive heat transfer rate. Furthermore, when the R/r ratio increases the shell radius increases, reducing the radial temperature gradient. This is because, as the radial distance increases, the heat transfer rate from the solidification front to the PCMs' outer surface decreases. As a result, the temperature gradient decreases and solidification occurs at a slower rate. Quantitatively, it is observed that if the R/r ratio is reduced from 4 to 2, the solidification rate improves to about 89%. On the contrary, the thermal storage capacity is reduced by the lower R/r ratio. So, it is important to take into account the impact of the radius ratio of the shell to the tube when constructing a thermal energy storage system, as it can have a substantial impact on the system's solidification time and

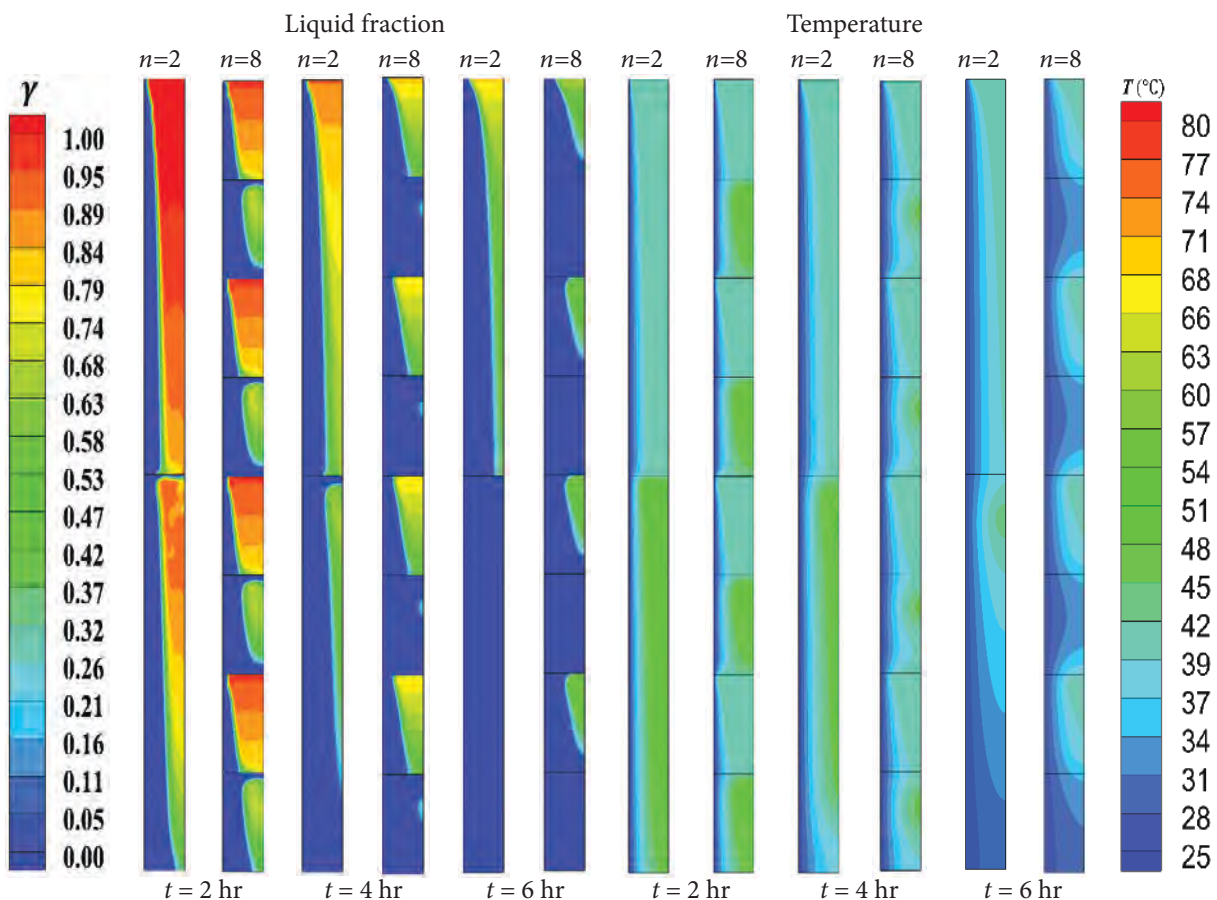


Figure 9. Contours of liquid fraction and temperature at different flow times and number of layers during solidification of PCMs at $Re = 10000$, $R/r = 3$.

overall performance. In such situations, such as in emergency backup power systems, the storage with a lower R/r ratio can quickly store and release large amounts of heat energy.

Figure 8c describes the variation of liquid fraction with Reynolds number, keeping all other parameters fixed. It can be seen that the solidification rate is dependent on the HTF flow Reynolds number. However, the changes due to the Reynolds number are less significant. When the Reynolds number rises, the efficiency of heat transport from the PCM to the HTF increases, which causes the liquid fraction of the PCMs to decrease during solidification. As the Reynolds number rises from $Re = 5000$ to $Re = 10000$, turbulence increases noticeably, which causes the liquid fraction of the PCM to noticeably decrease. However, the influence of turbulence on heat transfer becomes less significant as the Reynolds number rises from 10,000 to 15,000, while the liquid fraction of the PCMs remains mostly unchanged. This is because turbulence intensity and kinetic energy increase at very high Reynolds numbers, which may hinder solidification by causing fluctuations in temperature and flow velocity that may slow down the formation of a smooth and uniform solidification front. As a result, the effect of turbulence on the solidification rate

normally reduces over a threshold Reynolds number. In addition, when the Reynolds number increases, the ability of HTF to remove heat from the PCMs is restricted by the conductive resistance of the PCMs themselves. This means that even though the HTF may be capable of removing heat more effectively, the rate at which the PCMs can transport this heat to the solidification front does not rise proportionally. The overall solidification time minimizes about 1% if the HTF Reynolds number increases from 5000 to 15000.

Figure 8d presents a comparative analysis of the time variation of liquid fraction during solidification across three different cases, namely, *1-layer-PCM-1*, *1-layer-PCM-2*, and *8-layer-PCM-1&2* alternately arranged configuration. Initially, the solidification rate appears to rise significantly in the 8-layer alternating arrangement as compared to the single-layer PCM-1 configuration. However, as compared to single-layer PCM-2, it is clear that the solidification rate in the 8-layer configuration begins to decline after 4 hours. The decline can be attributed to the dynamic interaction between thermal characteristics and phase transition kinetics. While PCM-2 initially solidifies rapidly due to its proximity to the solidification temperature, the subsequent presence of solid PCM-2 may act as an insulating barrier, preventing additional heat transfer to the remaining PCM-1

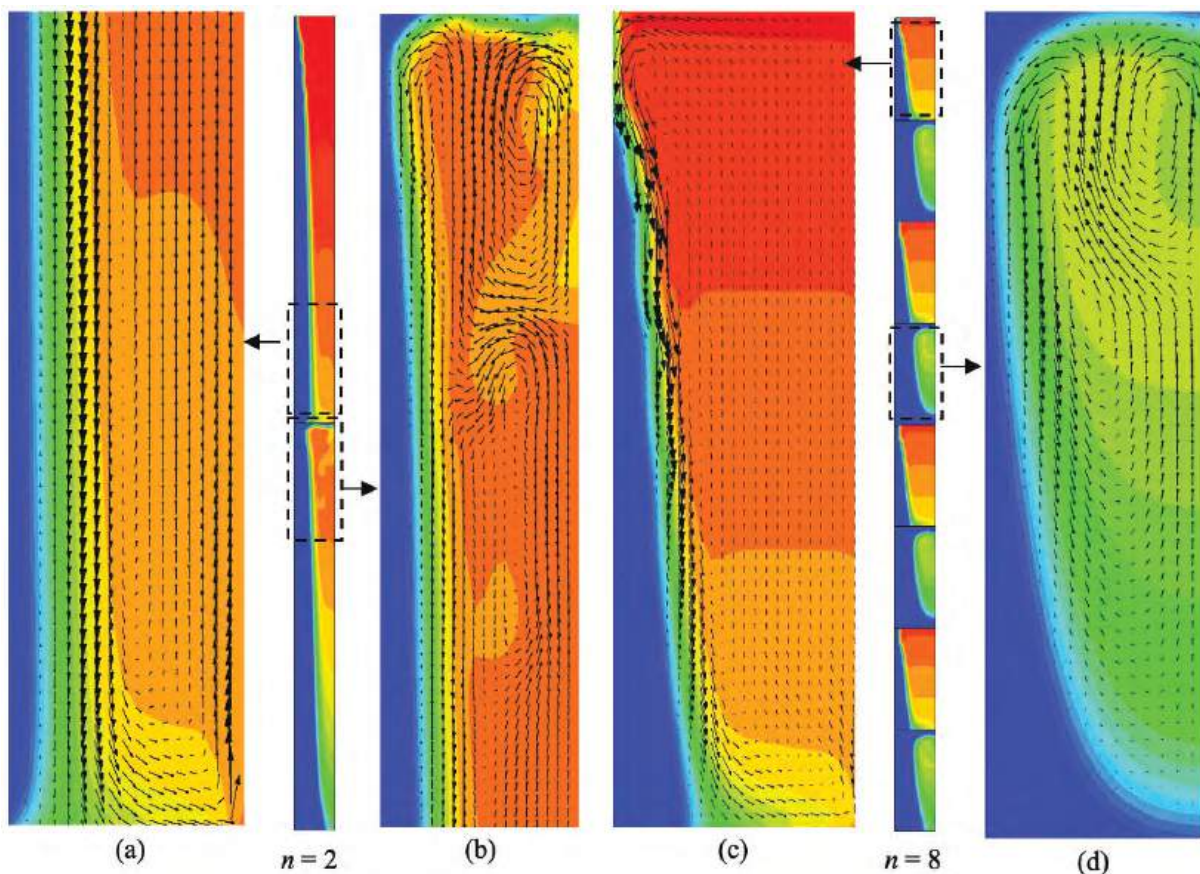


Figure 10. Velocity vector during solidification of PCMs at $Re=10000$, $R/r=3$, $t=2$ hr.

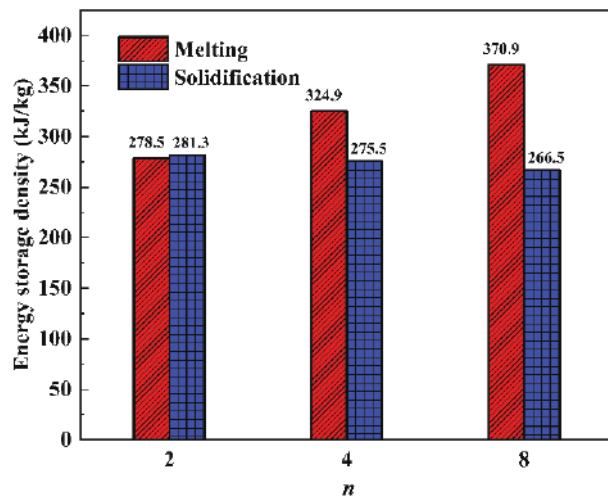
layers. From the liquid fraction contour depicted in Figure 9, it can be seen that after 4 hours, PCM-2 almost solidifies, leaving only PCM-1, causing the solidification rate to decline. Nevertheless, the complete solidification time in an 8-layer case is reduced approximately by 21.9% in comparison with a single-layer PCM-1. Therefore, the multi-layered arrangement of two PCMs alternating arrangement proves to be a more efficient way to reduce the overall solidification time while maintaining a reasonable solidification rate.

Storage performance evaluation

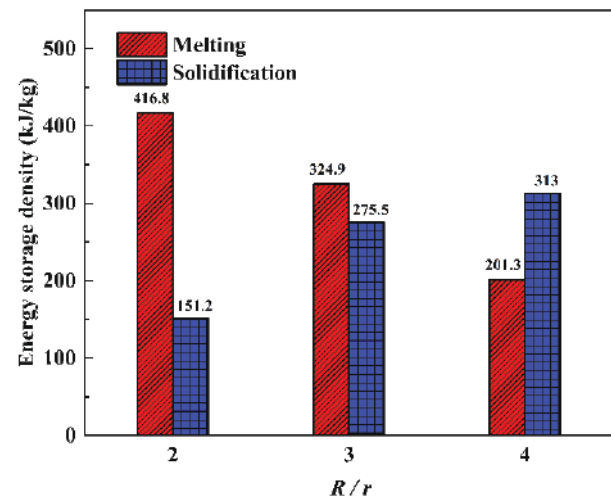
The evaluation of a heat storage system's performance stands essential for ensuring efficient storage and release of thermal energy. For evaluating the impact of all parameters on energy storage and retrieval performances, at a fixed time energy storage density has been calculated. Figure 11

represents the energy storage density with different parameters after 0.75 hr completion of melting and solidification.

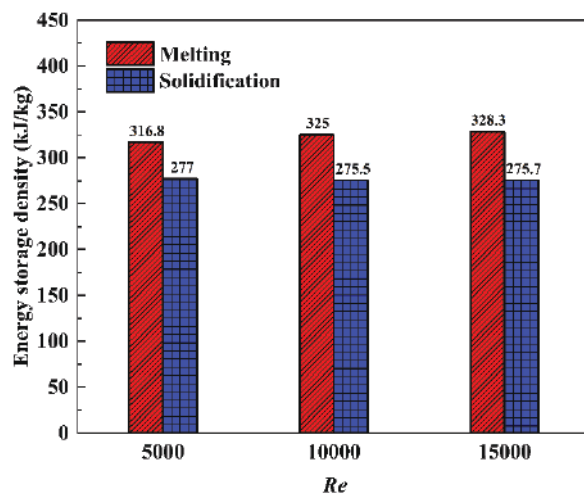
From Figure 11a, it is seen that at $t = 0.75$ hr, energy storage density is enhanced if the number of layers (n) increases from 2 to 8 during the melting process. In comparison with a 2-layer system, an 8-layer alternating arrangement can store 33.2% more energy at a fixed time. Besides, it can be also observed that at $t = 0.75$ hr, energy storage density is reduced if the layer increases from 2 to 8 during the solidification. It indicates that with the rise in the number of layers, more energy can be retrieved. In comparison with a 2-layer system, an 8-layer system can retrieve about 5.3% more energy at a fixed time. The key factors of these findings are applicable especially when quicker energy storage and retrieval is desired.



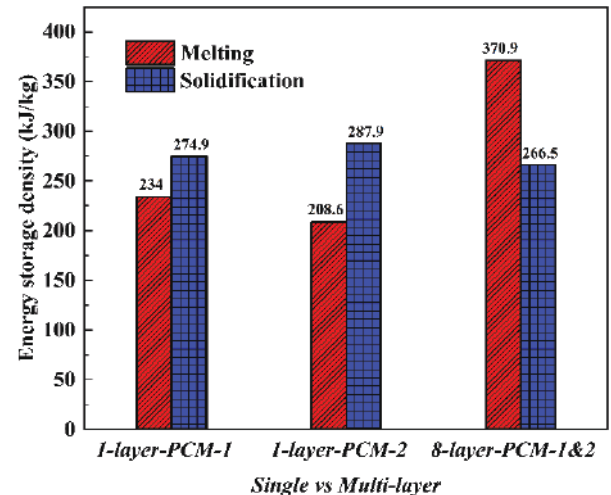
(a)



(b)



(c)



(d)

Figure 11. Quantitative performance evaluation of storage system based on energy storage density with different parameters after 0.75 hr completion of melting and solidification at (a) $Re = 10000$, $R/r = 3$, (b) $Re = 10000$, $n = 4$, (c) $n = 4$, $R/r = 3$, (d) $Re = 10000$, $R/r = 3$.

Figure 11b offers valuable insight into the storage performance of a system, particularly concerning the radius ratio of the shell to the tube at a specific time. Upon examination, it becomes apparent that the storage unit with a lower radius ratio is capable of efficiently storing the maximum amount of energy during the melting process, as compared to other storage units. The energy storage density tends to decrease as the radius ratio increases from 2 to 4. Based on quantitative analysis, it can be seen that the energy storage unit with $R/r = 2$ can store about 107% more energy than the one with $R/r = 4$, at a fixed time of $t = 0.75$ hr. Besides, it can also be found that during the solidification process, a lower R/r ratio-based storage system maximum energy can be retrieved quickly. Quantitatively, using a storage system having $R/r = 2$, 51.7% more energy can be retrieved quickly as compared to $R/r = 4$.

Figure 11c represents the energy storage and retrieval performance based on the HTF flow Reynolds number (Re) after 0.75 hr completion on melting and solidification. The results of this study illustrate that during the melting process, increasing the Reynolds number of the flow from 5000 to 15000 leads to an increase in the energy storage density. Specifically, the data shows that there is 3.6% more energy that can be stored. Similarly, during the solidification process, at a higher Reynolds number more quickly energy can be retrieved. Quantitatively, 0.5% more energy can be retrieved during solidification. This study reveals that for quicker storage and removal of energy, the flow Reynolds number is also a vital factor that needs to be under consideration for the designing of energy storage systems.

Figure 11d presents a comparative analysis of the storage performance of three different cases, namely *1-layer-PCM-1*, *1-layer-PCM-2*, and *8-layer-PCM-1&2* alternating arrangement. The results indicate that after 0.75 hr of melting, the 8-layer two PCM alternating arrangement can store significantly more energy compared to the single-layer individual cases. In quantitative terms, the 8-layer two PCM alternating arrangement can store approximately 58.5% and 77.8% more energy than the single-layer individual PCMs, respectively. Additionally, the study also reveals that during solidification, the 8-layer two PCM alternating arrangement can retrieve more energy than the individual cases. At a fixed time, the 8-layer two PCM alternating arrangement can retrieve energy at 3.1% and 7.4% more than the single-layer individual cases, respectively. Therefore, the results of the study suggest that using a multi-layered dual PCM alternating arrangement can improve energy storage and retrieval performance compared to using a single-layered PCM.

CONCLUSION

In conclusion, this parametric study offers a range of benefits that make it a compelling choice for designing an efficient thermal energy storage system. This study reveals the effect of different design and flow parameters on the storage performances during both the melting

and solidification of PCMs. This study also emphasized an insight comparison between single-layer PCM over multi-layered dual PCM alternating arrangement. The following are some key conclusions:

- As the number of layers (n) is increased from 2 to 8 the time required for complete melting and solidification time is reduced by about 44.3% and 19% respectively. In the context of energy storage/retrieval, an 8-layer system exhibits a capacity to store or release energy more rapidly compared to the 2-layer system with improvements of approximately 33.2% and 5.3% during the melting and solidification process respectively.
- If the radius ratio, R/r decreases from 4 to 2, the overall melting and solidification time is reduced by about 78.6% and 89% respectively. Storage of $R/r = 2$ can store/release energy quicker than $R/r = 4$ at about 107% and 51.7% during the melting and solidification process respectively.
- The impact of HTF flow Reynolds number (Re) on melting and solidification is comparatively small. The rate of melting and solidification improves by about 5.8% and 1% as the HTF flow Reynolds number raises from 5000 to 15000. The energy storage/retrieved density also advanced by about 3.6% and 0.5% as the flow Reynolds number increased.
- Finally, in comparison with *1-layer-PCM-1* and *1-layer-PCM-2*, *8-layer-PCM-1&2* alternating arrangement can augment melting and solidification rate. The total melting time decreases in the 8-layered configuration by 33.65% and 55.84% as compared to the single-layer of PCM-1 and PCM-2 respectively. Similarly, the complete solidification time is reduced by 21.9% compared to a single-layer PCM-1 case. Energy storage/retrieval performance advanced in an 8-layer system. It can store energy more rapidly about 58.5% and 77.8% compared to single-layer individual PCMs. Similarly, retrieval performances also accelerated by approximately 3.1% and 7.4% compared to single-layer individual PCMs.

In summary, multi-layer alternating PCM configurations ensure efficient operation, especially in applications where rapid energy storage/retrieval is crucial. This versatility makes them ideal for diverse applications, including renewable energy systems such as solar power plants, industrial processes, building HVAC systems, electronics, etc. Although this study shows improved melting and solidification performance adopting multi-layer twin PCMs in alternately arranged manners, multiple PCMs in the multi-layer arrangement need to be studied further. The arrangement of PCMs is also to be a major factor for study. The optimization of different parameters on the maximization of energy storage and retrieval performance needs to be extended further.

NOMENCLATURE

Abbreviation

PCM	Phase change material
HTF	Heat transfer fluid

LHTES Latent heat thermal energy storage
 TES Thermal energy storage

List of symbols

C_{mush} Mushy zone constant [kg/ (m³ s)]
 C_p Specific heat [J/ (kg K)]
 $C_{p,v}$ PCM variable specific heat [J/ (kg K)]
 g Acceleration due to gravity [m/s²]
 h Enthalpy [J/kg]
 K Thermal conductivity [W/ (m K)]
 k Turbulent kinetic energy [J]
 L Latent heat of fusion [J/kg]
 m Mass [kg]
 n Number of layers
 r Radius of tube
 R Radius of shell
 Re Reynolds number
 t Time [hr]
 T Temperature [°C]
 $\vec{V}$ Velocity vector [m/s]
 V Volume [m³]
 x Radial coordinate
 z Axial coordinate
 K_t Thermal conductivity due to turbulence [W/ (m K)]

Greeks

γ Liquid fraction
 ε Turbulent K.E. dissipation rate [m² /s³]
 μ Dynamic viscosity [kg/ (m s)]
 μ_t Turbulent viscosity [kg/ (m s)]
 ρ Density [Kg / m³]

ACKNOWLEDGMENTS

The authors would like to convey their sincere gratitude to the Ministry of Education, Government of India, for generously providing financial support in the form of a Ph.D. fellowship during this research.

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Korti AIN. Numerical simulation on the effect of latent heat thermal energy storage unit. *J Therm Eng* 2016;2:599-607. [\[Crossref\]](#)
- [2] Benbrika M, Teggat M, Benbelhout M. Effect of orientation of elliptic tube on the total melting time of latent thermal energy storage systems. *J Therm Eng* 2021;7:1479-1488. [\[Crossref\]](#)
- [3] Patil R, Desai A. Performance of phase changing material in an artificially created cold region to promote latent heat thermal energy storage. *J Therm Eng* 2021;7:1694-1703. [\[Crossref\]](#)
- [4] Aljabair S, Alesbe I, Ibrahim SH. Review on latent thermal energy storage using phase change material. *J Therm Eng* 2023;9:247-256. [\[Crossref\]](#)
- [5] Elfeky KE, Ahmed N, Wang Q. Numerical comparison between single PCM and multi-stage PCM based high temperature thermal energy storage for CSP tower plants. *Appl Therm Eng* 2018;139:609-622. [\[Crossref\]](#)
- [6] Prieto C, Cabeza LF. Thermal energy storage (TES) with phase change materials (PCM) in solar power plants (CSP). Concept and plant performance. *Appl Energy* 2019;254:113646. [\[Crossref\]](#)
- [7] Gurupatham SK, Manikandan GK, Fahad F. Harnessing and storing solar thermal energy using phase change material (pcm) in a small flat plate collector. *J Therm Eng* 2020;6:511-520. [\[Crossref\]](#)
- [8] Ben Zohra M, Riad A, Alhamany A, Sennoune M, Mansouri M. Improvement of thermal energy storage by integrating pcm into solar system. *J Therm Eng* 2020;6:816-828. [\[Crossref\]](#)
- [9] Kumar AR, Ramakrishnan M. A scoping review on recent advancements in domestic applications of solar thermal systems. *J Therm Eng* 2022;8:426-444. [\[Crossref\]](#)
- [10] Azeez K, Ahmed Ri, Obaid Za, Azzawi Idj. Heat transfer enhancement and applications of thermal energy storage techniques on solar air collectors: A review. *J Therm Eng* 2023;9:1356-1371. [\[Crossref\]](#)
- [11] Nwosu Ec, Nsofor K, Nwaji Gn, Ononogbo C, Ofong I, Ogueke N V., et al. Extended experimental investigation of a double-effect active solar still with a paraffin wax, in Owerri, Nigeria. *J Therm Eng* 2023;9:1189-1207. [\[Crossref\]](#)

- [12] Rastogi M, Chauhan A, Vaish R, Kishan A. Selection and performance assessment of Phase Change Materials for heating, ventilation and air-conditioning applications. *Energy Convers Manag* 2015;89:260-269. [\[Crossref\]](#)
- [13] Ljungdahl V, Taha K, Dallaire J, Kieseritzky E, Pawelz F, Iradi M, et al. Phase change material based ventilation module-Numerical study and experimental validation of serial design. *Energy* 2021;234:121209. [\[Crossref\]](#)
- [14] Ismail M, Zahra WK, Ookawara S, Hassan H. Enhancing the Air Conditioning Unit Performance via Energy Storage of Different Inorganic Phase Change Materials with Hybrid Nanoparticles. *JOM* 2023;1-15. [\[Crossref\]](#)
- [15] Bahrami L, Kasaeian A, Pourfayaz F, Ghafarian S. Modeling of the effect of nano-enhanced phase change material on the performance of a large-scale wallboard. *J Therm Eng* 2021;7:1857-1871. [\[Crossref\]](#)
- [16] Kladisios P, Stegou-Sagia A. Using phase change materials in photovoltaic systems for cell temperature reduction: A finite difference simulation method. *J Therm Eng* 2016;2:897-906. [\[Crossref\]](#)
- [17] Ali HM, Arshad A, Jabbal M, Verdin PG. Thermal management of electronics devices with PCMs filled pin-fin heat sinks: A comparison. *Int J Heat Mass Transf* 2018;117:1199-1204. [\[Crossref\]](#)
- [18] Bondareva NS, Sheremet MA. Numerical simulation of natural convection melting in 2D and 3D enclosures. *J Therm Eng* 2019;5:51-61. [\[Crossref\]](#)
- [19] Ren Q, Guo P, Zhu J. Thermal management of electronic devices using pin-fin based cascade microencapsulated PCM/expanded graphite composite. *Int J Heat Mass Transf* 2020;149:1-16. [\[Crossref\]](#)
- [20] Ramesh KN, Sharma TK. Thermal analysis of PCM-based hybrid micro-channel heat sinks: A numerical study. *J Therm Eng* 2023;9:1015-1025. [\[Crossref\]](#)
- [21] Merlin K, Soto J, Delaunay D, Traonvouez L. Industrial waste heat recovery using an enhanced conductivity latent heat thermal energy storage. *Appl Energy* 2016;183:491-503. [\[Crossref\]](#)
- [22] Yazdani MR, Laitinen A, Helaakoski V, Farnas LK, Kukko K, Saari K, et al. Efficient storage and recovery of waste heat by phase change material embedded within additively manufactured grid heat exchangers. *Int J Heat Mass Transf* 2021;181:121846. [\[Crossref\]](#)
- [23] Yadav C, Sahoo RR. Thermal analysis comparison of nano-additive PCM-based engine waste heat recovery thermal storage systems: an experimental study. *J Therm Anal Calorim* 2022;147:2785-802. [\[Crossref\]](#)
- [24] Humphries WR. Performance of finned thermal capacitors. 1974.
- [25] Swanson TD, Birur GC. NASA thermal control technologies for robotic spacecraft. *Appl Therm Eng* 2003;23:1055-1065. [\[Crossref\]](#)
- [26] Agyenim F, Hewitt N, Eames P, Smyth M. A review of materials, heat transfer and phase change problem formulation for latent heat thermal energy storage systems (LHTESS). *Renew Sustain Energy Rev* 2010;14:615-628. [\[Crossref\]](#)
- [27] Baghaei Oskoue S, Bayer Ö. Performance improvement in a vertical latent thermal energy storage tank with crossing heat transfer tubes. *J Energy Storage* 2024;88. [\[Crossref\]](#)
- [28] Chaabane M, Mhiri H, Bournot P. Thermal performance of an integrated collector storage solar water heater (ICSSWH) with phase change materials (PCM). *Energy Convers Manag* 2014;78:897-903. [\[Crossref\]](#)
- [29] Cano D, Funéz C, Rodriguez L, Valverde JL, Sanchez-Silva L. Experimental investigation of a thermal storage system using phase change materials. *Appl Therm Eng* 2016;107:264-270. [\[Crossref\]](#)
- [30] Elbahjaoui R, El Qarnia H. Thermal performance of a solar latent heat storage unit using rectangular slabs of phase change material for domestic water heating purposes. *Energy Build* 2019;182:111-130. [\[Crossref\]](#)
- [31] Şimşek F, Demirci H. Investigation of the use of new fin on the melting time of the phase change material stored in the heat exchanger by computational fluid dynamics analysis. *J Build Eng* 2024;91:109505. [\[Crossref\]](#)
- [32] Trp A. An experimental and numerical investigation of heat transfer during technical grade paraffin melting and solidification in a shell-and-tube latent thermal energy storage unit. *Sol Energy* 2005;79:648-660. [\[Crossref\]](#)
- [33] Zhang S, Pu L, Xu L, Liu R, Li Y. Melting performance analysis of phase change materials in different finned thermal energy storage. *Appl Therm Eng* 2020;176. [\[Crossref\]](#)
- [34] Feng L, Liu J, Lu H, Chen Y, Wu S. A parametric study on the efficiency of a solar evacuated tube collector using phase change materials: A transient simulation. *Renew Energy* 2022;199:745-758. [\[Crossref\]](#)
- [35] Alshihmani H, Maghrebi MJ, Sardarabadi M. Thermal performance prediction of a phase change material based heat-sink cooling system for a printed circuit board, using response surface method. *J Energy Storage* 2022;55:105499. [\[Crossref\]](#)
- [36] Refaey HA, Abdo S, Saidani-Scott H, El-Shekeil YA, Bendoukha S, Barhoumi N, et al. Thermal regulation of photovoltaic panels using PCM with multiple fins configuration: Experimental study with analysis. *Therm Sci Eng Prog* 2024;102457. [\[Crossref\]](#)
- [37] Xu H, Wang N, Zhang C, Qu Z, Cao M. Optimization on the melting performance of triplex-layer PCMs in a horizontal finned shell and tube thermal energy storage unit. *Appl Therm Eng* 2020;176:115409. [\[Crossref\]](#)

- [38] Mebarek-Oudina F, Chabani I. Review on Nano Enhanced PCMs: Insight on nePCM Application in Thermal Management/Storage Systems. *Energies* 2023;16:1066. [\[Crossref\]](#)
- [39] Li Z-R, Hu N, Fan L-W. Nanocomposite phase change materials for high-performance thermal energy storage: A critical review. *Energy Storage Mater* 2023;55:727-753. [\[Crossref\]](#)
- [40] Sathishkumar A, Sundaram P, Cheralathan M, Kumar PG. Effect of nano-enhanced phase change materials on performance of cool thermal energy storage system: A review. *J Energy Storage* 2024;78:110079. [\[Crossref\]](#)
- [41] Wang Q, Yang L, Song J. Preparation, thermal conductivity, and applications of nano-enhanced phase change materials (NEPCMs) in solar heat collection: A review. *J Energy Storage* 2023;63:107047. [\[Crossref\]](#)
- [42] Maghrabie HM, Elsaid K, Sayed ET, Radwan A, Abo-Khalil AG, Rezk H, et al. Phase change materials based on nanoparticles for enhancing the performance of solar photovoltaic panels: A review. *J Energy Storage* 2022;48:103937. [\[Crossref\]](#)
- [43] Bhutto YA, Pandey AK, Saidur R, Sharma K, Tyagi V V. Critical insights and recent updates on passive battery thermal management system integrated with nano-enhanced phase change materials. *Mater Today Sustain* 2023;23:100443. [\[Crossref\]](#)
- [44] Algarni S, Mellouli S, Alqahtani T, Almutairi K, Khan A, Anqi A. Experimental investigation of an evacuated tube solar collector incorporating nano-enhanced PCM as a thermal booster. *Appl Therm Eng* 2020;180:115831. [\[Crossref\]](#)
- [45] Bashirpour-Bonab H. Investigation and optimization of PCM melting with nanoparticle in a multi-tube thermal energy storage system. *Case Stud Therm Eng* 2021;28:101643. [\[Crossref\]](#)
- [46] Elarem R, Alqahtani T, Mellouli S, Aich W, Ben Khedher N, Kolsi L, et al. Numerical study of an Evacuated Tube Solar Collector incorporating a Nano-PCM as a latent heat storage system. *Case Stud Therm Eng* 2021;24:100859. [\[Crossref\]](#)
- [47] Rothan YA. Freezing process modeling within a cold storage tank with implement of nanoparticles. *J Energy Storage* 2024;88:111501. [\[Crossref\]](#)
- [48] Zahid I, Farhan M, Farooq M, Asim M, Imran M. Experimental investigation for thermal performance enhancement of various heat sinks using Al₂O₃NePCM for cooling of electronic devices. *Case Stud Therm Eng* 2023;41:102553. [\[Crossref\]](#)
- [49] Swamy KA, Verma S, Bhattacharyya S. Experimental and numerical investigation of nanoparticle assisted PCM-based battery thermal management system. *J Therm Anal Calorim* 2024. [\[Crossref\]](#)
- [50] Cao Y, Faghri A. Performance characteristics of a thermal energy storage module: a transient PCM/forced convection conjugate analysis. *Int J Heat Mass Transf* 1991;34:93-101. [\[Crossref\]](#)
- [51] Ismail KAR, Melo CA. Convection-based model for a PCM vertical storage unit. *Int J Energy Res* 1998;22:1249-1265. [\[Crossref\]](#)
- [52] Trp A, Lenic K, Frankovic B. Analysis of the influence of operating conditions and geometric parameters on heat transfer in water-paraffin shell-and-tube latent thermal energy storage unit. *Appl Therm Eng* 2006;26:1830-1839. [\[Crossref\]](#)
- [53] Tao Y-B, Li M-J, He Y-L, Tao W-Q. Effects of parameters on performance of high temperature molten salt latent heat storage unit. *Appl Therm Eng* 2014;72:48-55. [\[Crossref\]](#)
- [54] Zheng Z-J, Xu Y, Li M-J. Eccentricity optimization of a horizontal shell-and-tube latent-heat thermal energy storage unit based on melting and melting-solidifying performance. *Appl Energy* 2018;220:447-454. [\[Crossref\]](#)
- [55] Shen G, Wang X, Chan A, Cao F, Yin X. Investigation on optimal shell-to-tube radius ratio of a vertical shell-and-tube latent heat energy storage system. *Sol Energy* 2020;211:732-743. [\[Crossref\]](#)
- [56] Vyshak NR, Jilani G. Numerical analysis of latent heat thermal energy storage system. *Energy Convers Manag* 2007;48:2161-2168. [\[Crossref\]](#)
- [57] Akgün M, Aydın O, Kaygusuz K. Thermal energy storage performance of paraffin in a novel tube-in-shell system. *Appl Therm Eng* 2008;28:405-413. [\[Crossref\]](#)
- [58] Seddegh S, Tehrani SSM, Wang X, Cao F, Taylor RA. Comparison of heat transfer between cylindrical and conical vertical shell-and-tube latent heat thermal energy storage systems. *Appl Therm Eng* 2018;130:1349-1362. [\[Crossref\]](#)
- [59] Alaraji A, Alhussein H, Asadi Z, Ganji DD. Investigation of heat energy storage of RT26 organic materials in circular and elliptical heat exchangers in melting and solidification process. *Case Stud Therm Eng* 2021;28:101432. [\[Crossref\]](#)
- [60] Dukhan WA, Dhaidan NS, Al-Hattab TA, Al-Mousawi FN. Phase-change of paraffin inside heat exchangers: an experimental study. *Int J Environ Sci Technol* 2022;19:3155-164. [\[Crossref\]](#)
- [61] Zauglanmics E, Demircan T, Gemicioğlu B. Thermal analysis of phase changing material in heat exchanger. *Arab J Sci Eng* 2022:1-15.
- [62] Paria S, Baradaran S, Amiri A, Sarhan AAD, Kazi SN. Performance evaluation of latent heat energy storage in horizontal shell-and-finned tube for solar application. *J Therm Anal Calorim* 2016;123:1371-1381. [\[Crossref\]](#)
- [63] Joshi V, Rathod MK. Constructal enhancement of thermal transport in latent heat storage systems assisted with fins. *Int J Therm Sci* 2019;145:105984. [\[Crossref\]](#)
- [64] Nakhchi ME, Esfahani JA. Improving the melting performance of PCM thermal energy storage with novel stepped fins. *J Energy Storage* 2020;30:101424. [\[Crossref\]](#)

- [65] Nie C, Deng S, Liu J. Effects of fins arrangement and parameters on the consecutive melting and solidification of PCM in a latent heat storage unit. *J Energy Storage* 2020;29:101319. [\[Crossref\]](#)
- [66] Nóbrega CRES, Ismail KAR, Lino FAM. Solidification around axial finned tube submersed in PCM: Modeling and experiments. *J Energy Storage* 2020;29:101438. [\[Crossref\]](#)
- [67] Tiari S, Hockins A, Mahdavi M. Numerical study of a latent heat thermal energy storage system enhanced by varying fin configurations. *Case Stud Therm Eng* 2021;25:100999. [\[Crossref\]](#)
- [68] Brousseau P, Lacroix M. Study of the thermal performance of a multi-layer PCM storage unit. *Energy Convers Manag* 1996;37:599-609. [\[Crossref\]](#)
- [69] Li YQ, He YL, Song HJ, Xu C, Wang WW. Numerical analysis and parameters optimization of shell-and-tube heat storage unit using three phase change materials. *Renew Energy* 2013;59:92-99. [\[Crossref\]](#)
- [70] Ezra M, Kozak Y, Dubovsky V, Ziskind G. Analysis and optimization of melting temperature span for a multiple-PCM latent heat thermal energy storage unit. *Appl Therm Eng* 2016;93:315-329. [\[Crossref\]](#)
- [71] Beemkumar N, Karthikeyan A, Yuvarajan D, Lakshmi Sankar S. Experimental Investigation on Improving the Heat Transfer of Cascaded Thermal Storage System Using Different Fins. *Arab J Sci Eng (Springer Sci Bus Media BV)* 2017;42. [\[Crossref\]](#)
- [72] Sunku Prasad J, Muthukumar P, Anandalakshmi R, Niyas H. Comparative study of phase change phenomenon in high temperature cascade latent heat energy storage system using conduction and conduction-convection models. *Sol Energy* 2018;176:627-637. [\[Crossref\]](#)
- [73] Kareem BE, Adham AM, Yaqob BN. Design and Optimization of Air to PCM Heat Exchanger Using CFD. *Arab J Sci Eng* 2022;1-15. [\[Crossref\]](#)
- [74] Rubitherm Technologies GmbH. RT- LINE Data Sheet 2020.
- [75] Brent AD, Voller VR, Reid KJ. Enthalpy-Porosity Technique For Modeling Convection-Diffusion Phase Change: Application To The Melting Of A Pure Metal. *Numer Heat Transf* 1988;13:297-318. [\[Crossref\]](#)
- [76] Zhang D, Zhang F, Tang S, Guo C, Qin X. Performance evaluation of cascaded storage system with multiple phase change materials. *Appl Therm Eng* 2021;185:116384. [\[Crossref\]](#)

APPENDIX

The Finite Volume Method (FVM) is a key technique in computational fluid dynamics (CFD), and it is especially relevant to our research on thermal energy storage systems due to its multiple benefits. These benefits include conservation properties, grid flexibility, conservative interpolation, consistent stencils, physics adaptation, stability, precision, parallelizability, and strong commercial support from ANSYS Fluent. The advantages of the Finite Volume Method (FVM) and the FVM simulation steps are illustrated in the flow chart in Figure 12.

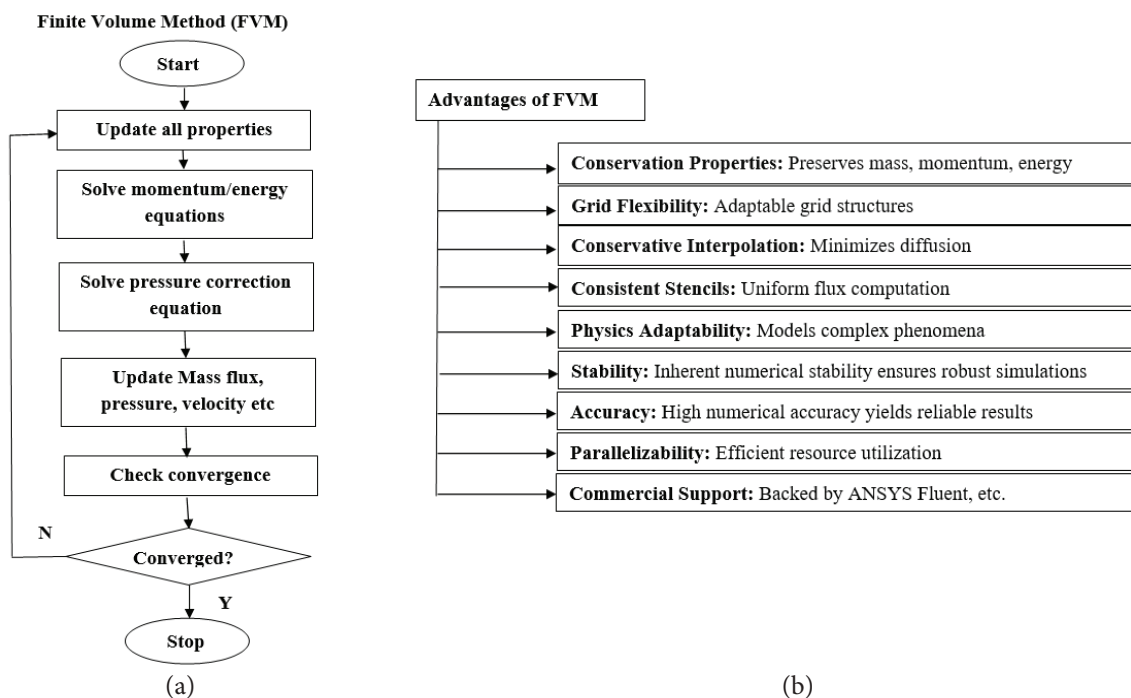


Figure 12. Flow chart of Finite Volume Method (FVM) (a) steps, (b) advantages



Research Article

Enhancing carbon neutrality through renewable energy and demand-side management

Yeqing ZHU¹, Yu WANG¹, Shen HOU¹, Yanjun CHEN², Luzheng ZHANG^{2,*}

¹State Power Environmental Protection Research Institute, Nanjing, Jiangsu, 210031 China

²Jiangsu Skytech Industrial Internet CO., LTD, Nanjing, Jiangsu, 211899, China

ARTICLE INFO

Article history

Received: 22 April 2025

Revised: 18 May 2025

Accepted: 20 May 2025

Keywords:

Carbon Reduction; Integrated Renewable Energy Systems; Carbon Dioxide Flux; Research and Office Parks

ABSTRACT

In the context of China's dual carbon goals—carbon peaking and carbon neutrality—the implementation of high-efficiency carbon reduction and control technologies is of paramount importance. This study evaluates the impact of renewable energy integration and energy management strategies on carbon emissions in a research and office institute park located in Nanjing, China. A 1.162 MW photovoltaic system was deployed to supply on-site electricity demand. The system exhibited substantial performance, achieving electricity self-sufficiency rates exceeding 100% on 50 days and surpassing 50% on 58.8% of the monitored days over the study period. To enhance energy efficiency on the demand side, the air conditioning temperature set point was raised from 25 °C to 27 °C. This adjustment led to a 15.1% reduction in air conditioning energy consumption and increased the average summer photovoltaic self-sufficiency rate from 51.27% to 56.85%.

In addition, a carbon flux tower was installed to facilitate continuous monitoring of carbon flux and atmospheric CO₂ concentrations. The measured data indicated consistently low carbon dioxide concentrations and negative carbon flux values, with average concentrations of 464.87 ppm and mean CO₂ flux of -0.0087 mg/m²·s, respectively. These results underscore the effectiveness of integrated renewable energy systems and active demand-side management in reducing operational carbon emissions in urban building clusters.

Cite this article as: Zhu Y, Wang Y, Hou S, Chen Y, Zhang L Enhancing carbon neutrality through renewable energy and demand-side management. J Ther Eng 2025;11(5):1540–1551.

INTRODUCTION

The modern world is facing serious environmental challenges, with climate change and global warming emerging as critical concerns. These issues are largely driven by carbon emissions originating from industrial activities,

transportation systems, and fossil fuel-based power generation [1].

The development of low-carbon office and research parks has emerged as a critical strategy in addressing climate change and contemporary energy challenges. Research in this domain encompasses a wide range

*Corresponding author.

*E-mail address: zhangluzheng@skynj.com

This paper was recommended for publication in revised form by Editor-in-Chief Ahmet Selim Dalkılıç



of topics, including sustainable architectural design, advanced energy management systems, carbon emissions assessment methodologies, and user behavior interventions. Current studies underscore an increasing demand for sustainable and energy-efficient work environments, as organizations strive to align their infrastructure with long-term carbon reduction targets—particularly those set for 2030 and 2060. Modern low-carbon parks emphasize not only the attainment of green building certifications but also the continuous optimization of building performance, enhancement of energy efficiency, electrification of energy systems, and large-scale integration of renewable energy technologies.

Sustainable design plays a pivotal role in reducing carbon emissions in office and research park developments. Low-carbon architectural strategies prioritize passive design approaches, including natural ventilation, daylight utilization, and the incorporation of energy-efficient building materials. Additionally, the integration of green infrastructure—such as rainwater harvesting systems, vegetative landscaping, and water recycling technologies—further supports the achievement of long-term sustainability objectives. For instance, Jiang et al. proposed a planning framework for agricultural rural industrial parks that emphasizes modernization and sustainability, leveraging the “negative carbon” potential of biochar and advanced agricultural facilities. Their approach utilizes a two-stage distributionally robust optimization model with a box-shaped uncertainty set to minimize total system costs, while simultaneously accounting for carbon emission reductions via carbon trading mechanisms [2]. Huang et al. [3] analyzed the low-carbon practices of an industrial park, focusing on measures like optimizing energy structure and transforming infrastructure. The findings suggested that optimizing energy structure and infrastructure transformation are vital for energy savings in industrial parks [3]. Wu et al. [4] delved into the multi-parameter optimization design approach for energy systems within low-carbon parks, revealing that energy storage devices play a significant role in elevating renewable energy utilization and promoting energy conservation and emission reduction [4]. Li et al. [5] discussed the performance and photovoltaic (PV) benefits analysis of multi-source renewable energy systems applied to various buildings on a university campus. It highlights the potential of utilizing renewable energy, specifically photovoltaic-assisted Heating, Ventilation, and Air Conditioning (PV-HVAC) systems, to reduce energy consumption on campus [5]. Aghamolaei et al.’s research showed that adopting renewable energy, optimizing energy systems, enhancing building envelope structures, and implementing intelligent technologies can make significant contributions to reducing carbon emissions and achieving a net zero carbon campus [6]. Hiltunen et al. [7] investigated the potential of reducing carbon emissions and enhancing energy efficiency on a university campus by connecting its heating system to the district heating network of the city. The key

findings revealed that both connections lead to substantial CO₂ emission reductions compared to the existing natural gas boiler system of the campus [7]. The research on the University of Palermo campus as small-scale models of cities demonstrated that initiatives such as installing photovoltaic plants, green roofs, LED street lighting, and promoting sustainable mobility through mobile apps have significantly reduced energy consumption and CO₂ emissions, and university campuses have the potential to act as models for sustainable urban development [8].

Intelligent energy management systems are widely applied in low-carbon parks, optimizing energy use through IoT (Internet of Things) technologies and data analytics. Distributed energy resources such as micro grids and energy storage further enhance energy efficiency by balancing supply and demand, particularly during peak usage times. These systems help to manage energy loads efficiently, reducing peak energy demand and improving energy reliability within the park. Li et al. [9] optimized CO₂ utilization in a chemical industrial park in China using a carbon allocation network, guided by the Automated Carbon Tracking Algorithm (ACTA). Four scenarios are analyzed and achieved over a 50% reduction in carbon emissions at the park [9]. Zhao et al. established a comprehensive assessment framework with a multi-objective optimization model for reducing both carbon and air pollutant emissions in industrial parks. Five different scenarios were designed to quantify the reduction potential of greenhouse gases and air pollutants and evaluate the co-emission reduction control effects under four measures, including the industrial structure adjustment, the energy structure adjustment, the energy efficiency improvement, and the industrial synergy [10]. Qian et al. [11] proposed a low-carbon optimization scheduling method for energy-intensive industrial parks, integrating a parameter-adaptive demand response incentive mechanism. A three-layer optimization model focusing on energy supply, load-side optimization, and demand response parameter optimization was established [11]. Lyu et al. [12] addressed uncertainty in the integrated energy systems (IES) by proposing a low-carbon, robust economic dispatch model based on price. It incorporates vehicle-to-grid technology, a price-based integrated heat and electricity demand response model, and a carbon trading model [12]. Ho et al. [13] presented a multi-objective planning model to support energy conservation and renewable energy in low-carbon campuses. Using multi-objective linear programming (MOLP) and a fuzzy two-stage algorithm, it aims to reduce CO₂ emissions by considering electricity and heat from solar panels and water heaters, along with energy efficiency improvements from rooftop gardens [13]. A demand response (DR) model for heat and electricity, incorporating prices and a vehicle-to-grid (V2G) system, was put forward by Li et al. It employed the Monte Carlo method to analyze uncertainties in emission reduction technologies and economic development. The results indicate that the model effectively maintains system

security and acceptable carbon emissions, serving as a roadmap for low-carbon industrial park development [14]. Hu et al. [15] adopted a multi-objective bi-level optimization approach, utilizing the non-dominated sorting genetic algorithm II (NSGA-II) in an optimal planning model for electric-heating integrated energy systems (EH-IES) in low-carbon parks. The upper-level model optimizes district heating network (DHN) equipment configuration based on economic costs and thermal energy satisfaction. The lower-level model then optimizes the hybrid energy storage system and accesses locations in the distribution network (DN), aiming to reduce net load fluctuation and voltage deviation [15]. Huo et al. [16] developed a dual-layer optimization strategy for a photovoltaic storage-hydrogen system in coal chemical industry parks. It proposed an energy management scheme that maximizes daily income based on time-of-use electricity prices, renewable energy generation, and load demands, providing a strategic roadmap for sustainable energy transitions in coal chemical industries [16].

Besides, assessing the carbon footprint of parks is essential for formulating strategies to reduce emissions. Life-cycle assessments (LCA) and carbon footprint calculations enable the precise evaluation of emissions from building construction, transportation, energy use, and waste management. These assessments inform carbon neutrality strategies, such as carbon capture, carbon sinks (e.g., afforestation), and participation in carbon trading schemes. Zheng et al. analyzed the sustainability of Huazhong University of Science and Technology (HUST) in China based on ecological footprint evaluation and machine learning [17]. Wang et al. [17] investigated the synergistic effect between pollution reduction and carbon emissions in an industrial park through LCA. Results showed a positive synergistic relationship between pollutants and carbon emissions, with strong synergy observed for SO_2 , NO_x , and $\text{PM}_{2.5}$ due to their close link with energy consumption [18]. Wang et al. [18] assessed the carbon footprint of a medium-sized university campus in eastern China through an innovative hybrid methodology combining the Long-range Energy Alternatives Planning System (LEAP) and LCA. To reduce the carbon footprint, seven mitigation strategies are proposed, including electricity decarbonization, waste recycling, greening, HVAC control system optimization, installation of solar PV panels, lighting control, and control of appliances [19].

User behavior is another critical factor influencing energy consumption in low-carbon parks. Research shows that energy-saving behaviors can be encouraged through interventions such as energy-use feedback, behavioral nudges, and energy-saving incentives. Moreover, the implementation of shared economy models, such as shared workspaces and transportation, significantly reduces energy use. Reference [17] also gathered the students' daily carbon emissions related to clothing, food, housing, consumption, and transportation. The study underscores the

importance of individual behavior changes, especially in energy-intensive activities like air conditioning use and food consumption.

Case studies from China's Yangtze River Delta, Pakistan, and ten Asian countries are analyzed using methods such as the Low Emissions Analysis Platform (LEAP) model, Autoregressive Distributed Lag (ARDL) model, and complex network evolutionary game model, exploring the impacts of renewable energy integration, demand-side management, and carbon trading policies on carbon neutrality.

Zhou et al. [20] focused on the China's Yangtze River Delta region, constructing a LEAP model to simulate five scenarios (64 sub-scenarios). They found that only 22 sub-scenarios could achieve carbon neutrality by 2060. A scenario with a 6% annual reduction in energy intensity achieved peak emissions ahead of schedule, while mixed policies (energy efficiency improvement, structural reform, and technological innovation) significantly reduced emissions, requiring an electrification rate of 64.19% in end-use sectors. The Analytic Hierarchy Process (AHP) validated that mixed policies enhance system reliability and cost-effectiveness.

Raza et al. [21] analyzed Pakistan's energy transition, comparing the Fossilized Energy System (FES) and Defossilized Energy System (DES). The DES scenario, with a 59.1% renewable energy share, reduced carbon emissions by 56,152.74 million tons but required an annual investment of \$13.1 trillion. Economic indicators showed stabilization in inflation and unemployment after 2045, demonstrating the long-term economic benefits of renewable transitions.

Zhang et al. [22] examined carbon emissions from China's tourism sector using the ARDL model, revealing positive correlations between tourist numbers, fossil fuel consumption, and emissions, and a negative correlation with renewable energy use.

Wei et al. [23] studied demand-side flexible resources in China's power system, including demand response, electric vehicle charging, and power-to-hydrogen. Demand-side flexible resources reduced transition costs by 20%, decreasing investments in energy storage and flexible generation. Electric vehicle and power-to-hydrogen loads played pivotal roles in regional power balancing, reducing gas generation demand by 42% by 2060 and enhancing renewable energy integration.

Shafiei et al. [24] proposed a multi-objective optimization model integrating wind, photovoltaic (PV), energy storage, and demand response, using the Gravity Search Algorithm (GSA). Case simulations showed that wind power reduced generation by 6.61%, energy storage by 9.4%, and PV by 10.8%, while demand response reduced daily peak load by 45%.

Lu et al. [25] constructed a complex network evolutionary game model to analyze interactions between energy supply and demand sides under carbon trading policies.

They found that carbon price trends influence strategic choices.

Chau et al. [26] focused on ten Asian countries, exploring how natural resources and eco-financing impact renewable energy production. It highlighted that rational resource utilization and financial instruments can accelerate low-carbon transitions and enhance the feasibility of carbon neutrality pathways.

The current study investigates the energy load characteristics of a research and office institute park in Nanjing, China, with a focus on the distinct demands of office and research facilities. A renewable energy system centered on photovoltaic (PV) generation was implemented, achieving an average overall power self-sufficiency rate of 69.75%. To enhance energy utilization efficiency, a demand-side regulation strategy was applied by adjusting the indoor air conditioning set point. Notably, this study introduces the integration of a carbon flux tower within an urban office park environment to enable real-time monitoring of carbon emissions and sequestration. This combination of renewable energy deployment, active demand-side management, and direct carbon flux measurement provides a novel, data-driven approach to evaluating the carbon reduction performance of low-carbon campus systems.

System Description

Park profile

The research and office institute park is located at Nanjing Jiangbei New Area. As a state-level development zone, Jiangbei New Area focuses on promoting green energy and environmental protection industries.

The total building area of the park is 25,180 square meters, with a well-planned layout and distinct functional zones. As listed in Table 1, the main building types in the park consist of research laboratories, office buildings, apartments and canteen.

Energy load characteristics in the park

The energy load patterns are influenced by the type of building, functional use, equipment characteristics, and usage time.

(1) Office Buildings

The energy load of office buildings is primarily driven by air conditioning, lighting, and office equipment. During summer and winter, the air conditioning load peaks,

particularly during working hours. In summer, as external temperatures rise, the air conditioning systems operate to maintain indoor comfort, resulting in two distinct peaks in energy consumption: one in the morning (9 AM to 11 AM) and another in the afternoon (1 PM to 3 PM). Lighting loads are mainly concentrated during office hours from 8:00 AM to 18:00 PM, with significantly lower consumption during weekends and evenings. The energy consumption of office equipment, such as computers and printers, is relatively stable and correlates with working hours.

(2) Laboratories

The energy load in laboratories is mainly attributed to experimental equipment, ventilation systems, and air conditioning. Additionally, the laboratories often need to maintain specific temperature and humidity conditions, which increases the air conditioning load. Moreover, to ensure safety and compliance with environmental standards, efficient ventilation systems are necessary, further increasing energy consumption. Laboratories experience energy load peaks during working hours, and due to longer experiment cycles, energy consumption may present a more stable pattern throughout weekdays.

(3) Researchers Apartments

Energy consumption in researchers' apartments is primarily concentrated in lighting, hot water supply, household appliances, and air conditioning. As residential spaces, the energy load in apartments is relatively dispersed, with higher consumption during evenings and weekends. In winter and summer, the heating and cooling demands lead to increased energy consumption, especially during colder or hotter weather. The energy consumption patterns in apartments typically exhibit diurnal fluctuations, but overall remain stable, primarily influenced by the living habits of occupants and external climate conditions.

(4) Canteens

The energy load of canteens is closely linked to food service operations, including cooking equipment, lighting, and ventilation systems. Unlike other types of buildings, canteens also utilize gas energy for cooking processes, which adds another layer to their energy consumption profile. Peak energy consumption occurs during breakfast and lunch periods, particularly during the lunch rush, when both electrical and gas demands significantly increase. Canteens require substantial hot water and ventilation, contributing to the overall energy consumption. The operational hours of canteens usually align with weekdays, leading to distinct energy load patterns, with notably lower consumption during weekends.

To sum up, the energy load characteristics of different buildings exhibit significant temporal and spatial patterns in the park. During weekdays, the combined peak energy loads from office buildings and laboratories can result in high overall campus energy consumption. Conversely, during weekends, the energy consumption patterns of researcher apartments and cafeterias remain stable, while the energy consumption of office buildings and laboratories declines

Table 1. The main buildings in the park

Building type	Number	Total area (m ²)
Office building	2	11200
Laboratory	3	6300
Apartment	4	10400
Canteen	1	1180

Table 2. Layout area and power of solar photovoltaic panel in the park

Building	Roof area (m ²)	Number of photovoltaic modules	Power capacity (kWh)
Office building	1280	110	59.95
Laboratories	4975	1190	648.55
Electric bicycle carports	1496	576	313.9
Ground	800	256	139.5

significantly. To optimize the energy management within the park, the following strategies can be implemented:

Smart Control Systems: Introduce intelligent control systems to dynamically adjust the operation of air conditioning, lighting, and equipment to balance energy loads across various buildings.

Demand-Side Management: Implement demand-side management strategies during peak consumption periods, encouraging users to operate high-energy-consuming equipment during off-peak times.

Utilization of Renewable Energy: Promote the use of renewable energy sources, such as solar and wind, to reduce overall park energy costs and environmental impact.

Energy Monitoring and Analysis: Regularly energy consumption monitoring and analysis to identify high-energy-consuming equipment and optimize or retrofit them accordingly.

Renewable Energy Application Measures

Solar photovoltaic panels

To reduce the CO₂ emission in the park, the utilization of renewable energy is necessary. Nanjing Jiangbei New Area is in a subtropical monsoon climate zone, with an average annual sunshine duration exceeding 2,000 hours, which provides favorable natural conditions for solar photovoltaic (PV) power generation. Comparatively, wind energy resources in Nanjing Jiangbei New Area are relatively limited, primarily due to topographical and climatic factors. The geothermal energy resources in Nanjing Jiangbei New Area are relatively abundant, particularly in terms of geothermal water and shallow geothermal energy. But it is more suitable for applications such as air conditioning systems and hot water supply, and not suitable for electricity generation due to its low grade.

Therefore, within the Nanjing Jiangbei New Area, utilizing solar energy resources for power generation is the most feasible method for renewable energy utilization in the park. The abundant solar radiation and the ongoing promotion of photovoltaic projects make solar energy the primary renewable resource for meeting the park's electricity demands.

In the park, monocrystalline silicon solar photovoltaic panels have been installed on the roofs of laboratory buildings, office buildings, and electric bicycle carports, with a photoelectric conversion efficiency of 21.3% and a peak

power of 545Wp. Additionally, solar photovoltaic panels have also been placed in certain ground spaces within the park, resulting in a total generation area of 8551 m². The specific distribution of these panels across various building spaces is detailed in Table 2. The park currently hosts a 1.162 MW smart photovoltaic capacity, equipped with a 60 kW/60 kWh energy storage system and two 60 kW DC (Direct Current) charging stations. The average daily electricity generation is over 3,000 kWh, operating under the model of “self-consumption with excess electricity fed back to the grid,” which essentially covers the electricity load during office hours on the campus.

Control regulation and energy storage

Additionally, the park features a micro grid energy cloud management system and a demonstration platform for integrated resource optimization and control of the source-grid-load-storage (SGLS) system. The micro grid energy cloud management system is built on the park's photovoltaic systems, energy storage, and charging stations, incorporating elements such as photovoltaic generation, energy storage, meteorological monitoring, network communication, direct/alternating current load applications, energy management, and centralized information control. This system enables functions including data collection and processing, sequential control, power control, generation forecasting, load forecasting, and optimized operation. The integrated resource optimization and control platform aggregates resources from both the generation side and the load side within the park. It enhances energy consumption monitoring and facilitates the collection and analysis of data from photovoltaic systems and energy storage, thereby improving the utilization rate of energy assets and the absorption rate of photovoltaic generation.

The supply of solar PV resources exhibits instability, as light intensity is influenced by weather conditions (such as cloudy or rainy days) and seasonal variations, leading to fluctuations in power generation under different times and conditions. This instability makes it challenging for solar power to match stable electricity demand, increasing the complexity of grid management. Therefore, it is necessary to integrate energy storage technologies and flexible scheduling to balance supply and demand, enhancing the reliability and adaptability of the system. Based on this, within the park, photovoltaic energy storage will be facilitated

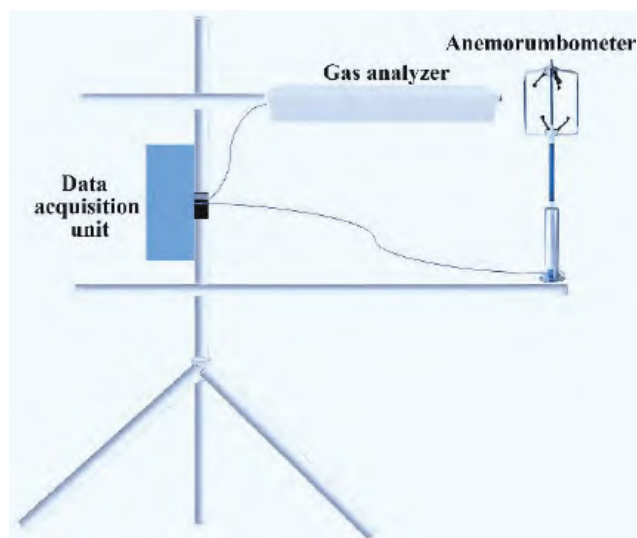


Figure 1. Schematic diagram of carbon flux tower.

through charging stations located in the electric bicycle carports, achieving a total charging power of 34.56 kW.

Carbon flux tower

Accurate monitoring and evaluation of carbon flux and carbon sequestration capacity are crucial for assessing the actual carbon reduction effects within the park and represent significant tasks in ecological research and management. Therefore, a carbon flux tower has been established within the park, as shown in Figure 1. The carbon flux tower is an important device for real-time dynamic monitoring, which measures gas concentrations and airflow movements at high frequencies, enabling precise capture of the carbon uptake and release processes by plants, soils, and other ecosystems.

The main components of the carbon flux tower include CO₂ flux analyzer (see Figure 2), anemorumbometer, and data acquisition unit. The CO₂ flux gas analyzer employs Tunable Diode Laser Absorption Spectroscopy (TDLAS) technology and its main components include a light source, a beam splitting system, a sampling system, a detector, and a data processing system. The near-infrared DFB diode laser with wavelength of 2004 nm is adopted to achieve online measurement of CO₂ concentration. The measurement range is 0–2000 ppm, with a measurement accuracy of 1% of the reading. The operating temperature ranges from -10°C to 50°C, which can meet the application requirements for carbon flux measurement in practical processes. The carbon flux tower is installed on the roof of the office building, with parameters designed to obtain the most accurate carbon flux data. This data is crucial for evaluating the carbon sequestration capacity within the park and for understanding the responsiveness of renewable energy and its regulation mechanisms to climate change, thereby providing data support for the management and protection of ecosystems.

RESULTS AND DISCUSSION

In this section, we present the outcomes of renewable energy systems and carbon flux monitoring at the research and office institute park. The deployment of solar photovoltaic power generation, demand-side regulation, real-time carbon flux monitoring has proven effective in reducing the park's carbon emissions, enhancing energy efficiency, and optimizing renewable energy utilization.

Energy Generation and Utilization

The park's energy generation capacity is centered around its 1.162 MW solar PV system, which serves as the primary renewable energy source. The system's performance is monitored by an integrated micro grid energy

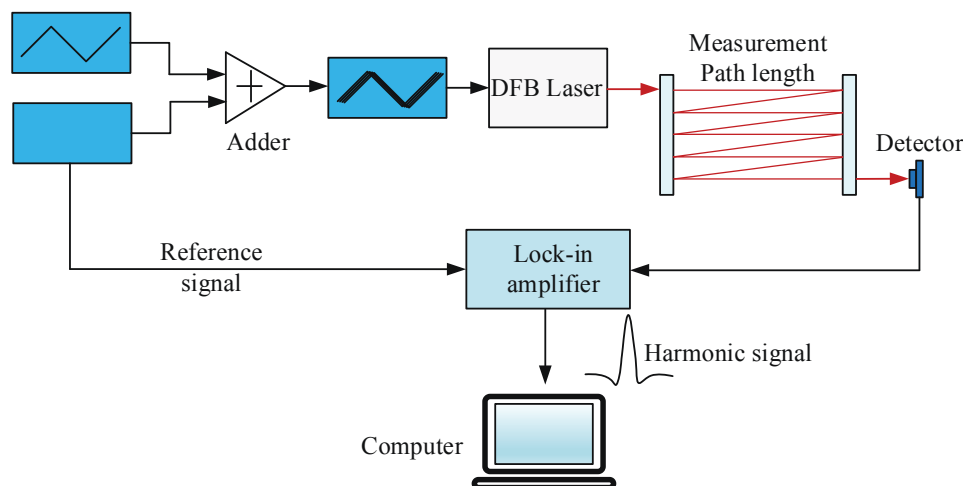


Figure 2. Schematic diagram of the CO₂ flux analyzer.

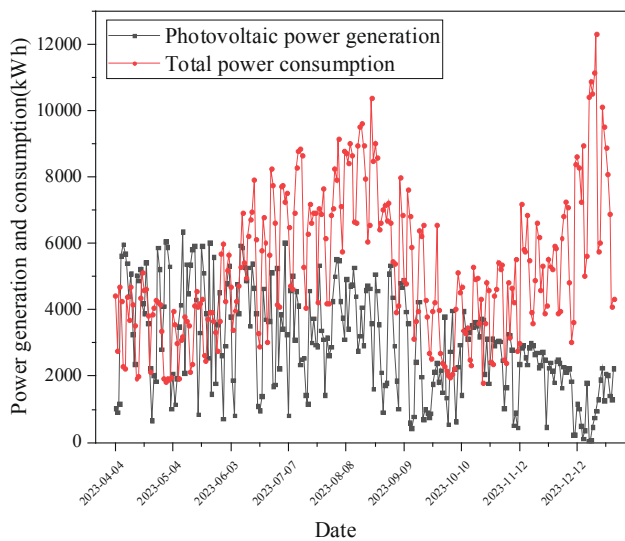


Figure 3. Energy production and consumption curve in the park from April to December.

cloud management system, which ensures that electricity is distributed efficiently across the park.

Figure 3 illustrates the daily variation in electricity generation of the PV power system within the park from April to December, after its deployment in practical operation. As observed in the figure, the average daily power generation from April to June—during the spring and early summer period—reached 3779.86 kWh. This higher output can be attributed to the favorable air quality and enhanced atmospheric ventilation in Nanjing during this season, which resulted in stronger solar radiation intensity. In contrast, the average daily power generation decreased to 3215.96 kWh from July to September, during the summer months. This reduction is primarily due to the frequent rainfall and high humidity in Nanjing during summer, where the presence of abundant water vapor attenuated solar radiation intensity. From October onwards, in the autumn and winter months, the average daily power generation declined further, reaching only 2146.18 kWh. This reduction is attributed to the lower solar altitude angle relative to the ground in winter, which results in reduced solar radiation intensity in the Nanjing region.

Furthermore, Figure 3 reveals the comparative relationship between the PV power generation and the total electricity consumption of the park. From April to June, the PV power generation generally exceeds the park's total daily electricity consumption, indicating that the PV system is capable of achieving self-sufficiency in meeting the park's electricity demand during this period. However, in most days after July, the reduction in solar radiation intensity results in the park's total electricity consumption surpassing the PV power generation.

Figure 4 further illustrates the daily self-sufficiency rate of solar PV power generation from April to December. The

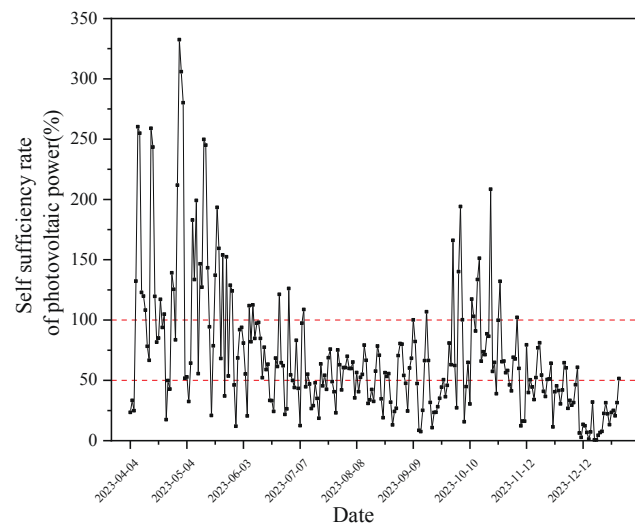


Figure 4. Daily self-sufficiency rate of solar photovoltaic power generation.

self-sufficiency rate is defined as the ratio of PV power generation to total electricity consumption. Clearly, a higher self-sufficiency rate, especially values reaching or exceeding 1, is desirable, as it indicates that the renewable energy generated by the PV system is sufficient to meet the park's electricity demands. According to the data in Figure 3, out of the 260 days from April to December, the self-sufficiency rate exceeded 100% on 50 days. Among these, 35 days occurred from April to June, 4 days from July to September, and 11 days from October to December. Additionally, 33 days were recorded from April to June, 39 days from July to September, and 31 days from October to December

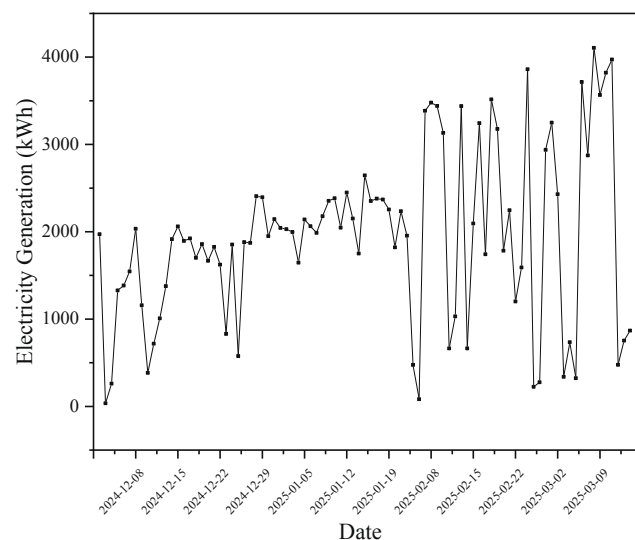


Figure 5. Energy production in the park from December to March.

where the self-sufficiency rate was between 50% and 100%. According to this, the PV power generation surpassed 50% of the park's electricity demand on 58.8% of the days, demonstrating how significant renewable energy is to the park's power supply.

Figure 5 presents the photovoltaic power generation data from December 2024 to March 2025. The corresponding monthly average power generation values are 1519.62 kWh, 1790.58 kWh, 1991.33 kWh, 2244.12 kWh and 2230.74 kWh, respectively. The reasons for the changes are analyzed as follows: In the Northern Hemisphere, November, December, and January belong to winter, while February and March gradually transition to spring. During winter, the solar altitude angle is relatively low, and the sunshine duration is short, which reduces the amount of solar radiation reaching the photovoltaic panels, resulting in relatively low photovoltaic power generation. The December average of 1519.62 kWh is the lowest value. As time progresses to February and March, the solar altitude angle gradually increases, the sunshine duration lengthens, and the solar radiation intensity increases, leading to an increase in photovoltaic power generation. The February average reaches 2244.12 kWh, and the March average is 2230.74 kWh. Overall, the trend of photovoltaic power generation is closely related to the changes in solar radiation brought about by seasonal changes.

Based on the actual operational data, it is estimated that the PV system can generate an annual average of 1.062 MWh of electricity for the park. In comparison, the total electricity consumption of the park for the year 2023 was 1.689 MWh. According to the latest electricity carbon emission factor for Jiangsu Province, officially released by the Ministry of Ecology and Environment of China (0.6451 kg CO₂/kWh), the PV system is expected to reduce carbon dioxide (CO₂) emissions by approximately 685.44 tons annually.

Regulation and Demand Management

As shown in Figures 3 and 4, the self-sufficiency rate of PV power generation is relatively low in both summer and winter. In winter, this is primarily due to the inherently low levels of solar radiation. In summer, however, the increased load and energy consumption from air conditioning systems represent another significant contributing factor. To enhance the proportion of renewable energy in the power supply (i.e., the self-sufficiency rate mentioned above), improvements can be achieved through measures such as regulation and demand-side management.

The total building area of the entire park is 25,180 m². In the Nanjing region, the summer climate is hot and humid, resulting in significant air conditioning loads and energy consumption. To address this issue, this study employs a simulation approach using EnergyPlus software to model the change in energy consumption within the park when the indoor air conditioning temperature set

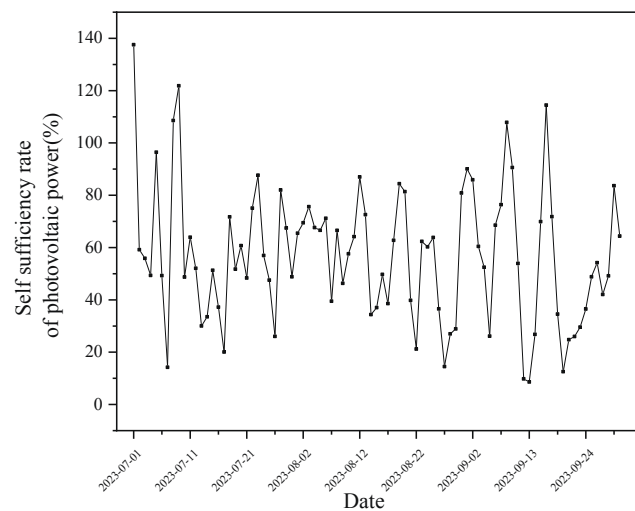


Figure 6. Electricity self-sufficiency rate after implementation of demand-side management adjustments.

point is adjusted from 25°C to 27°C. Overall, this adjustment led to a 15.1% reduction in the electrical energy consumption of the air conditioning system, thereby decreasing the total electricity consumption of the park and improving the self-sufficiency rate of the PV power generation system.

By implementing this demand-side management measure of adjusting the air conditioning system's set temperature, as shown in Figure 6, the average self-sufficiency rate of PV power generation from July to September increased from 51.27% to 56.85%, as compared to the data in Figure 3. The number of days with a self-sufficiency rate exceeding 50% increased from 43 days to 52 days, with the number of days within the range of 50% to 100% rising from 39 to 47 days. This significantly enhanced the proportion of renewable energy in the power supply, thereby reducing carbon emissions from traditional non-renewable electricity sources.

Carbon Flux Monitoring

One of the key objectives of this study was to evaluate the impact of renewable energy deployment on the park's carbon footprint. To achieve this, a carbon flux tower using was set up to monitor the dynamic changes in carbon flux and sequestration in real-time.

Figure 7 presents the daily average CO₂ concentrations recorded by the CO₂ flux tower during the period from October 1 to November 25, 2024. The data revealed that the CO₂ concentration remains at a relatively low level from 428–492 PPM, with an average value of 464.87 PPM. This suggests that the use of renewable energy for power generation has effectively suppressed CO₂ emissions within the park.

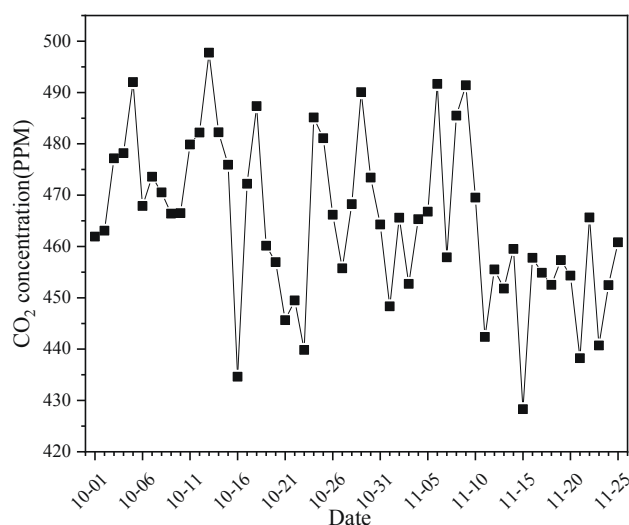


Figure 7. Daily average carbon dioxide concentration monitored by carbon flux tower.

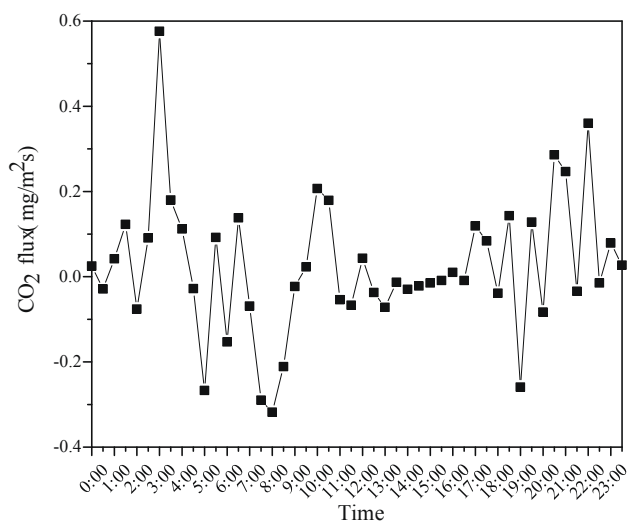


Figure 9. Hourly variation of carbon dioxide flux in a working day

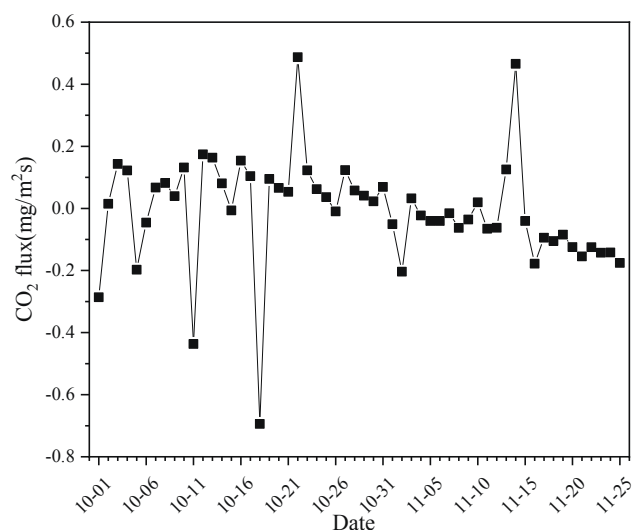


Figure 8. Daily average CO₂ flux monitored by carbon flux tower.

Figure 8 illustrates the variations in CO₂ flux observed from October 1 to November 25, where negative values indicate absorption and positive values represent emissions. As shown in the figure, the flux values are relatively small, with an average of $-0.0087 \text{ mg/m}^2\text{s}$, indicating that the park has effectively achieved carbon neutrality. The data highlight the critical role of green vegetation within the site in sequestering carbon, demonstrating their effectiveness in absorbing CO₂.

Figure 9 depicts the variation in CO₂ flux over a 24-hour period on October 30, a typical workday, with data

recorded at 30-minute intervals. As shown in the figure, the hourly changes in CO₂ flux exhibit no significant patterns, with values fluctuating around zero. This indicates a minimal net CO₂ emission, reflecting a favorable carbon reduction effect.

Carbon Sink

To assess carbon sequestration potential in the park, a detailed analysis of carbon sink characteristics for different areas was performed. The research not only assessed the average biomass and carbon bio-sequestration rates of each tree species but also focused on the different forest types present throughout and within the park. The potential of carbon sequestration for individual zones was determined using aboveground and belowground biomass data, forest age, and area-related data (Table 3).

The results show the potential for carbon sequestration varies greatly between the areas, depending on tree species, forest density and age. In a comparison of the Mapped Species, advanced categories of species like Pine Forest, Maple Grove show greater potential for biomass and sequestration than younger plantations like Cherry Trees in East Corner. The Camphor Tree Groves also further contribute to the park's total carbon sink on average because of their general distribution.

Notably all 12 zones of the park had a wide biomass range from the lowest 1.35 tons in East Corner to the highest 7.00 tons in Garden Pond. Likewise for carbon sequestered until 2024, the range is 1.80 tCO_2 – 9.10 tCO_2 , implying the dominant role of vegetation in carbon emission reduction. All green zones have carbon sequestration potential, which, over the next decade, is expected to increase by a factor of about 4.

Table 3. Data analysis of carbon sink for different areas in the park

Area	Type	Tree Species	Forest Age	Area (sqm)	Above-Ground Biomass (kg)	Below-Ground Biomass (kg)	Biomass (tons)	Carbon Sequestration Potential till 2024 (tCO ₂)	Annual Change in Carbon Sequestration Potential (10-Year Forecast) (tCO ₂)
In front of Courtyard	Arbor	Camphor Tree 1	5	100	1050	450	1.5	2.2	0.35
Behind Laboratory	Arbor	Camphor Tree 2	7	150	1650	650	2.3	3.1	0.4
Side of Experimental Building	Arbor	Camphor Tree 3	6	120	1350	550	1.9	2.5	0.38
Rear Garden	Arbor	Bamboo Grove	8	200	2500	100	3.5	4.8	0.6
Central Lawn	Arbor	Camphor Tree 4	10	300	3600	1400	5	6.7	0.8
West Pavilion	Arbor	Chinese Fir	4	250	2200	900	3.1	3.8	0.55
East Hill	Arbor	Pine Forest	12	350	4500	1800	6.3	7.9	1
South Entrance	Arbor	Poplar Grove	8	280	3000	1200	4.2	5.5	0.7
North Pathway	Arbor	Willow Trees	6	180	1600	700	2.3	2.9	0.45
Garden Pond	Arbor	Maple Grove	9	400	5000	2000	7	9.1	1.2
Training Ground	Arbor	Eucalyptus	11	320	4100	1600	5.7	7.3	0.9
East Corner	Arbor	Cherry Trees	3	90	950	400	1.35	1.8	0.25

CONCLUSION

This study conducted an applied investigation into the regulation control measures and carbon reduction outcomes of a research and office institute park in Nanjing, China, following the implementation of renewable energy systems. The key conclusions are as follows:

The deployment of a 1.162 MW photovoltaic power system significantly contributed to the park's electricity supply. From April to December, the system achieved a daily electricity self-sufficiency rate exceeding 100% on 50 days and surpassed 50% on 58.8% of the days. The average self-sufficiency rate during this period was 69.75%. Seasonal fluctuations were observed, with lower self-sufficiency rates during summer and winter due to increased energy demand and reduced solar radiation. On an annual basis, the photovoltaic system was estimated to generate approximately 1.062 million kWh, covering a substantial

portion of the park's total electricity consumption of 1.689 million kWh.

A demand-side management strategy was implemented by increasing the air conditioning temperature set point from 25 °C to 27 °C. This measure led to a 15.1% reduction in energy consumption related to cooling and increased the average summer photovoltaic self-sufficiency rate (July to September) from 51.27% to 56.85%. This improvement enhanced the utilization of renewable energy while reducing reliance on conventional energy sources.

A carbon flux tower was installed to monitor real-time carbon flux and sequestration within the park. Data from the tower showed consistently low CO₂ concentration and a negative carbon flux, with average values of 464.87 ppm and −0.0087 mg/m²·s, respectively. These results indicate a measurable carbon reduction effect and highlight the effectiveness of combining renewable energy systems with

demand-side management and environmental monitoring in low-carbon campus development.

While the current system has been successful in improving energy efficiency and reducing carbon emissions, several opportunities for further optimization exist. Future directions include:

- Expanding energy storage: Increasing the storage capacity of the system would provide more reliable backup power during periods of low solar generation, further reducing dependence on the grid, especially when solar generation fluctuates due to weather conditions or time of day.
- Integrating additional renewable sources: Exploring the integration of other renewable energy sources, such as wind or geothermal energy, could diversify the park's energy mix and improve its resilience to fluctuations in solar power generation.
- Enhancing carbon flux monitoring: Expanding the network of carbon flux towers to monitor different environmental variables, such as soil carbon flux and plant respiration, would provide a more comprehensive understanding of the park's overall carbon dynamics.

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article

REFERENCES

- [1] Akkoyunlu MT, Abdallatif YA. Comprehensive investigation of solar panel cleaning technologies: A review study. *J. Thermal Engineering* 2024;10:1715-41. [\[Crossref\]](#)
- [2] Jiang D, Gao H, He S, Huang Y, Liu J. A Negative-Carbon Planning Method for Agricultural Rural IndustrialParkIntegratedEnergySystemConsidering Biomass Energy and Modern Agricultural Facilities. *J Clean Prod* 2024;479:143837. [\[Crossref\]](#)
- [3] Huang B, Jiang P, Wang S, Zhao J, Wu L. Low Carbon Innovation and Practice in Caohejing High-Tech Industrial Park of Shanghai. *Int J Prod Econ* 2016;181:367-373. [\[Crossref\]](#)
- [4] Wu D, Han S, Wang L, Li G, Guo J. Multi-Parameter Optimization Design Method for Energy System in Low-Carbon Park with Integrated Hybrid Energy Storage. *Energy Convers Manag* 2023;291:117265. [\[Crossref\]](#)
- [5] Li T, Wang X, Yu Y, Fu Q, Chen M, Xu C, et al. Mao, Q. Performance and PV Benefits Analysis of Multi-Source Renewable Energy Systems for Different Types of Buildings on University Campus. *Renew. Energy* 2024;237:121522. [\[Crossref\]](#)
- [6] Aghamolaei R, Fallahpour M. Strategies towards Reducing Carbon Emission in University Campuses: A Comprehensive Review of Both Global and Local Scales. *J Build Eng* 2023;107183. [\[Crossref\]](#)
- [7] Hiltunen P, Volkova A, Latšov E, Lepiksaar K, Syri S. Transition towards University Campus Carbon Neutrality by Connecting to City District Heating Network. *Energy Reports* 2022;8:9493-505. [\[Crossref\]](#)
- [8] Guerrieri M, La Gennusa M, Peri G, Rizzo G, Scaccianoce G. University Campuses as Small-Scale Models of Cities: Quantitative Assessment of a Low Carbon Transition Path *Renew Sustain Energy Rev* 2019;113:109263. [\[Crossref\]](#)
- [9] Li Z, Ge Z, Wang C, Zhang D, Foo DCY, Liu G, Shi L. An Automated Approach for Carbon Integration in Industrial Park. *Process Saf Environ Prot* 2024;190:677-87. [\[Crossref\]](#)
- [10] Zhao Y, Wang S, Gao G, Xue X, Song H, Zhang, R. Exploring the Green and Low-Carbon Development Pathway for an Energy-Intensive Industrial Park in China. *J Clean Prod* 2024;459:142384. [\[Crossref\]](#)
- [11] Qian L, Lin S, Li F, Wang W, Li D. Low Carbon Optimization Dispatching of Energy Intensive Industrial Park Based on Adaptive Stepped Demand Response Incentive Mechanism *Electr Power Syst Res* 2024;233:110504. [\[Crossref\]](#)
- [12] Lyu X, Liu T, Liu X, He C, Nan , Zeng H. Low-Carbon Robust Economic Dispatch of Park-Level Integrated Energy System Considering Price-Based Demand Response and Vehicle-to-Grid. *Energy* 2023;263:125739. [\[Crossref\]](#)
- [13] Ho YF, Chang CC, Wei CC, Wang HL. Multi-Objective Programming Model for Energy Conservation and Renewable Energy Structure of a Low Carbon Campus. *Energy Build* 2014;80:461-8. [\[Crossref\]](#)
- [14] Li K, Yang R, He X. Realizing Low-Carbon Development of Industrial Parks in China: Model Construction and Its Application. *Energy* 2024;301:131664. [\[Crossref\]](#)

- [15] Hu Y, Yang B, Wu P, Wang X, Li J, Huang Y, et al. Optimal Planning of Electric-Heating Integrated Energy System in Low-Carbon Park with Energy Storage System. *J. Energy Storage* 2024;99:113327. [\[Crossref\]](#)
- [16] Huo Q, Liu Q, Deng H, Wang W, Shi C, Wei T. Research on Dual-Layer Optimization Strategy of Photovoltaic-Storage-Hydrogen System in Coal Chemical Industry Park. *Renew. Energy*; 2024: 120813. [\[Crossref\]](#)
- [17] Zheng N, Li S, Wang Y, Huang Y, Bartocci P, Fantozzi R, et al. Research on Low-Carbon Campus Based on Ecological Footprint Evaluation and Machine Learning: A Case Study in China *J Clean Prod* 2021;323:129181. [\[Crossref\]](#)
- [18] Wang Y, Zhang X, Zhu L, Wang X, Zhou L, Yu X. Synergetic Effect Evaluation of Pollution and Carbon Emissions in an Industrial Park: An Environmental Impact Perspective. *J Clean Prod* 2024;142891. [\[Crossref\]](#)
- [19] Wang C, Parvez AM, Mou J, Quan C, Wang J, Zheng Y, et al. The Status and Improvement Opportunities towards Carbon Neutrality of a University Campus in China: A Case Study on Energy Transition and Innovation Perspectives. *J Clean Prod* 2023;414:137521. [\[Crossref\]](#)
- [20] Zhou W, Zhuang G, Liu L. Comprehensive Assessment of Energy Supply-Side and Demand-Side Coordination on Pathways to Carbon Neutrality of the Yangtze River Delta in China. *J Clean Prod* 2023;404:136904. [\[Crossref\]](#)
- [21] Raza MA, Aman MM, Kumar L, Al-Khasawneh MA, Fahemm M, Ehyaei MA. Carbon Neutrality and Economic Stability Nexus: An Integrated Renewable Energy Transition to Decarbonize the Energy Sector. *Energy Rep* 2025;13:4586-08. [\[Crossref\]](#)
- [22] Zhang W, Cuijing J, He P, Wuhao E. Examining the Impact of Tourism on Carbon Neutrality and Environmental Sustainability in China: The Role of Renewable Energy. *Energy Strateg Rev* 2024;56:101579. [\[Crossref\]](#)
- [23] Wei H, Zhang N, Du E, Jiang H, Zhuo Z, Davidson MR et al. Emerging Demand-Side Flexible Resources Accelerate China's Power System Transition toward Carbon Neutrality. *iScience* 2025;28:112372. [\[Crossref\]](#)
- [24] Shafiei K, Seifi A, Hagh MTA. Novel Multi-Objective Optimization Approach for Resilience Enhancement Considering Integrated Energy Systems with Renewable Energy, Energy Storage, Energy Sharing, and Demand-Side Management. *J Energy Storage* 2025;115:115966. [\[Crossref\]](#)
- [25] Lu Q, Fang H, Hou J. The Impact of Energy Supply Side on the Diffusion of Low-Carbon Transformation on Energy Demand Side under Low-Carbon Policies in China. *Energy* 2024;307:132817. [\[Crossref\]](#)
- [26] Chau KY, Sadiq M, Chien F. The Role of Natural Resources and Eco-Financing in Producing Renewable Energy and Carbon Neutrality: Evidence from Ten Asian Countries. *Resour. Policy* 2023;85:103846. [\[Crossref\]](#)



Review Article

Novel approaches to zeolite deactivation mitigation and regeneration in biomass gasification

Imron ROSYADI¹, Suyitno^{1,*}, Zainal ARIFIN¹, Tata SUTARDI²

¹Department of Mechanical Engineering, Faculty of Engineering, Universitas Sebelas Maret (UNS), Central Java, 57126, Indonesia

²Research Centre for Energy Conversion and Conservation, BRIN, 10110, Indonesia

ARTICLE INFO

Article history

Received 18 February 2024

Revised: 24 June 2024

Accepted 24 September 2024

Keywords:

Biomass Gasification; Catalyst Regeneration; Deactivation Mechanisms; A Mitigation; Environmental Applications; Zeolite Catalys

ABSTRACT

The comprehensive study delves into the intricate realm of zeolite catalyst deactivation within the context of biomass gasification, aiming to provide a thorough understanding of deactivation mechanisms, innovative strategies for mitigation and regeneration, and potential applications for deactivated zeolites. Key findings reveal reversible and irreversible deactivation processes, heavily influenced by physical and chemical interactions with contaminants such as nitrogen, sulfur, and heavy metals. Empirical data-driven mitigation strategies showcase the effectiveness of metal modifiers like nickel and cobalt in reducing coke yield, alongside novel approaches such as core-shell zeolite structures and the integration of redox metal oxides to maintain catalyst basicity and stability. Moreover, successful regeneration methods including thermal regeneration, chemical washing, and steaming demonstrate the restoration of catalytic activity post-deactivation. Despite reduced efficiency, deactivated zeolites exhibit promises in environmental remediation, achieving heavy metal removal efficiencies surpassing 90%, and enhancing durability while reducing permeability in construction materials and concrete additives. Furthermore, the review emphasizes the necessity for refined strategies adaptable to diverse conditions, promoting sustainable catalyst utilization in biomass gasification and beyond. Key contributions highlighted include the identification of deactivation processes, recognition of pivotal factors affecting zeolite catalysts, validation of data-driven mitigation strategies, demonstration of novel approaches, successful application of regeneration methods, and exploration of potential applications for deactivated zeolites. These findings signify significant progress in addressing zeolite catalyst deactivation mitigation and regeneration challenges and enhancing efficiency and sustainability in biomass gasification technologies.

Cite this article as: Rosyadi I, Suyitno, Arifin Z, Sutardi T. Novel approaches to zeolite deactivation mitigation and regeneration in biomass gasification. J Ther Eng 2025;11(5):1552–1584.

*Corresponding author.

*E-mail address: suyitno@staff.uns.ac.id

This paper was recommended for publication in revised form by
Editor-in-Chief Ahmet Selim Dalkılıç



INTRODUCTION

The catalysts, which actively participates in accelerating reactions without being consumed, play a fundamental role in the chemical industry by enabling efficient production of various chemicals and fuels [1–3]. Among these, zeolite catalysts have gained prominence due to their unique micro-porous structure and tunable acidity, which facilitate versatile applications by reducing activation energy and allowing more particles to engage in effective reactions [4–6]. The micro-porous structure, consisting of uniformly sized and shaped pores, allows for the selective adsorption and interaction with specific molecules, thereby facilitating targeted catalytic reactions [7]. The concept of activation energy (E_a), which denotes the minimum energy necessary to initiate a chemical reaction, is pivotal in this context. Zeolites facilitate the reduction of activation energy by strategically orienting reacting particles to enhance the probability of collisions and by forming intermediate compounds with reactants that necessitate lower energy for the production of desired products [8]. Additionally, the acidity of zeolite catalysts is adjustable, a property known as ‘tunable acidity,’ which allows for the optimization of catalytic activity and selectivity. This tunability is achieved by modifying the concentration and strength of acidic sites through synthesis and post-synthesis treatments [9]. Zeolites contain both Brønsted and Lewis acid sites, which act as catalyst centers. The strength of Brønsted acid sites is influenced by the Si/Al ratio of the zeolite, and achieving the optimal ratio can reduce catalyst deactivation [10]. Effective methods for adjusting the Si/Al ratio, such as dealumination and desilication, optimize zeolite properties and enhance catalytic performance in various chemical reactions [11–16].

As a result, the catalyzed reactions surpass normal reactions and benefit from lower activation energy requirements. However, the long-term performance of zeolite catalysts is often hampered by deactivation, which leads to a decline in their catalytic activity and selectivity. The motivation behind this work lies in addressing the critical challenges associated with zeolite catalyst deactivation in biomass gasification processes. These challenges include the need to prolong catalyst lifespan, reduce operational costs, and maintain high conversion efficiency. By comprehensively understanding deactivation mechanisms and developing innovative mitigation and regeneration strategies, the aim is to enhance the sustainability and efficiency of biomass gasification technologies. Moreover, exploring potential applications for deactivated zeolites underscores the broader goal of sustainable catalyst utilization, not only in biomass gasification but also in various environmental and construction-related applications.

The choice of the catalyst support material significantly affects zeolite catalyst deactivation in biomass conversion processes [17]. Studies indicate that catalysts supported by metal oxides such as CeO_2 , MgO , TiO_2 , ZrO_2 , Al_2O_3 , and others exhibit distinct reactivities [18]. However, stability

of the support material is crucial. For example, Al_2O_3 , an economical support, is prone to deactivation, particularly when Ni is used as the active metal [19,20]. Dolomite or $\text{MgCO}_3 \cdot \text{CaCO}_3$ is a cost-effective support that effectively reduces tar content in syngas during biomass gasification, showing promising results [21]. Zeolites, whether natural or synthetic, possess great potential as metal carriers in biomass conversion and hydrocarbon reforming because of their high acidity, large surface area, thermal stability, and remarkable ability to crack macromolecules [22–25]. However, zeolite catalysts are also susceptible to deactivation. Understanding the mechanisms of zeolite catalyst deactivation and selecting appropriate support materials, such as stable metal oxides or dolomite, are crucial for enhancing catalyst performance and stability during biomass conversion processes.

In recent studies, zeolite catalyst deactivation challenges within the realm of biomass gasification have been extensively addressed. Mei et al. (2023) explored the use of zeolites for alkali recovery from catalytic gasification ash (CGA), successfully transforming CGA into zeolite through hydrothermal processing [26]. However, the specific mechanisms and factors leading to zeolite deactivation in this process require further exploration. Similarly, Porawati et al. (2021) investigated the impact of temperature on zeolite-catalyzed biomass gasification, offering valuable insights into gasifier optimization, yet overlooking a thorough examination of zeolite deactivation at different temperatures [27]. The temperature-dependent mechanisms influencing zeolite deactivation are crucial for designing resilient catalysts across diverse operating conditions. Furthermore, State et al.’s (2019) review on zeolite catalysts in microwave-assisted gasification recognized deactivation challenges [28] but lacked a comprehensive understanding of the specific deactivation conditions or mechanisms. Exploring the intricacies of zeolite deactivation in microwave-assisted processes is vital for advancing catalyst design and ensuring prolonged stability. Lastly, Zhang et al. (2018) delved into Ni-based catalysts for biomass tar reforming, highlighting challenges related to deactivation [29]. However, the study missed exploring innovative strategies to overcome these deactivation hurdles, leaving a gap in identifying modifications for enhancing the resistance of Ni-based catalysts to ensure prolonged catalytic activity.

To examine the research trends in the deactivation of natural zeolite catalysts in the last decade (2013–2023), searches were conducted on various research platforms such as Google Scholar and www.sciencedirect.com. Selecting both sources for reviews involves justifying their use due to broad coverage and multidisciplinary content. Both platforms offer extensive academic literature access, including curated databases and freely accessible search engines, ensuring a comprehensive overview. Their widespread use and Google Scholar’s open access enhance accessibility, aiding researchers without institutional database access. Several keywords, including deactivation, zeolite, natural

zeolite, gasification, pyrolysis, syngas, and biofuel, were used. As shown in Table 1, searches were performed using these keywords in all fields of articles and all titles of articles on Google Scholar. Meanwhile, in ScienceDirect, the search encompassed all articles, titles, abstracts, and keywords. Table 1 provides an overview of the research trend in articles related to zeolite deactivation in biomass gasification over the last decade (2013–2023). The data, sourced from Google Scholar and ScienceDirect, presents the number of articles across various keywords and platforms.

Zeolite Gasification has garnered significant attention with 17,200 articles in Google Scholar and 6,332 in ScienceDirect. This indicates a substantial interest in the utilization of zeolites in the gasification process. Similarly, Zeolite Pyrolysis and Zeolite Biofuel exhibit substantial numbers, showcasing the diverse applications of zeolites in these domains. Moreover, the focus on Zeolite Deactivation is evident with 17,700 articles in Google Scholar and 13,304 in ScienceDirect. This highlights the recognition of challenges associated with zeolite deactivation in both gasification and pyrolysis processes. Interestingly, while Zeolite Deactivation Pyrolysis has a significant presence in Google Scholar, it is relatively lower in ScienceDirect, indicating a potential gap or variation in the coverage of the topic between the platforms.

Natural Zeolite categories also demonstrate considerable interest, with Natural Zeolite Gasification and Natural Zeolite Pyrolysis leading in article count. The emphasis on Natural Zeolite Deactivation further underscores the significance of understanding deactivation challenges in the context of natural zeolites. It is apparent that while there is a substantial volume of research on zeolite applications in gasification, pyrolysis, and biofuel production, the specific focus on zeolite deactivation is comparably limited.

This suggests an avenue for more targeted research on the challenges and mitigation strategies associated with zeolite deactivation in the context of biomass gasification.

Therefore, the current investigation aims to comprehensively explore the complexities associated with zeolite catalyst deactivation within the framework of biomass gasification. The primary objectives of this study encompass deciphering the underlying mechanisms and pivotal factors that contribute to zeolite deactivation during biomass gasification, formulating innovative strategies to mitigate deactivation and regeneration issues, and identifying potential applications for deactivated zeolites. The pressing need to pursue these research goals is underscored by existing knowledge gaps that hinder the optimal utilization of zeolite catalysts in biomass gasification processes. Effectively addressing these challenges is not only imperative for enhancing the effectiveness and sustainability of biomass conversion technologies but also presents an opportunity to contribute to the principles of a circular economy.

ROLE OF ZEOLITES IN BIOMASS GASIFICATION

Natural and Synthetic Zeolites

Natural zeolites are minerals that form over geological timescales through the interaction of volcanic ash and groundwater [30]. They are often found in deposits around volcanic regions. In contrast, industrial or synthetic zeolites are typically synthesized in laboratories or manufacturing facilities using various techniques, such as hydrothermal synthesis and alkali fusion, to achieve specific compositions and structures [31, 32]. Natural zeolites and synthetic zeolites represent distinct categories of crystalline materials,

Table 1. The research trend on articles concerning zeolite deactivation in the last decade (2013–2023)

No	Keywords	Google scholar	Google scholar	ScienceDirect	ScienceDirect
		All fields	Filtered by Titles	All fields	Filtered by Titles
1	Zeolite Gasification	17,200	31	6,332	10
2	Zeolite Pyrolysis	22,000	380	13,787	7
3	Zeolite Biofuel	16,900	54	218	4
4	Zeolite Deactivation	17,700	120	13,304	10
5	Zeolite Deactivation Pyrolysis	16,000	10	3,862	7
6	Zeolite Deactivation gasification	10,800	0	2,351	10
7	Zeolite Deactivation Biofuel	13,600	0	2,331	4
8	Natural Zeolite Gasification	17,100	3	4,326	11
9	Natural Zeolite Pyrolysis	18,100	35	7,869	5
10	Natural Zeolite Biofuel	16,600	5	4,024	2
11	Natural Zeolite Deactivation	17,300	0	5,642	6
12	Natural Zeolite Deactivation Pyrolysis	15,700	0	2,035	5
13	Natural Zeolite Deactivation gasification	10,600	0	1,577	6
14	Natural Zeolite Deactivation Biofuel	15,500	0	1,341	2

each with unique characteristics and applications. Natural zeolites, such as clinoptilolite, mordenite, heulandite, analcime, chabazite, natrolite, phillipsite, and stilbite originate from geological processes involving volcanic ash and groundwater [33, 34]. Their compositions can vary significantly, introducing heterogeneity in terms of structure and impurities. In contrast, synthetic zeolites like Zeolite A, Zeolite X, Zeolite Y, Beta Zeolite, Faujasite zeolite and ZSM-5 are intentionally manufactured through controlled laboratory processes, offering precise control over composition and structure [35–37].

One key difference lies in their origins and compositions. Natural zeolites are minerals formed over geological timescales, exhibiting inherent variability and impurities. Industrial applications of natural zeolites often face challenges due to this variability. On the other hand, synthetic zeolites are precisely engineered in laboratories, resulting in consistent compositions and structures. This controlled synthesis allows for tailoring zeolites to specific applications, such as catalysis and adsorption. Catalytic performance is another distinguishing factor. While natural zeolites may possess inherent catalytic activity, it is often less predictable compared to their synthetic counterparts. Synthetic zeolites, designed with specific catalytic properties, offer more reliability and consistency in performance. Additionally, the limitations of natural zeolites, including impurity variability and a lack of control over composition, make them less suitable for applications demanding precision.

Background on Biomass Gasification

Biomass gasification is a thermochemical process that transforms biomass feedstock into gaseous products by undergoing a series of chemical reactions with a limited supply of air [38–40, 41]. The process occurs under high temperatures and limited oxygen conditions, resulting in the production of syngas, consisting of hydrogen (H_2), carbon monoxide (CO), carbon dioxide (CO_2), methane (CH_4), and other trace components. Syngas is a valuable intermediate that can be further processed into various energy carriers, such as hydrogen or synthetic fuels [41, 42]. Catalysts play a crucial role in biomass gasification by enhancing reaction rates, improving selectivity, and mitigating undesired byproducts [17]. Zeolite catalysts have shown promise for biomass gasification owing to their high surface area, uniform pore structure, and tunable acidity [43]. Zeolites are crystalline microporous materials composed of aluminum, silicon, and oxygen atoms arranged in a regular framework. During biomass gasification, zeolites promote the conversion of complex biomass molecules into smaller and more reactive intermediates [27]. These intermediates undergo further reactions to produce the desired syngas components. The key chemical reactions involved in biomass gasification are as follows:

- **Drying and devolatilization.** Biomass is first dried to remove moisture, followed by devolatilization, where

high temperatures cause the release of volatile compounds (hydrocarbons and tar) from the biomass.

- **Pyrolysis.** Pyrolysis is the thermal decomposition of biomass in the absence of oxygen. This leads to the production of gases, such as methane (CH_4), carbon monoxide (CO), hydrogen (H_2), and small amounts of higher hydrocarbons. The general equation for pyrolysis is as follows:



- **Gasification.** Gasification occurs in the presence of a gasifying agent, typically steam (H_2O) or a mixture of steam and oxygen (O_2). The primary reactions during gasification are as follows:

- a. **Water-Gas Shift Reaction:**



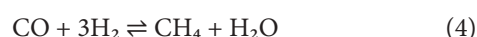
This reaction involves the conversion of carbon monoxide (CO) and water vapor (H_2O) into carbon dioxide (CO_2) and hydrogen (H_2). The water-gas shift reaction helps increase the hydrogen content in the syngas.

- b. **Boudouard Reaction:**



The Boudouard reaction is an equilibrium reaction in which carbon monoxide (CO) reacts to produce carbon dioxide (CO_2) and additional carbon. This reaction leads to the formation of solid carbon (char) during gasification.

- c. **Methanation:**



Methanation is the reaction in which carbon monoxide (CO) and hydrogen (H_2) combine to form methane (CH_4) and water (H_2O). Methanation is desirable for biomass gasification to increase the methane content of syngas.

- **Tar Cracking.** The tar compounds produced during gasification can be problematic and lead to fouling and corrosion [40]. Tar cracking involves the thermal decomposition, catalytic cracking, plasma cracking, and physical separation of tar molecules into smaller and less complex compounds [44].

In biomass gasification with zeolite as a catalyst, the process initiates with pyrolysis, breaking down biomass into volatile compounds and char. Zeolites, whether of natural or synthetic origin and functioning as catalysts, play a role in promoting the cracking of tar, transforming undesired tar. The natural zeolite displayed enhanced tars conversion, peaking at 79% at 900 °C, and facilitated a catalytic tar conversion with a lower activation energy (62.1 kJ/mol) compared to the thermal cracking counterpart (85.0 kJ/mol), showcasing the catalytic benefits of the natural zeolite while concurrently lowering activation energies and revealing distinct frequency factors [45]. During steam gasification of char, zeolites enhance the production of hydrogen

and carbon monoxide. Further, zeolites catalyze methane reforming, contributing to the synthesis of carbon monoxide and hydrogen. The water-gas shift reaction (equation 2), facilitated by zeolites, increases hydrogen production by reacting carbon monoxide with water vapor. Additionally, zeolites play a crucial role in adsorption and desorption processes, aiding in the purification and separation of gas components.

It cannot be definitively stated that the gasification reaction mechanisms between natural zeolite and synthetic zeolite are fundamentally identical. While both can act as catalysts in biomass gasification, substantial differences in structure, composition, and catalytic properties can lead to notable variations in their reaction pathways. For example, in the case of natural zeolite, an aluminosilicate mineral with inherent heterogeneity, consider the steam gasification of char ($\text{Char} + \text{H}_2\text{O} (\text{steam}) \rightarrow \text{CO} + \text{H}_2$). The varied surface sites of natural zeolite may influence the adsorption and desorption of water vapor, affecting the overall gasification process. Conversely, synthetic zeolite, engineered with controlled structures, may exhibit enhanced catalytic activity in methane reforming ($\text{CH}_4 + \text{Zeolite} \rightarrow \text{C} + 2\text{H}_2$), due to its precisely tuned acidic sites. Therefore, while certain aspects of the reaction mechanisms may share similarities, the differences could manifest in the kinetics and selectivity of specific reactions.

Significance of Catalysts in Biomass Gasification

In biomass gasification, catalysts play a significant role in enhancing process parameters and improving the overall efficiency of the gasification process. Zeolites are crystalline aluminosilicate materials with porous structures, and their unique properties make them suitable for various catalytic applications, including biomass gasification [46]. Figure 1 illustrates the role of zeolite catalysts in biomass gasification. The figure was compiled using data and illustrations adapted from references [47–55], and presents a schematic overview of key processes, including tar cracking, reforming reactions, catalytic layer stability, selectivity control, and enhanced hydrogen production.

Biomass gasification holds immense promise for sustainable energy production, and the integration of zeolite-based catalysts enhances the efficiency and effectiveness of this technology. Various biomass feedstocks, including palm kernel shell (PKS), sugarcane bagasse (SB), and Casuarina wood chips, have been subjected to gasification with different zeolite catalysts, showcasing diverse applications and outcomes [27, 56–59] as shown in Table 2.

The application of modified mordenite-zeolite with acid in PKS gasification demonstrated a remarkable reduction in tar content by up to 98%, resulting in a hydrogen-rich syngas [56]. Similarly, natural zeolites in Sugarcane Bagasse gasification substantially increased combustible gas ratios, improving key gas concentrations [27]. Various Ni-based catalysts in Maize gasification exhibited temperature-dependent effects on syngas composition, emphasizing the

importance of catalyst selection [60]. By effectively cracking tar and reforming complex organic compounds, zeolites contribute to improved gas quality by reducing tar concentrations in the syngas [53, 54]. The catalytic activity of zeolites in reforming reactions, such as steam reforming and carbon gasification, enables the conversion of biomass components into valuable syngas components such as CO and H_2 . In biomass tar reforming, the catalyst with Ni supported on ZSM-5 and a Si/Al ratio of 80 demonstrated the highest level of activity at a temperature of 650 °C [50]. The introduction of steam into the system resulted in a syngas yield of up to 80.5 mmol/g. However, achieving a Si/Al ratio of 80 in natural zeolites remains challenging [50]. Nevertheless, the porous structure of zeolites further enhances their conversion efficiency by providing a large surface area for efficient catalytic reactions. The optimal catalytic performance for syngas production and tar elimination from rice straw gasification was achieved at a catalytic temperature of 250 °C with the prepared Ni-based zeolite catalyst, yielding 5.92 MJ/kg a lower heating value (LHV) and 73.9% tar removal efficiency [55].

Moreover, the studies on rice straw, heavy metal-contaminated biomass (HMCB), poplar wood, and wood chips further contribute to our understanding of zeolite catalysts' versatility in different contexts [55, [61–67]. Zeolite 13X in Casuarina wood chips gasification generated high-purity hydrogen, highlighting its potential for clean energy applications [59]. Clinoptilolite natural zeolite in lignocellulosic biomass gasification significantly decreased tar and CO_2 concentration, leading to higher inflammable gas content [68]. The varying operating conditions and catalyst compositions explored in these studies emphasize the nuanced nature of biomass gasification and the need for tailored solutions. These findings collectively underscore the pivotal role of zeolite-based catalysts in optimizing biomass gasification processes.

Furthermore, zeolite thermal stability and resistance to coke deposition are highly beneficial in biomass gasification processes because coke formation on catalyst surfaces can diminish their activity and lifespan. The robust nature of zeolites allows them to maintain their catalytic activity over an extended period, ensuring the stability and longevity of the gasification process. Thermodynamic modelling has revealed that factors such as cation abundance, silica activity, and temperature influence zeolite stability during gasification [69]. Siliceous zeolites, such as clinoptilolite, mordenite, and erionite, are favored in environments with elevated silica activity, whereas aluminous zeolites such as natrolite, mesolite, scolecite, and thomsonite are favored in reduced silica environments [69]. Gasification conditions, including temperature and aqueous composition, further impact stability [70, 71]. By leveraging zeolite stability, gasification processes can be optimized to improve the efficiency and high-quality syngas production.

Moreover, zeolites provide the ability to control selectivity, allowing the production of specific syngas components

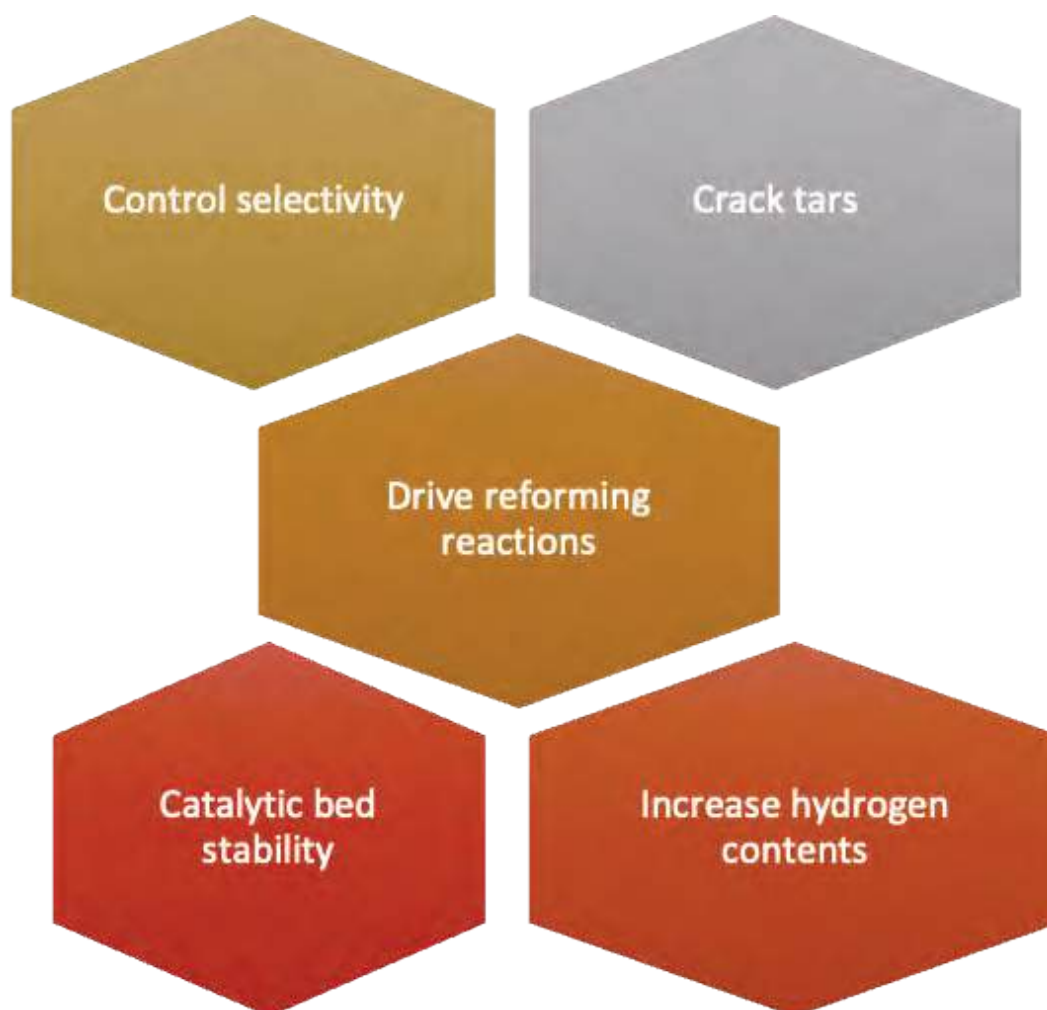


Figure 1. General role of zeolite catalysts in biomass gasification.

as desired [52]. By selectively choosing zeolite catalysts with tailored properties, the gasification process can be optimized to achieve the desired syngas composition for various applications. This versatility opens up new possibilities for power generation, biofuel production, and chemical synthesis. Moreover, the catalyst design principle of the oxide-zeolite-based composite (OXZEO) allows for the direct conversion of syngas into mixed light olefins, achieving an impressive 80% selectivity among hydrocarbons [51].

Challenges of Zeolites in Biomass Gasification

Natural zeolites can exhibit variations in their composition, Si/Al ratio, crystal properties, acidity, and coke deposition depending on their type as shown in Table 3. The challenges associated with using natural zeolite catalysts in biomass gasification include variability in composition, catalyst deactivation, pore size limitations, pore blockage, coke formation, tar deposition, dealumination, thermal stability, and advances in natural zeolite development as shown in Table 4. These variations can affect the performance and

consistency of the catalysts in biomass gasification processes. The differing compositions may affect the ability of the catalyst to precisely control the reaction conditions and product quality, making it challenging to achieve consistent results.

Moreover, biomass feedstocks often contain impurities, such as ash, sulfur, and alkali metals [74, 75]. These impurities can accumulate on the catalyst surface during gasification resulting in catalyst deactivation [45, 76]. Ash deposits can block active sites and reduce catalytic activity, whereas sulfur compounds and alkali metals can poison the catalyst. Moreover, biomass gasification involves complex reactions and harsh operating conditions, that can lead to catalyst deactivation over time. Although natural zeolites are relatively stable, they can still undergo deactivation due to factors such as coke deposition, sintering, and poisoning by impurities present in biomass feedstocks. Additionally, different biomass feedstocks have different chemical compositions and molecular sizes. Natural zeolites may have limitations in terms of their pore size and shape, which can

Table 2. Summary of Catalysts and Operating Conditions for Biomass Gasification to Syngas using Zeolite-Based Catalysts

Feedstocks	Catalysts	Operating Conditions	Findings	Ref
Biomass: Palm Kernel Shell	- Modified mordenite-zeolite with acid	- Steam Gasification. - Temperature varying between 750 and 850 °C. - The ratio of steam to biomass ranged from 0 to 2.25.	- The syngas hydrogen content ranged from 52-64%. - Acid washing of natural zeolites reduced specific impurities, thus improve efficacy. - Introducing a zeolite catalyst at 750°C with a steam-to-biomass ratio of 2.25 decreased tar content by up to 98%, achieving 0.7 g/Nm ³ .	[56]
Biomass: Sugarcane bagasse	- Natural zeolites	- Air/steam gasification - Temperature varying between 300 and 500 °C. - The ratio of steam to biomass ranged from 0 to 2.25.	- At 500°C, adding 20 mL/minute of water led to notable increases in CO (38.25 mol%), H ₂ (4.01 mol%), and CH ₄ (10.17 mol%). - Zeolite catalysis was pivotal in elevating the combustible gas ratio, with the inclusion of water further enhancing it significantly.	[27]
Biomass: bagasse	- Natural zeolites	-	- The syngas composition: 9.294% for CO, 1.348% for CH ₄ , and 2.773% for H ₂ .	[57]
Biomass	- Synthetic zeolites: Ni/HZSM-5 and Ni-Ca-Co/HZSM-5	- water scrubber and zeolite catalyst as a filter. - downdraft gasifier. - air as gasifying agent.	- The Ni-based catalyst (Ni/HZSM-C-Co5) exhibited strong efficacy in VOC degradation. - Test of various VOCs including toluene, phenol, furan, acetic acid, and cyclohexane revealed notable catalytic activity. - Heating the model compounds to 800°C significantly enhanced their conversion into gases, with over 90% of each compound undergoing transformation, yielding gases such as H ₂ and CH ₄ . - Toluene and cyclohexane decomposition resulted in 93% and 98% conversion into H ₂ and CH ₄ , respectively, at 800°C.	[58]
Biomass	- Natural Jordanian zeolite	- Temperatures: 700 °C, 800 °C, and 900 °C.	- Zeolite surpassed thermal cracking, attaining 79% toluene conversion at 900°C with an activation energy of 62.1 kJ/mol. - Notably, the zeolite catalyst lowered the overall activation energy, exhibiting a significant frequency factor of $3.1754 \times 10^3 \text{ s}^{-1}$ at 800°C.	[45]
Casuarina wood chips	- Zeolite 13X	- Feed streams: oxy-steam and air gasification of casuarina wood chips. - Adsorption pressure: varied. - Feed flow rate: varied. - Gas contaminant analysis: conducted for precise optimization. ion	- A recovery range of 40.7–74.4% for purity >98 mol% (ISO grade A and above). - With a feed of 100 Nm ³ /h, fuel cell compliant hydrogen was generated with an average recovery of $68.4 \pm 3.5\%$.	[59]
Maize	- Ni/ZSM-5 - Ni/CaO - Ni/Al ₂ O ₃ - Ni/Clinoptilolite	- Temperature Range: - 1 st Reactor Zone: 200–800 °C - 2 nd Reactor Zone: 500–700 °C - Regeneration: 800 °C	- Optimal temperatures for the 1 st and 2 nd reactor zones: 400 °C and 700 °C respectively; regeneration: 800 °C. - Ni/ZSM-5: increased lighter hydrocarbons, decreased CO and CO ₂ ; Ni/CaO: declining CO ₂ till 3 rd cycle, reduced CO and H ₂ till 10 th cycle. - Ni/Al ₂ O ₃ : significant CO ₂ and CO reductions each cycle; Ni/Clinoptilolite: decreased H ₂ and CO ₂ , increased CO, enhanced syngas yield. - Ni/Clinoptilolite enhanced syngas and captured CO ₂ even at low pyrolysis temps.	[60]

Table 2. Summary of Catalysts and Operating Conditions for Biomass Gasification to Syngas using Zeolite-Based Catalysts (continued)

Feedstocks	Catalysts	Operating Conditions	Findings	Ref
Lignocellulosic biomass	- Clinoptilolite natural zeolite	- Utilization of Mg/Al-layered double hydroxides clinoptilolite (Mg/Al-LDH@clinoptilolite) - Incorporation of zeolite-layered double hydroxides impregnated with clinoptilolite and Mg/Al-layered double hydroxide into conventional water scrubbers	- Mg/Al-LDH@clinoptilolite was designed to enhance tar removal in lignocellulosic biomass gasification. - Water scrubbers with Zeolite-layered double hydroxides, including clinoptilolite and Mg/Al-LDH, were employed to boost tar removal and improve syngas quality. - A significant decrease in tar and CO₂ concentration, leading to a higher content of inflammable gas. - These findings propose a practical method to enhance power efficiency and durability in lignocellulosic biomass gasification.	[68]
Biomass	- zeolite	- Gasifier Conditions: - Contact Time: 240 minutes - Column Height: 30 cm - Temperature: 450 °C	- Biomass gasification holds promise for renewable energy, but improving yield percentages poses a challenge. - Effective strategies include optimizing gasifier design, employing active catalysts, and understanding decomposition pathways. - At a 240-minute contact time at 450 °C, yielding promising syngas concentrations.	[27]
Rice straw	- Zeolite - Ni-based catalysts	- Hot Gas Temperature Range: 250 °C to 400 °C - Use of Zeolite and Prepared Ni-based Catalysts	- Increasing hot gas temperature and using zeolite or Ni-based catalysts improves energy yield efficiency in rice straw gasification. - Adjusting zeolite catalyst and temperature (250 °C to 400 °C) moderately increases H₂ and CO concentrations, with 70% to 90% tar removal efficiency. - Substituting zeolite with Ni-based catalysts at 250 °C significantly boosts H₂ content, reducing tar and methane. - At 400 °C, deactivated Ni-based catalyst hinders syngas and tar reduction. - Ni-based catalysts at 250 °C exhibit higher stability, activity, and less coke decomposition, achieving optimal performance with 5.92 MJ/kg LHV and 73.9% tar removal efficiency.	[55]
Heavy metal-contaminated biomass (HMCB)	- Zeolite, - Tincal, and - Potassium Carbonate	- Pyrolysis and Gasification Processes - Highest Temperatures: 700 and 900 °C - Heating Rates: 30 and 5 °C/min - Use of Zeolite, Tincal, and Potassium Carbonate Catalysts	- Ionic liquid (IL) pretreatment of HMCB reduced syngas yield and altered composition in pyrolysis. - Tincal and zeolite additives improved syngas yield and H₂ concentration during catalytic pyrolysis of IL-treated HMCB but didn't exceed raw biomass pyrolysis. - Combining IL pretreatment with catalytic gasification led to significantly higher syngas yield and H₂/CO ratio compared to noncatalytic gasification. - Catalytic gasification of IL-treated HMCB produced five times more syngas (with higher H₂ content and lower CO₂) compared to tincal catalytic pyrolysis.	[61]

Table 2. Summary of Catalysts and Operating Conditions for Biomass Gasification to Syngas using Zeolite-Based Catalysts (continued)

Feedstocks	Catalysts	Operating Conditions	Findings	Ref
Wood	- Comparison between zeolite and fire clay chips as catalysts	- Catalytic gasification of wood - Variation in the content of the heat-carrying agent - Introduction of K_2CO_3 as an additive	- Zeolite's lower combustion temperature than fire clay may cause incomplete oxygen reaction and unburned carbon in gasification products. - Zeolite in wood pyrolysis generates fewer liquid products than fire clay. - Lowering the heat-carrying agent content reduces the difference in liquid product yields between zeolite and fire clay. - K_2CO_3 alters gaseous product composition during wood gasification.	[62]
Biomass gasification tar	- zeolite	- Thermal cracking - Catalytic cracking in the presence of zeolite, magnesium oxide, and aluminum oxide catalysts	- Biomass gasification tar underwent thermal and catalytic cracking to yield biodiesel, with zeolite, magnesium oxide, and aluminum oxide catalysts producing biodiesel in the ranges of 62-75 wt.%, 55-66 wt.%, and 67-71 wt.%, respectively. - The resulting bio-oil closely resembled conventional diesel but had slightly higher viscosity and acidic value.	[63]
Poplar wood	- ZSM-5	- Biomass gasification produces syngas rich in hydrogen and carbon monoxide, which is purified before being used to synthesize methanol and convert it into gasoline using catalysts like copper/zinc oxide/alumina and ZSM-5, respectively.	- The economic assessment analyzes biomass-derived syngas for gasoline production, covering gasification, syngas cleanup, methanol synthesis, and gasoline conversion with a ZSM-5 zeolite catalyst. - Estimated at 2007 U.S. dollars, the plant gate price for gasoline and LPG was \$15.73/GJ (\$16.60/MMBtu), with unit prices of \$0.52/L (\$1.95/gal) for gasoline and \$0.40/L (\$1.53/gal) for LPG. - Yields were 229.9 L of gasoline and 38.8 L of LPG per metric tons of dry biomass.	[64]
Biomass	- ZY - ZSM5 - ZY-5.2 - ZY-80 - Ni-supported ZY-30 - Ni-supported ZY-80 - Ni-supported ZY-5.2 - SiO_2/Al_2O_3 - Chabazite	- Investigation of catalytic activity for tar removal - Use of zeolites with varying pore sizes and acidity - Impregnation of nickel on zeolites - Long-term catalytic activity tests over a 97-hour period -	- Efficient biomass gasification relies on effective tar removal, with ZY showing superior catalytic activity over ZSM5. - ZY-5.2 exhibits higher acidity and better tar conversions than ZY-80. - Ni impregnation significantly boosts catalytic activity, particularly in ZY-30 and ZY-80, maintaining optimal tar conversions over time. - Reduced activity in some cases is attributed to coke deposition and reduced catalyst surface area.	[65]
Rice straw	- Natural zeolite	- Fixed-bed setup with temperature: 350°C - H_2S absorption using $Zn(NH_3)_6(OH)_2$ solution in impingers	- At 350 °C, the adsorption capacity for H_2S is 2.22.	[66]
Wood chips	- Natural zeolite - Calcined natural zeolite	- Wood chips (0.4-2.0 mm, 15 mg) are heated at 800 °C with N_2 gas flow. - Catalysts: natural zeolite and calcined natural zeolite, with surface areas of 25.1 and 6.91 g/m^2 , respectively.	- Equivalence ratio: 0.1595, air separation ratio: 0.325. - Achieved efficiencies: CGE 62.33%, CCE 80.90%, feedstock conversion 94.15%. - Syngas tar concentration: 3.64 g/Nm^3 , tar yield: 5.21 g/kg . - Syngas lower heating value: 7.34 MJ/Nm^3 .	[67]

Table 3. Properties of Natural Zeolites

Type of natural zeolites	Formula	Si/Al	BET m ² /g	Smicro m ² /g	V cm ³ /g	Vmicro cm ³ /g	Dav, nm	Acidity μmol/g	Coke deposition wt%	Ref
Clinoptilolite-fresh	(Na,K,Ca) ₂ –3Al ₃ (Al,Si) ₂ Si ₁₃ O ₃₆ •12H ₂ O	4.9	18	5	0.0972	0.0022	18.4	1953	-	[60]
Clinoptilolite-after the 10 th cycle		4.52	14	0.9	0.0003	0.1028	21.3	-	2.33	[60]
Mordenite	(Ca, Na ₂ , K ₂)Al ₂ Si ₁₀ O ₂₄ •7H ₂ O	4.88	25.28	16.75	0.05	0.0087	-	1190	-	[72]
Mordenite-synthetic-6.5		6.5	338.2	320.6	0.221	0.159	-	-	-	[73]
Mordenite synthetic-9.2		9.2	384.1	354.6	0.306	0.198	-	-	-	[73]
Mordenite synthetic-10.2		10.2	307.7	289.5	0.217	0.157	-	-	-	[73]
Chabazite-fresh	(Ca,K ₂ ,Na ₂ ,Mg)Al ₂ Si ₄ O ₁₂ •6H ₂ O	4.0	283	-	-	-	-	-	-	[65]
Chabazite-spent		-	2.25	-	-	-	-	-	0.4	[65]

affect their ability to accommodate and effectively catalyze all biomass components. Some larger molecules may not fit into the zeolite pores, limiting their effectiveness for certain reactions. Nevertheless, most medium and large pore zeolites in the range 5.2–5.9 Å can accommodate the majority of aromatic products and reactants within their pores [77].

In addition, natural zeolite catalysts face inherent limitations related to their structure and properties, including pore blockage, coke formation, dealumination, and thermal stability. Zeolites have a specific pore structure, and during the course of a reaction, products or byproducts can accumulate within the pores, leading to blockages. This pore blockage restricts reactant access to the

Table 4. Challenges of natural zeolite in biomass gasification

Challenges	Impact and Limitations	Ref
Variability in Composition	- Natural zeolites show variations in composition and catalytic properties, impacting performance consistency. - Composition variations impact reaction control and product quality.	[78]
Catalyst Deactivation	- Impurities from biomass (ash, Sulfur, alkali metals) lead to catalyst deactivation. - Ash deposits block active sites, reducing catalytic activity. - Sulfur compounds and alkali metals poison the catalyst. - Complex reactions and harsh conditions cause catalyst deactivation over time.	[79, 80]
Pore Size Limitations	- Biomass components may not fit zeolite pores. - Limitations in pore size affect effectiveness for certain reactions.	[81–83]
Pore Blockage	- Products or byproducts accumulate within pores, leading to blockage. - Reactant access to active sites is restricted, reducing catalytic activity.	
Coke Formation	- Carbonaceous species deposition occurs, blocking active sites and reducing surface area for catalysis. - Removal of coke is challenging, leading to irreversible deactivation.	[84, 85]
Tar Deposition	- Tar deposition hampers catalyst activity. - Reactants' accessibility to active sites is reduced, promoting catalyst deactivation.	[47, 86]
Dealumination	- Removal of aluminum atoms under harsh conditions leads to active site destruction and structure changes. - Catalytic performance is reduced due to dealumination.	[87]
Thermal Stability	- Natural zeolites have lower thermal stability compared to synthetic counterparts. - High-temperature reactions cause structural rearrangements and loss of catalytic activity.	[88, 89]
Advances in Natural Zeolite Development	- Enhance catalytic performance, stability, and selectivity. - Advancements contribute to the efficiency and viability of biomass gasification as a sustainable process.	[79, 90]

active sites and reduces catalytic activity. Furthermore, larger molecules may not fit into the zeolite pores, limiting their effectiveness for certain reactions. Coke refers to the deposition of carbonaceous species on the catalyst surface, which can occur owing to cracking or polymerization of the reactants. Coke deposition can block the active sites and reduce the overall surface area available for catalysis, leading to decreased activity and selectivity. Furthermore, coke removal is often challenging and result in irreversible deactivation. Additionally, during biomass gasification, tar compounds can condense and deposit on the catalyst surface, inhibiting catalytic activity. Tar deposition reduces the accessibility of reactants to the active sites and promotes catalyst deactivation.

Moreover, dealumination involves the removal of aluminum atoms from the zeolite framework [91]. This phenomenon can occur under harsh reaction conditions, such as high temperatures or strong acid/base environments. Dealumination leads to the destruction of active sites and changes in zeolite structure, resulting in reduced catalytic performance. Furthermore, natural zeolites, compared to their synthetic counterparts, tend to have lower thermal stability [92]. High-temperature reactions can cause structural rearrangements, collapse of the zeolite framework, and formation of amorphous phases, leading to a loss of catalytic activity. To address these challenges and limitations, researchers are actively developing advanced natural zeolites. These advancements aim to enhance the catalytic performance, stability, and selectivity, contributing to the

efficiency and viability of biomass gasification as a sustainable energy conversion process.

The Mechanisms of Zeolites Deactivation

Zeolites deactivation refers to the loss of catalytic activity over time due to various processes occurring during the course of a chemical reaction. This phenomenon can significantly impact the performance and efficiency of the catalysts, leading to reduced reaction rates, selectivity, and overall productivity. Deactivation mechanisms can be broadly categorized into two types: reversible and irreversible. Reversible deactivation involves processes that can be reversed under appropriate conditions, such as desorption or poisoning. Irreversible deactivation, on the other hand, involves permanent changes in the catalyst structure or active sites, rendering them less effective or completely inactive. Specific deactivation mechanisms depend on the nature of the catalyst, reaction conditions, and reactants involved.

Deactivation primarily occurs through two mechanisms as shown in Figure 2: physical and chemical interactions with detrimental components present in the feed or reaction environment [102]. Physical deactivation involves the obstruction or damage of active sites by the deposition of coke, tar, or carbonaceous species, which hinders reactant molecule access and compromises the structural integrity of zeolite catalysts. Chemical deactivation, on the other hand, involves chemical reactions between the catalyst and certain components in the feed, leading to the formation of species that block or chemically modify the active sites,

Table 5. Zeolite catalyst deactivation in industrial process

Industrial application	Reaction conditions	Zeolite used	Deactivation process	Impact of deactivation	Catalyst lifetime	Ref
Methyl Mercaptan Conversion	Temperature range: 573-823 K, N ₂ , CH ₄ , H ₂ O present	H-ZSM-5, H-SAPO-34, H-MOR, H-Y, H-Ferrierite, H-BEA	Coke formation	CH ₃ SH conversion drops from >99% to 87% over 6 hours on stream	Short to Moderate term, approximately 6-17 h	[93]
MTO (Methanol-To-Olefins)	Temperature: 450 °C Pressure: 4.0 MPa. H ₂ /MEOH/H ₂ O/N ₂ = 3/1/2.67/2.66	SAPO-34	Coke deposition	In H ₂ atmosphere, coke amount was 74.2%, while with H ₂ and H ₂ O, deposition rate decreased to 2.0 mg/(g·h), with 19.7% coke.	- In N ₂ : Short lifetime. - In H ₂ O: 15 h. - In H ₂ : 75 h - In H ₂ and H ₂ O coexisting: 118 h	[94]
Benzene from CH ₂	T = 993 K, P = 1 atm, SV = 1500 mL/g·h, CO ₂ = 2%	MCM-22	Coke deposition	Coke deposition reduced micropore and external surface areas by 60% and 10% respectively	100 h	[95]
Methanol Conversion	T = 723 K, P = 1 bar, WHSV = 9.5 g methanol/g zeolite/h	Z50-H1	Alkaline treatment with TPA+	Conventional zeolites accumulate nearly 100% internal coke within 15 h, while hierarchical zeolites show less than 20% coke deposition.	24 h (sharp deactivation)	[96]

Table 5. Zeolite catalyst deactivation in industrial process (continued)

Industrial application	Reaction conditions	Zeolite used	Deactivation process	Impact of deactivation	Catalyst lifetime	Ref
2-methyl-2-butene (2M2B) cracking	T = 773 K, P = 1 bar, WHSV = 12 g 2M2B h ⁻¹	OH-M	Coke deposition	The rates of deactivation exhibited a range spanning from -0.701 to -0.028% per minute.	15 h	[97]
		OH-M-H	ranging from		63 h	[97]
		OH-N	-0.701% to		39 h	[97]
		OH-N-H	-0.028% min ⁻¹ .		347 h	[97]
		F-M	OH-N: Coke deposited = 6.6 wt%.		15 h	[97]
		F-N	OH-N-H: Coke deposited = 22.6 wt%.		39 h	[97]
		F-N-H	F-N: Coke deposited = 5.8 wt%.		190 h	[97]
		OH-M	F-N-H: Coke deposited = 15.5 wt%		15 h	[97]
		OH-M-H	-		63 h	[97]
Methanol-to-olefins (MTO)	Inlet methanol partial pressure: 22 kPa at 623 K	HSSZ-13	Coke deposition	At 22 kPa methanol partial pressure and 623 K using HSSZ-13 catalyst (Si/Al = 8.4), methanol conversion stopped after about 102 s.	~102 s (for certain conditions)	[98]
Methanol-to-Olefins (MTO) Process	T = 723 K, WHSV = 6.6 g MeOH g cat. h ⁻¹ . Steam Cracking: T = 953 K, WHSV = 2.6 g Steam g cat. h ⁻¹	SAPO-34	Transformation of coke to naphthalene through steam cracking	-	~25 minutes (pilot plant)	[99]
Methanol-to-Aromatics	MTA at 450°C, 0.15 MPa, WHSV = 1.5 h ⁻¹	HZSM-5	Steam and Alkaline Treatment	The average coking rates range from Z having the highest rate (0.391%) to SAZ0.2 with the lowest (0.149%)	212 h for SAZ0.2, 125 h longer than AZ0.2, 63 h longer than SAZ0.3, 154 h longer than untreated HZSM-5	[13]
Methanol to Olefins	Room Conditions	Nano-SAPO-34	Exposure to Moisture	Optimizing catalysts like SAPO-34 and ZSM-5 by reducing crystal size, improving humidity stability, and boosting mesoporosity extends lifespan and enhances selectivity in methanol-to-olefins conversion.	14 d	[100]
Methanol to Olefins	Room Conditions	Standard-SAPO-34	Exposure to Moisture	Higher stability compared to nano-SAPO-34, slower changes after exposure to moisture		[100]
Aromatics production from syngas	-	Z15, Z25, Z140, ZN50 (microcrystalline and nanocrystalline ZSM-5 zeolites)	Acid site poisoning by heavy hydrocarbons (coke)	CO conversion increased from 79% to over 90% indicating mitigation of pore blockage and maintenance of active sites by zeolite acid sites.	15–17 h	[101]

thereby reducing their activity and selectivity. For instance, carbon monoxide (CO) or sulfur compounds in the feed can react with active sites, forming inactive species such as carbonates or sulfides, whereas reactions with oxygen-containing species can degrade the zeolite framework.

The deactivation mechanisms of zeolites exhibit variability across different industrial processes as outlined in Table 5. In the conversion of methyl mercaptan, the formation of coke on catalysts such as H-ZSM-5, H-SAPO-34, and other zeolites resulted in a decline in CH₃SH conversion

from >99% to 87% over a period of 6 hours, accompanied by a catalyst lifespan of short to moderate duration. Methanol-to-olefins (MTO) reactions employing SAPO-34 demonstrated diminished rates of coke deposition under conditions involving H_2 and H_2O , leading to an extension of catalyst lifetimes up to 118 hours. MCM-22 zeolites utilized in benzene production exhibited reductions in micropore and external surface areas by 60% and 10%, respectively, over a timeframe of 100 hours due to coke deposition. Alkaline-treated hierarchical zeolites exhibited coke deposition levels of less than 20% during methanol conversion, thereby prolonging the catalyst lifespan to 24 hours. Furthermore, variations in coke deposition during 2-methyl-2-butene cracking were observed across different zeolite types, resulting in lifetimes ranging from 15 to 347 hours. In the contexts of methanol-to-olefins and aromatics production, the detrimental effects of coke deposition and acid site poisoning were mitigated through the optimization of catalysts such as SAPO-34 and ZSM-5, leading to lifespans extended up to 14 days and 212 hours, respectively.

Factors Contributing to Zeolites Deactivation

Zeolites deactivation in industrial processes is influenced by various factors that affect the catalyst performance and longevity [103]. Understanding these factors is crucial for developing effective strategies to mitigate the deactivation effects. Key factors include contaminants, abrasion, structural changes in the zeolite framework, pore blockage, sintering and agglomeration of active sites, and poisoning of the active sites, as shown in Figure 2.

Contaminants such as nitrogen, Sulfur, and heavy metals chemically react with active sites of zeolite catalysts, deactivating or poisoning them [105]. Contaminants like nitrogen and nitrogen compounds present significant challenges to the catalytic activity of natural zeolites, contributing to deactivation or poisoning through multifaceted mechanisms. Nitrogen compounds pose a substantial threat by obstructing acid centers within zeolite catalysts and serving as precursors for coke formation. The significant molecular size of nitrogen often results in obstructing access to active sites within zeolite structures, thereby potentially poisoning acid sites and reducing catalytic activity [106–108]. Additionally, they promote coke formation, block pores in catalysts, reduce conversion rates, lower reaction rate constants, and alter selectivity [106–108].

Furthermore, different nitrogen compounds interact distinctively with the acid centers of zeolites. Nitrogen compounds like ammonia (NH_3) acts as a base and undergoes reactions with the acid centers, potentially altering their chemical properties [109]. Such alterations can substantially diminish the catalytic activity of the zeolite, hindering its ability to facilitate reactions, particularly evident in processes like biomass gasification [109]. Meanwhile, examining the interaction between the natural zeolite clinoptilolite and ammonium ions (NH_4^{4+}), a common nitrogen compound, provides insights into the mechanisms involved. Clinoptilolite, characterized by a framework structure composed of $[SiO_4]$ and $[AlO_4]$ tetrahedra, forms a three-dimensional network of interconnected channels and cavities [109]. The channels in clinoptilolite are sufficiently large to

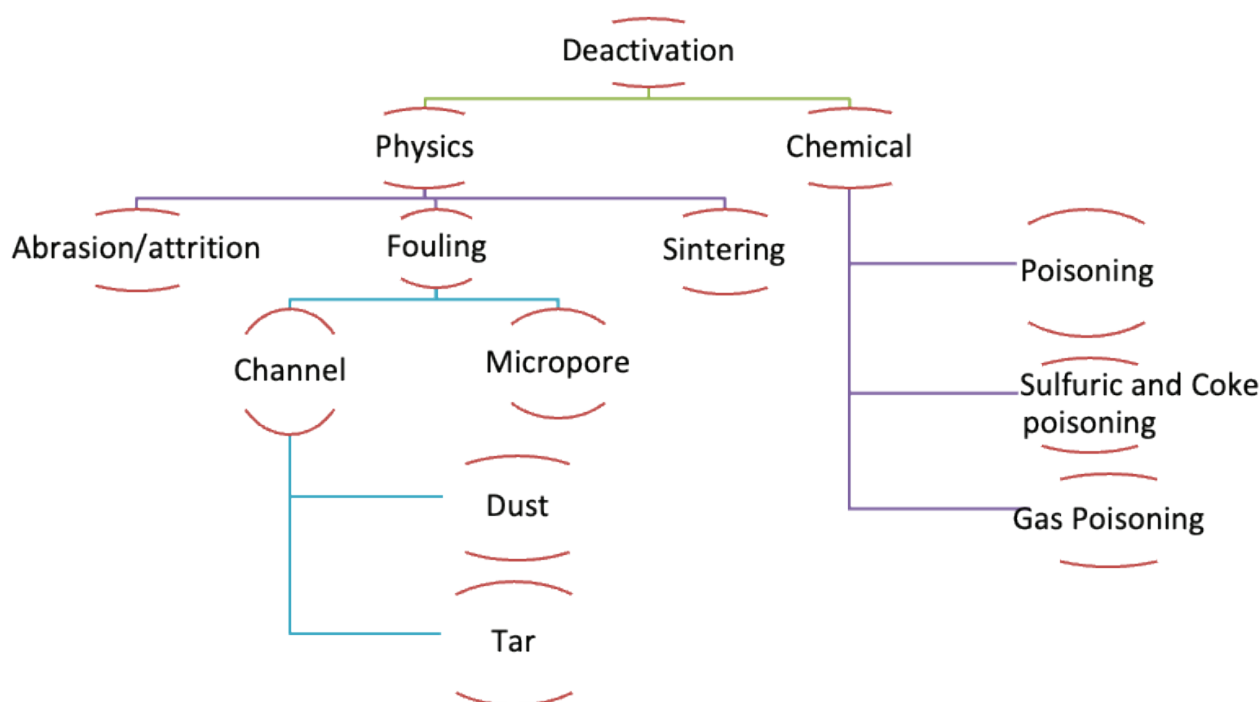


Figure 2. The various mechanisms of catalyst deactivation categorized by type.

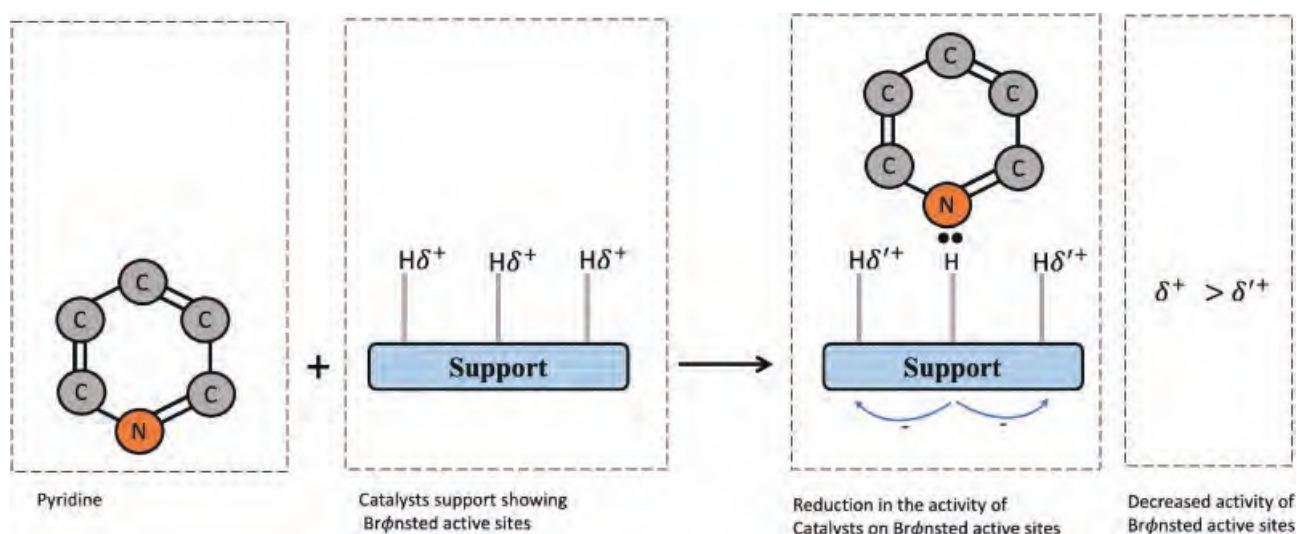


Figure 3. The mechanisms involved in the inhibitory effects of basic nitrogen compounds on the Brønsted acid centers. [From Adanenché et al. [104], with permission from Elsevier (Number: RLNK505682204/3002296642).

permit the passage of ammonium ions, which can interact with the zeolite through a process known as ion exchange [109]. In this exchange, ammonium ions in a solution can displace original cations (e.g., sodium, potassium, calcium, or magnesium ions) within the zeolite framework due to their attraction to the negatively charged framework [109]. Thus, nitrogen and nitrogen compounds adversely affect the catalytic activity of natural zeolites through mechanisms such as obstruction of acid centers, promotion of coke formation, and alteration of chemical properties. Understanding these mechanisms as shown in Figure 3 is critical for developing strategies to mitigate zeolite catalyst deactivation effectively [104].

A comprehensive understanding of these mechanisms is essential for optimizing catalytic performance in processes such as biomass conversion. Sulfidation stands out as a primary mechanism by which sulfur compounds impact zeolite catalyst deactivation as shown in Figure 4. Upon exposure to sulfur-containing compounds such as hydrogen sulfide (H_2S) or sulfur-containing organic compounds, zeolite active sites undergo interaction, forming metal-sulfur species. These species effectively compete for active sites, thereby reducing the availability of sites crucial for catalytic reactions [110, 111]. Consequently, the catalytic activity of zeolites diminishes, culminating in deactivation. Furthermore, sulfur compounds induce alterations in the structure and functionality of zeolites. The presence of sulfur can trigger chemical modifications or surface perturbations in zeolites, thereby influencing their catalytic prowess. This alteration further contributes to the decline in catalytic activity [110, 111]. Additionally, sulfur compounds exacerbate coke formation on the zeolite catalyst surface. Coke, a carbonaceous deposit, forms when undesirable side reactions are promoted or when active sites are blocked by

sulfur compounds. This phenomenon accelerates catalyst deactivation [110, 111].

In the context of sulfur dioxide (SO_2) exposure, sulfite (SO_3^{2-}) and sulfate (SO_4^{2-}) species emerge. Notably, sulfite can decompose or oxidize into sulfate at temperatures lower than those typical for conventional reactions in hydrocarbon processing (typically 400–600°C) [112, 113]. The generation of heat-stable sulfate species on the zeolite surface emerges as a primary factor in deactivation induced by SO_2 exposure [114, 115]. Moreover, sulfur compounds can influence catalytic cracking activity in metal catalysts, albeit to a lesser extent compared to nitrogen compounds. Sulfur's interaction with metal sites in the catalyst alters the catalytic cracking process, leading to decreased activity [116]. The susceptibility of aluminum-rich zeolites to sulfur uptake is noteworthy, with higher aluminum content correlating with increased sulfation at aluminum sites, consequently leading to decreased catalytic activity [116]. Thus, understanding the compositional characteristics of zeolite catalysts is imperative in evaluating their susceptibility to sulfur poisoning. Thus, sulfur's reactivity with the active sites of natural zeolites involves sulfidation, alteration, and coke formation mechanisms, collectively contributing to catalyst deactivation or poisoning. A profound comprehension of these sulfur-related factors is indispensable for optimizing catalytic performance in diverse industrial processes.

Nickel, another heavy metal contaminant, significantly contributes to the deactivation of zeolite catalysts during biomass gasification by promoting excessive coke formation as shown in Figure 5 [104]. Ni deposition on the catalyst surface leads to pore destruction and undesirable reactions. The conversion of nickel to nickel-oxides decreases the hydrogen generation ability, but caution is

needed to prevent reactivation of detrimental metallic nickel. However, unlike other heavy metals, Ni does not play a role in the destruction of zeolite structures [117]. Understanding the impact of nickel is crucial for optimizing catalyst performance in biomass gasification processes.

In metal catalysts such as nickel, the primary cause of accelerated catalyst poisoning is the formation and migration of sulfur elements [118].

Heavy metals interact with natural zeolites, causing deactivation of gasification processes primarily by

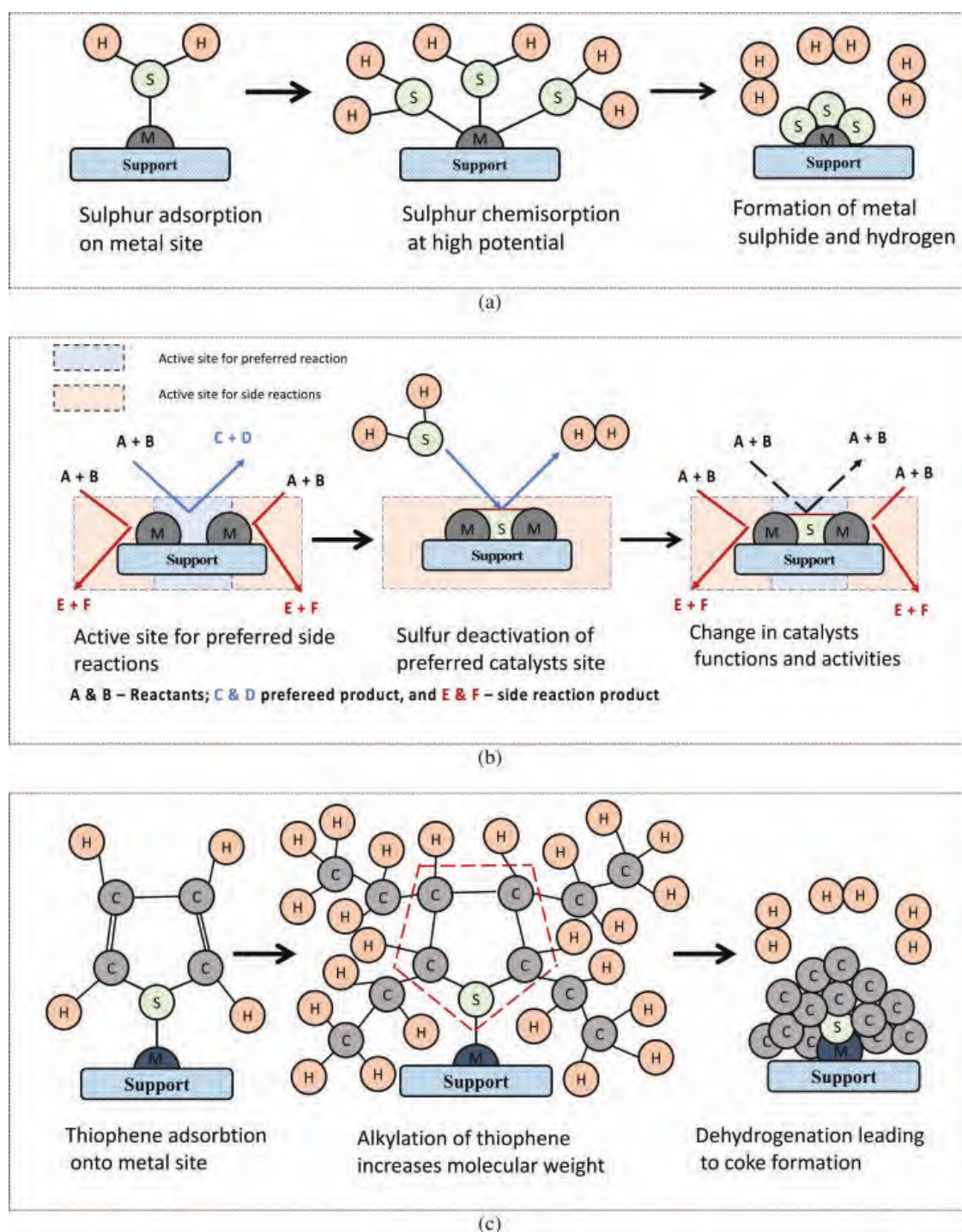


Figure 4. The three primary mechanisms responsible for the deactivation of catalysts due to Sulfur. These mechanisms include sulfidation, alteration, and coke formation. [From Adananche et al. [104], with permission from Elsevier (Number: RLNK505682204/3002296642).

poisoning the active sites crucial for catalytic reactions. This interaction involves the binding of alkali and alkaline earth metals to the acid centers of zeolites, essential for adsorbing and activating reactants like ammonia and NO_x in the SCR (Selective Catalytic Reduction) process [119]. Alkali metals such as potassium and sodium from coal fly ash or biomass combustion inhibit the adsorption of ammonia and reduce catalytic activity in SCR processes [119]. Additionally, heavy metals like Ni and Pb, immobilized within newly formed zeolite structures during processes like composting, reduce their mobility and environmental impact [120]. The mineralogical properties of coal fly ash in gasification processes can influence zeolite synthesis and properties, affecting their ability to adsorb heavy metals [121]. Experimental studies have shown that phosphates and sulfates can mitigate the poisoning effects of alkali metals on zeolite-based catalysts, preventing interactions with vanadium active sites [119]. Furthermore, alkali metals inhibit surface oxygen transfer crucial for activating adsorbed reactants, affecting zeolite deactivation [119]. Despite efforts to regenerate zeolites, only partial recovery of adsorption capacity is typically achievable, highlighting challenges in reusing zeolites once poisoned by heavy metals [119]. Understanding the intricate mechanisms by which heavy metals interact with natural zeolites is critical for developing strategies to mitigate catalyst deactivation in gasification processes.

Common structural changes in zeolite frameworks during deactivation in the biomass gasification process primarily involve the loss of crystallinity, decrease in acidity, and alterations in the distribution of active sites. One prevalent phenomenon contributing to structural changes is coke formation, where the accumulation of carbonaceous coke

on the catalyst surface obstructs the zeolite pores, impeding access to active sites. This coke is derived from heavy hydrocarbons and tars produced during gasification, hindering catalytic activity [122]. Another notable structural change is amorphization, wherein zeolites undergo structural collapse, transitioning from a crystalline to an amorphous state. This transformation is often induced by the harsh conditions encountered during biomass gasification, such as high temperatures and the presence of steam, which can degrade the zeolite framework [123]. Additionally, metal poisoning poses a significant concern, as alkali and alkaline earth metals present in biomass can interact with the zeolite framework, altering its acidity and leading to structural changes that compromise catalytic activity [122]. Furthermore, dealumination, characterized by the removal of aluminum atoms from the zeolite framework, can occur under the elevated temperature conditions of gasification, resulting in a reduction in acidic sites and impacting the zeolite's catalytic properties [122]. Lastly, high temperatures can induce phase transformations in the zeolite, altering its structural and chemical characteristics, further contributing to deactivation [122].

Another key factor that contributes to the deactivation of zeolite catalysts is abrasion. Abrasion significantly influences zeolite catalyst deactivation [17]. Both activated and deactivated catalysts experience mass loss, and the severity of attrition intensifies at higher temperatures and longer operation times. The activated catalysts exhibited improved attrition resistance and durability. Striking the balance between carbon removal and attrition resistance during regeneration ensures overall stability and performance. Moreover, the erosion or breakage of catalysts into

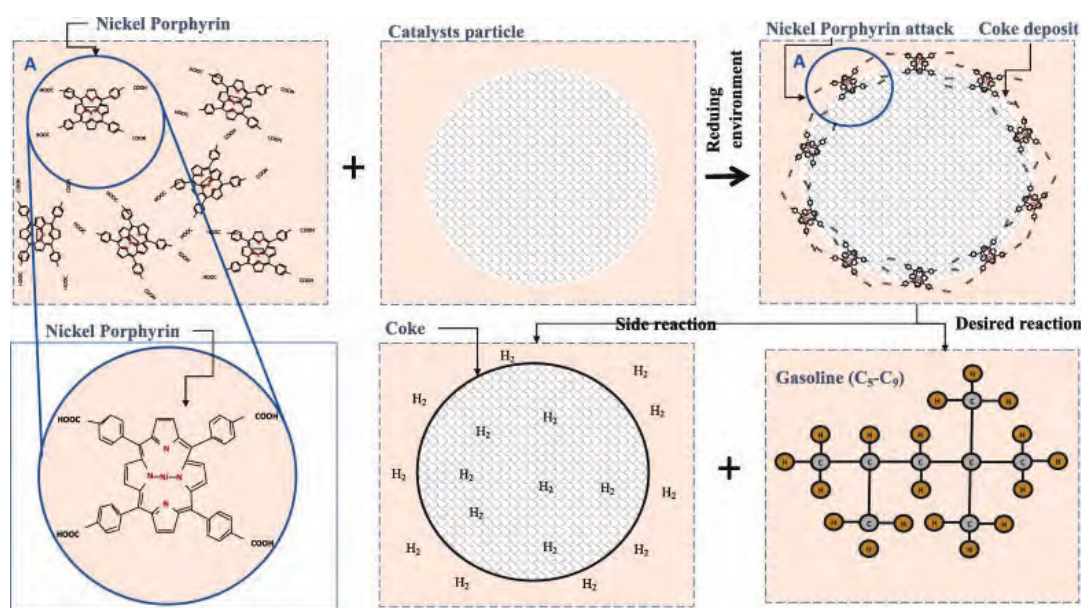


Figure 5. An illustration of the mechanisms involved in the poisoning of catalysts by nickel. [From Adananche et al. [104], with permission from Elsevier (Number: RLNK505682204/3002296642).

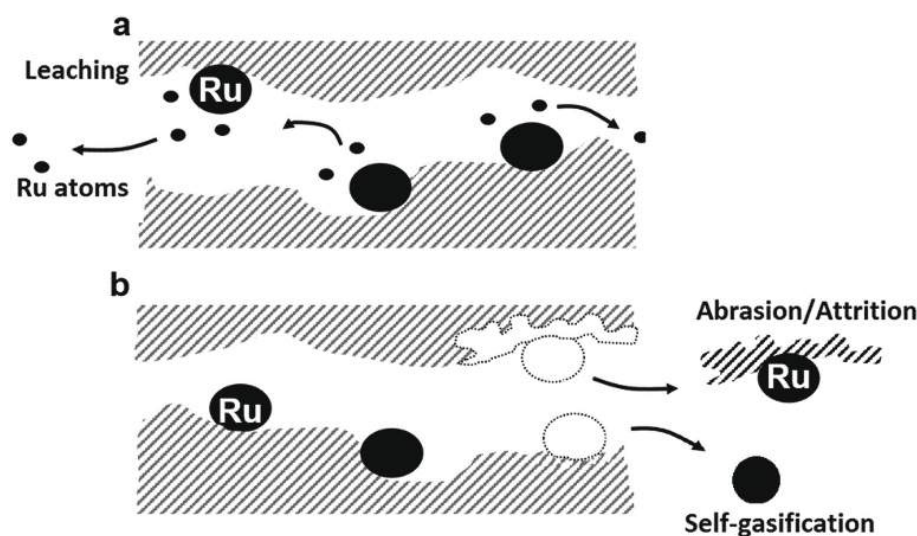


Figure 6. The schematic view of the mechanism for Ru loss from AC-supported catalyst. a) Ru leaching, b) loss of support fragments resulting in the loss of Ru NPs, which can be caused by the gasification process itself or from abrasion/friction between particles in the catalyst. [From Hunston et al. [124], with permission from The Royal Society of Chemistry (RSC) under CC-BY open access license.

smaller particles results from chemical reactions and prolonged exposure to high temperatures. However, smaller particles exiting the system can lead to a decline catalytic performance. Abrasion is crucial for designing efficient and durable catalysts. The schematic in Figure 6 depicts the mechanism of Ru abrasion and loss from activated carbon (AC)-AC-supported catalysts, highlighting two primary processes [124]. First, Ru leaching occurs, where Ru nanoparticles (NPs) dissolve and detach from the catalyst support. This leaching process can be influenced by various factors, including the gasification process. Second, the loss of support fragments leads to the detachment and loss of Ru NPs. This phenomenon can result from both the gasification process and abrasion or friction between particles within the catalyst.

To mitigate the impact of abrasion on zeolite catalyst activation, several strategies have been employed, each tailored to address specific challenges and operational requirements. One approach involves catalyst coating, where zeolite catalysts are encased in a protective layer during the synthesis of fine chemicals. This coating serves to shield the catalysts from abrasion, thereby preserving their activity over prolonged durations [9, 122]. Additionally, pelletizing, a technique commonly utilized in fluid catalytic cracking (FCC) processes in the petrochemical industry, has proven effective in enhancing the mechanical strength of zeolite catalysts, thus rendering them more resistant to abrasion [125, 126]. Furthermore, the addition of binders to zeolite catalysts in biomass conversion processes has been implemented to bolster their resistance to abrasion and extend their operational lifespan [9, 122]. Optimal reactor design also plays a crucial role in mitigating abrasion-induced

deactivation, with designs engineered to minimize high-impact points and turbulence, particularly in CO₂ utilization processes. Such designs aim to reduce wear on zeolite catalysts, thereby preserving their activity [9, 122]. Moreover, the implementation of regular regeneration cycles has proven effective in various industrial processes to restore the activity of zeolite catalysts following abrasion-related deactivation [9, 122]. Finally, the utilization of mesoporous zeolites represents a promising avenue for mitigating abrasion. By introducing mesopores into zeolite structures, diffusion limitations can be alleviated, and more active sites can be exposed, rendering the catalysts less susceptible to abrasion in catalytic applications [9, 122]. These diverse strategies collectively contribute to enhancing the durability and longevity of zeolite catalysts in the face of abrasion-related challenges.

Coke formation significantly affects zeolite catalyst deactivation by causing pore blockage and hindering the diffusion of reactants and products, leading to reduced catalytic activity [127, 128]. Understanding the characteristics and mechanisms of coke is vital for optimizing catalytic processes and mitigating catalyst deactivation, providing valuable insights from extensive studies on biomass gasification. The formation of coke is a complex process influenced by various factors, including operating conditions, catalyst composition, and feed properties [102, 129].

During biomass gasification, coke deposits can block pores and active sites, reducing catalytic activity, whereas contaminants such as alkali metals, sulfur, and chlorine further contribute to catalyst deactivation by chemically poisoning the catalyst and promoting coke formation. The increased presence of steam and/or oxygen supports the

gasification and/or combustion of coke precursors. The aromatic properties of the feed undergoing reforming can promote coke deposition via polymerization reactions. Therefore, addressing coke-related challenges and managing catalyst deactivation are crucial for enhancing the efficiency and effectiveness of biomass gasification.

Another influential factor in zeolite catalyst deactivation is the reduction in catalytic activity and selectivity due to sintering and agglomeration of active sites. The sintering and agglomeration of active sites play a crucial role in zeolite catalyst deactivation, leading to larger and fewer active sites. This process occurs at elevated temperatures and reduces the overall activity and selectivity of the catalyst. Factors such as high temperature, metal mobility, and the presence of reactive species can accelerate sintering, while the nature of the support material and metal influence the process [130]. Coke deposits on the catalyst surface promote sintering by acting as bridges between the metal particles, facilitating their migration and coalescence. In turn, sintering modifies the catalyst surface structure, influencing the adsorption and polymerization reactions that lead to coke formation during fouling or coking.

In the realm of natural zeolites, extensive research has been conducted to unravel the intricacies of deactivation, particularly in several types of natural zeolites such as clinoptilolite, chabazite, and mordenite. Deactivation in natural zeolites is driven by both surface and emboli mechanisms, exerting a profound impact on their sustained efficacy over time [131]. Detailed investigations into natural zeolites such as clinoptilolite, mordenite, and chabazite underscore distinctive deactivation processes influenced by factors such as metal poisoning, calcination procedures, and modifications.

The deactivation induced by thiophene in Ni/Clinoptilolite underscores the crucial role of the iron-nickel interaction in conferring sulfur resistance [132]. A more robust iron-nickel interaction is directly proportional to higher sulfur resistance, a critical factor for the catalyst's endurance. Furthermore, calcination before Ni impregnation in the ethylbenzene hydrogenation process enhances the interaction strength and sulfur resistance [133]. While clinoptilolite demonstrates promise in hydrogen sulfide removal through adsorption capacity of approximately 0.03 g S/g, its susceptibility to deactivation is evident in the oxidative removal of dissolved iron, especially in the presence of Mn^{2+} cations [134, 135].

Meanwhile, mordenite catalysts manifest diverse deactivation mechanisms contingent on specific catalytic processes. The significance of redox treatment and dealumination in influencing catalytic activity and stability is underscored [136]. Furthermore, chabazite zeolites exhibit diverse deactivation profiles in different catalytic scenarios. The incorporation of small platinum crystallites in Pt/H-chabazites and Pt/H-Y zeolite enhances deactivation profiles during n-butane conversion [137]. Ga+-chabazite zeolites emerge as highly selective catalysts for nonoxidative

propane dehydrogenation, boasting a remarkable 96% propylene selectivity [138]. These findings underscore the intricate interplay between zeolite structure, metal incorporation, and reactant species in determining deactivation behaviors in various catalytic processes.

In the realm of catalyst deactivation, both synthetic and natural zeolites encounter shared and distinctive factors influencing their performance over time. Common culprits include contaminants like nitrogen, sulfur, and heavy metals, which chemically react with active sites, inducing deactivation or poisoning in both zeolite types [105]. Additionally, coke formation stands out as a major contributor, impeding reactant and product diffusion, leading to pore blockage and reduced catalytic activity [127, 128]. Abrasion, characterized by mass loss and attrition, also plays a role, with its severity escalating at higher temperatures and extended operation times [17, 139]. Moreover, the sintering and agglomeration of active sites contribute to the reduction in catalytic activity, affecting both synthetic and natural zeolites [130, 140].

Natural zeolites, exemplified by clinoptilolite, mordenite, and chabazite, introduce unique dimensions to the deactivation narrative. These zeolites exhibit distinctive metal interactions, as seen in clinoptilolite's sensitivity to the iron-nickel interaction influencing sulfur resistance [132, 133]. Each natural zeolite showcases specific deactivation mechanisms; for instance, clinoptilolite displays notable potential for reactivation through nitrous oxide treatment and susceptibility to oxidative removal of dissolved iron [135, 141]. Mordenite experiences irreversible deactivation due to dealumination and pore blockage [142, 143]. Chabazite zeolites present trial-dependent deactivation patterns and demonstrate highly selective catalytic activity [138, 144].

Quantitative insights, encompassing sulfur resistance, adsorption capacity, and catalytic activity, serve as crucial metrics for understanding and comparing zeolite performance in both synthetic and natural contexts [104, 117]. Hence, a thorough understanding of these common and unique deactivation mechanisms, along with quantitative data, is essential for devising efficient strategies to alleviate deactivation consequences and improve the overall catalytic efficiency across various industrial applications.

Understanding the Mechanisms of Zeolites Deactivation in Biomass Gasification

The mechanisms of zeolite catalyst deactivation are crucial for developing strategies to mitigate their effects and improve catalyst performance. Several key factors contribute to catalyst deactivation, including the adsorption and diffusion of reactants and intermediates, reaction-induced structural changes, role of surface acidity and basicity, impact of contaminants, and use of modelling and simulation approaches. Surface acidity and basicity are integral factors influencing zeolite deactivation, particularly in the realm of catalyst performance within biomass gasification

applications. Zeolite structures, characterized by interconnected silica and alumina tetrahedra, engender Brønsted acid sites pivotal for catalytic reactions essential in biomass gasification, such as cracking and hydrocracking of petro-fractions [122]. However, the presence of these robust acid sites can hasten zeolite deactivation by fostering the accumulation of carbonaceous coke on the catalyst surface. Furthermore, acid sites are vulnerable to poisoning by basic nitrogen compounds and ions like Ca^{2+} or K^+ [125].

Conversely, surface basicity offers promise in bolstering catalytic stability by impeding carbon deposition on the catalyst surface during biomass gasification [145]. Basic sites facilitate the activation of acidic molecules, such as CO_2 , thereby potentially enhancing catalyst stability. Nevertheless, the generation of basic sites often mandates severe pre-treatment temperatures, necessitating careful control to avert adverse effects on catalyst performance [145]. Moreover, modifications to the acid-base characteristics of zeolite surfaces, achieved through techniques such as dealumination or metal incorporation, can alter catalyst activity, selectivity, and stability across various processes, including CO_2 -mediated dehydrogenation of propane [146, 147]. Additionally, the distribution of acidic and basic sites within zeolite structures influences catalyst resistance to deactivation by modulating the rate of carbon species production and oxidation [148].

Furthermore, the adoption of strategies such as core-shell structures in zeolites holds potential for enhancing catalyst performance by furnishing shape selectivity and resistance to sintering, particularly advantageous in biomass conversion and CO_2 conversion processes [149]. Additionally, the fine-tuning of physicochemical properties of catalyst supports, encompassing acidity and basicity, offers a feasible avenue to mitigate carbon deposition and enhance CO_2 activation, thereby augmenting catalyst stability and curtailing deactivation [148]. The adsorption and diffusion of reactants and intermediates on zeolite catalyst surfaces are prevalent mechanisms contributing to deactivation. These phenomena entail the blocking of active sites and consequent reduction in catalytic activity. Investigations have indicated that the adsorption of phenol on zeolite catalysts employed in biomass gasification can lead to a substantial decline in catalytic activity, reaching up to 50%, attributable to the obstruction of active sites [65, 150, 151]. Diffusion limitations of bulky reactant molecules within the zeolite framework can also affect the catalytic performance [152].

Furthermore, the high temperatures and reactive conditions prevalent in biomass gasification can induce structural modifications in zeolite catalysts, leading to deactivation. Thermal aging during biomass gasification can result in the collapse of the zeolite framework, reducing surface area and pore volume. Studies have indicated that zeolite catalysts' surface area may decrease by approximately 25–30% after thermal aging at 800°C for 24 h [153]. Additionally, contaminants like alkali metals and sulfur compounds in

biomass feedstocks can interact with zeolite catalysts, contributing to their deactivation. For example, alkali metals such as potassium and sodium can react with zeolite catalysts, diminishing their catalytic activity. Research findings suggest that the presence of 1 wt% potassium on zeolite catalyst surfaces can lead to a reduction in biomass gasification conversion rates by up to 40% [154]. Similarly, sulfur compounds can poison zeolite catalysts, leading to decreased activity and selectivity [155].

In the realm of gasification processes, the deactivation mechanisms of mordenite, clinoptilolite, and chabazite play pivotal roles in shaping the efficacy of these natural zeolite catalysts. Mordenite, utilized in isobutane alkylation, confronts potential deactivation through coke formation, particularly notable in lanthanum-exchanged Y-zeolite, where substantial carbonaceous deposits require elevated temperatures for elimination [156]. The presence of excessively strong acidity in certain catalysts accentuates deactivation, underscoring the intricate interplay between acidity, coke deposition, and catalytic activity [156]. Clinoptilolite, identified for its potential in tar removal during biomass gasification, undergoes deactivation during hydrogen sulfide adsorption. The adsorption capacity for H_2S , a toxic and corrosive impurity, underscores the material's efficacy in removing harmful components [134]. Modeling the deactivation process, with a focus on concentration-dependent activity, enhances the understanding of adsorption kinetics and contributes to the optimization of clinoptilolite's performance in gasification applications [134]. Meanwhile, chabazite zeolites, exhibiting versatility in diverse catalytic scenarios, showcase distinctive deactivation profiles [65]. Their susceptibility to deactivation in specific processes, such as nonoxidative propane dehydrogenation, emphasizes the need for tailored strategies to mitigate deactivation effects and optimize overall catalytic efficiency in gasification contexts.

Therefore, to gain a deeper understanding of the true mechanism behind zeolite catalyst deactivation, modelling and simulation approaches can be employed. Computational studies have provided insights into atomic- and molecular-scale interactions between reactant molecules, intermediates, and zeolite catalysts during biomass gasification. These methods aid in predicting the catalytic performance, comprehending the impact of structural modifications, and developing strategies to mitigate catalyst deactivation. It is important to note that quantitative data on catalyst deactivation in biomass gasification can vary depending on the specific zeolite catalyst, biomass feedstock, and operating conditions utilized in different research studies.

Strategies for Mitigating And Regenerating Zeolites Deactivation in Biomass Gasification

Mitigating zeolite deactivation is crucial for maintaining the stability and performance of catalysts in various industrial processes, especially in biomass conversion and hydrocarbon reforming. Designing catalysts resistant

to contaminants is a fundamental strategy, involving the incorporation of metal promoters or modifiers like nickel or cobalt into the zeolite structure. Studies have demonstrated a substantial reduction in coke formation and enhanced catalytic performance, with modified zeolite catalysts showing up to a 60% decrease in coke yield compared to their unmodified counterparts [29, 157]. An alternative and efficient method involves the continuous removal of coke deposits in situ through catalytic steam gasification, resulting in a reduction in the coke deposition rate by over 10 times. [158]. The use of a fluidized bed reactor aids in coke reduction by shuttling catalyst particles between the reactor and a burner, where coke deposits are burned off [122, 159]. Additionally, employing a two-stage reactor system, which separates gasification and combustion processes, proves beneficial in minimizing coke deposition on the catalyst surface [122, 160, 161].

A promising avenue for zeolite deactivation mitigation is the development of core-shell zeolite structures [162]. This novel class of nanomaterials involves a core material, typically a metal or metal oxide, surrounded by a zeolite shell. The shell acts as a protective layer, preventing coke deposition and metal sintering, thereby extending the catalyst's activity and selectivity [163]. Despite this potential, there is a notable gap in quantitative comparisons of the lifespan between conventional zeolites and core-shell zeolites under various biomass gasification conditions.

Furthermore, incorporating redox metal oxides like CeO_2 , FeO_x , or perovskite-based support enhances the basic strength of catalysts by creating oxygen vacancies [163]. These vacancies act as basic sites, and the redox metal

oxides serve as electron reservoirs, preventing deactivation by maintaining the basicity of the catalyst. Additionally, the addition of promoters or additives, such as rare earth metals or transition metals, has been shown to improve zeolite catalyst stability against alkali metal poisoning. Studies reveal that even a small addition, such as 2 wt% of a rare-earth metal promoter, can enhance zeolite catalyst activity by up to 50% in the presence of alkali metals [164]. Similarly, the inclusion of silica or alumina has proven effective in reducing coke formation and enhancing overall catalyst stability [165]. Thus, a comprehensive strategy involving catalyst design, reactor engineering, and the incorporation of innovative materials is essential for mitigating zeolite deactivation and optimizing the efficiency of catalytic processes. Ongoing research efforts are needed to quantify and refine the performance of these strategies under diverse operating conditions in biomass gasification.

Meanwhile, various methods have been employed to implement zeolite regeneration techniques aimed at restoring the catalytic activity of deactivated zeolite catalysts. Regenerating zeolite catalysts in biomass gasification presents several challenges because of the complex nature of biomass feedstock and the deactivating species involved. Effective regeneration strategies are crucial for maintaining the catalyst performance and extending its lifespan. To address these challenges, the following regeneration or rejuvenation strategies can be employed: thermal or oxidative regeneration [166–169], chemical regeneration [65, 170, 171], steaming or humid air regeneration [170, 172, 173], and combination of regeneration techniques as shown in Table 6.

Table 6. Catalyst regeneration and rejuvenation techniques

Catalyst	$\text{SiO}_2/\text{Al}_2\text{O}_3$ ratio	Surface area $[\text{m}^2/\text{g}]$	Application	Regeneration Methods	Results	Ref
Thermal Method						
HZSM-5	30-28	37-100	1-butene oligomerization in a fixed-bed reactor	- Temperature-programmed sweeping (TPS) with nitrogen (N_2)	- TPS- N_2 eliminates soft coke at 400 °C, whereas hard coke is eliminated through combustion	[174]
Cu-Nb/HZSM-5	30	-	Chlorinated volatile organic compound elimination	- Restoration at 400 °C in air	- Regenerating at 400 °C in air restores most Brønsted acid sites by converting coke into saturated hydrocarbons or CO_2 .	[172]
MOR zeolite	-	-	H_2S on Blast furnace.	- In-situ regenerable Cu/Zeolite, $T = 280$ °C.	- MOR zeolite enhances both Cu dispersion and desulfurization capacity.	[175]
Y and ZSM-5 zeolites	200, 500-		Adsorption	- $T = 30$ -575 °C	- Regenerated zeolite maintains perfect unit cell parameters and adsorption ability.	[176]
Ni/ Al_2O_3	NR	NR	Pyrolysis to reforming of waste tyres.	- $T = 600$ °C from ambient temperature.	- Above 600°C, filamentous carbon loses weight; below 600°C, amorphous carbon oxidizes more readily.	[177]

Table 6. Catalyst regeneration and rejuvenation techniques (continued)

Catalyst	SiO ₂ /Al ₂ O ₃ ratio	Surface area [m ² /g]	Application	Regeneration Methods	Results	Ref
Dealuminated Y-zeolite	20	704	Adsorption of acetone and toluene	- T = 296-473 K	- Roll-up effect was stronger with acetone than toluene. Acetone's Type-III isotherm at higher temperatures allowed faster recovery in higher concentrations.	[178]
Chemical Method						
ZSM-5 zeolites	-	284, 332	Methanol to propylene (MTP)	- Insitu ZSM-5 zeolite rejuvenated with toluene	- Mesoporous ZSM-5 exhibited over 50% longer lifespan due to enhanced rejuvenation.	[179]
NaY zeolites	5	-	Sulfur dioxide (SO ₂) removal	- Alcohol	- Alcohol promotes NaY crystal formation, increasing surface area and microporosity.	[180]
Zeolite Clinoptilolite commercial products	4.27	-	Ammonium removal	- NaCl	- Optimal zeolite regeneration and NH ₄ ⁺ conversion occur at 20 g/L NaCl concentration and pH 10.0.	[181]
Zeolite -Y	30, 80	750, 780	Biomass gasification	- Ni impregnated to Zeolite.	- Naphthalene conversions remained above 99% throughout the 97-h testing period, with minimal loss of activity.	[3]
HZSM-5	42, 59	181.4	Plastic gasification	- Co/SiO ₂ impregnated to Zeolite.	- This catalyst is recommended for stable hydrogen production, lasting over 15 hours.	[182]
MOR zeolite	20	546	CO methanation	- Ru doped MOR zeolite	- No deactivation occurs over 100 hours, demonstrating perfect thermal stability.	[179]
ZSM-5	50	291, 285	Cracking of JP-8 a hydrocarbon fuel	- Pt and Gd promoted ZSM-5	- Gd metal coexistence partially inhibits Pt particle agglomeration on the regenerated ZSM-5 catalyst.	[183]
FAU zeolites	300	-	CO ₂ adsorption	- Amina	- Impregnating FAU zeolites with PEI and DETA enhances regeneration stability over 10 cycles.	[184]
Zeolite -X	-	-	Pesticide adsorption	- Ethanol	- Ethanol emerges as the most efficient choice for rejuvenating zeolite-carbon composites with adsorbed MCPA and 2,4-D.	[185]
Steaming Method						
Dealuminated zeolite	-	-	Hydrophobic zeolites	- Steam 130 °C, 1 bar	- Desorption with small steam flow rates can be equally effective as with large flow rates.	[186]
HZMS-5.			Biomass pyrolysis oils	- Steam	- The presence of vapor leads to substantial gas product generation, surpassing a 70% yield at a catalyst temperature of 550 °C.	[173].
Zeolite HZSM-5	100	-	Methanol conversion	- Vapor at 270 °C	- Lower temperatures facilitated the complete removal of the substance, causing a delay and restoring the catalyst's activity entirely.	[187]
Combination Method						
ZSM-5 zeolite	280	-	Adsorption	- High temperature zeolite loaded with MTBE-DCE and DCE-TOL (T = 30-400 °C)	- Configuration can impact molecule mobility within the framework, with desorption temperature closely linked to molecule positions in the porous structure.	[188]
MFI zeolite	13.40	-	Methane dehydroaromatization (DHA)	- Mo-MFI zeolite at 973 K in oxidative environments	- Combining spent Mo-MFI and H-MFI adds extra H ⁺ sites, aiding in dispersing clustered Mo domains into ion-exchanged Mo species, thereby restoring DHA reaction rates.	[189]

Thermal regeneration involves subjecting a deactivated zeolite catalyst to high temperatures to remove coke deposits and reactivate the catalyst. Controlled oxidation or calcination in an oxygen-rich atmosphere can effectively burn off carbonaceous deposits and restore the catalyst activity. The regeneration temperature and duration must be optimized to achieve efficient coke removal without causing structural damage to the catalysts. Studies have shown that thermal regeneration at temperatures between 500–800 °C can lead to significant coke removal and catalyst reactivation [167, 168]. At high regeneration temperatures (~565 °C) in O₂ or N₂ atmospheres. However, repeated regeneration treatments can quickly decrease the desulfurization performance of the adsorbent [169]. In metal-impregnated zeolites, regeneration at high temperatures shows the presence of agglomeration so that it can gradually reduce the performance of the catalyst [189]. Therefore, the regeneration method at low temperatures (296 to 473 K) also becomes an alternative for regenerating zeolite, whether as a catalyst or adsorbent [169, 178].

Experimental findings provide compelling evidence of the efficacy of strategies such as metal modification and continuous coke removal in mitigating zeolite deactivation. Blending methanol with the catalyst material has been shown to significantly reduce coke deposition during bio-oil conversion. Previous study reported a noteworthy 1.55 wt.% reduction in coke deposition, highlighting the potential of this approach to mitigate zeolite deactivation [190]. Moreover, metal modification has emerged as another promising strategy for enhancing zeolite stability where zeolite Y modified with 0.5 wt% Pt exhibited a 50% increase in catalyst lifetime compared to the unmodified version [191]. Moreover, the presence of Pt reduced the rate of coke formation from 1.2 g_{coke}/g_{zeolite}/hour to 0.8 g_{coke}/g_{zeolite}/hour, underscoring the effectiveness of metal modification in mitigating zeolite deactivation [191]. Furthermore, continuous coke removal processes provide an energy-efficient alternative to thermal methods for preserving zeolite activity, with ozone-mediated continuous coke removal achieving a rate of 0.05 g_{coke}/g_{zeolite}/hour at 180 °C, significantly lower than the temperatures exceeding 500 °C required for thermal methods, thereby highlighting the potential of continuous coke removal as a less damaging approach to zeolite activity preservation [192]. Further substantiating the benefits of metal modification investigated zeolite modification with 1 wt% Pd, which notably increased resistance to deactivation by coking [193]. The Pd-modified zeolite maintained 90% of its initial activity after 20 hours of reaction time, while the unmodified zeolite's activity dropped to 40% under similar conditions, highlighting the effectiveness of metal modification in prolonging zeolite activity [193]. Moreover, continuous coke removal methods have demonstrated remarkable efficacy in reducing coke deposits on zeolite catalysts revealing a 70% reduction in coke deposits after 2 hours of continuous coke removal using steam at 500 °C [193]. This

method proved superior to traditional coke burning, which achieved only a 50% reduction in coke deposits under similar conditions [193]. Metal modification plays a crucial role in enhancing coking resistance and introducing novel active sites, consequently boosting zeolite catalytic activity [125]. Additionally, the effective removal of coke from HY zeolites using ozone at 180 °C underscores the efficacy of coke removal techniques within the temperature range of 100 °C to 200 °C, demonstrating the versatility of such processes under different conditions [170]. These quantitative data underscore the effectiveness of metal modification and continuous coke removal strategies in mitigating zeolite deactivation, offering valuable insights for enhancing zeolite stability and catalytic performance.

Understanding the intricate interplay among various factors influencing zeolite deactivation is paramount for enhancing biomass gasification processes. Among these factors, coke formation emerges as a significant contributor to zeolite deactivation, owing to its highly selective nature dictated by the porous structure of the zeolite [194]. Notably, the rate of coke formation exhibits substantial variation across zeolites with different pore sizes, with small and medium pore zeolites demonstrating rates ranging from 50 to 1000 times lower compared to wide-pore zeolites [194]. Unraveling the mechanisms governing coke formation is pivotal, with literature indicating the operation of two parallel pathways, namely the direct mechanism and dual-cycle mechanism, in accordance with the hydrocarbon pool theory. Moreover, the deposition and accumulation of coke on external zeolite surfaces significantly contribute to reaction deactivation [195]. Furthermore, contaminants play a pivotal role in exacerbating coke formation or sintering processes, thereby highlighting the interconnected nature of these deactivation mechanisms. For instance, the presence of nitrogen in metal-zeolites can exert pronounced effects on the energy and pathways of the nitrogen reduction reaction (NRR) [196]. Similarly, sulfur and heavy metals have been observed to form stable metal-sulfur adducts, thereby facilitating nucleation around metal precursors and ultimately promoting the formation of an encapsulation structure, which contributes to deactivation [197]. Hence, a comprehensive understanding of these interactions is imperative for devising strategies to mitigate zeolite deactivation and enhance the efficiency of biomass gasification processes.

Chemical regeneration methods involve the use of chemical agents to dissolve or remove deactivating species from a catalyst surface. For example, acidic and alkaline solutions can be employed to remove ash deposits and metal contaminants, respectively. Acid washing has been used to remove ash deposits from zeolite catalysts during biomass gasification, leading to improved activity and selectivity [65, 171]. Chemical regeneration methods must be carefully designed to avoid damaging the zeolite structure or altering its catalytic properties.

Steaming regeneration involves treating a deactivated zeolite catalyst with high-temperature steam to remove coke and tar deposits. Steam can react with carbonaceous species, facilitating volatilization and removal from the catalyst surface. Studies have demonstrated that steam regeneration can effectively restore the catalytic activity of zeolite catalysts during biomass gasification [173]. The regeneration temperature, steam flow rate, and duration should be optimized to achieve efficient coke and tar removal without adversely affecting zeolite structures. Furthermore, the combination of different regeneration techniques can enhance the effectiveness of catalyst regeneration. For instance, a sequential approach involving thermal regeneration, followed by chemical washing or steaming, can provide comprehensive catalyst regeneration. The thermal step removes coke deposits, while subsequent chemical treatment or steaming further removes contaminants and restores the activity of catalyst. This combined approach has been shown to improve the catalyst performance and extend its lifespan.

In some cases, catalyst rejuvenation involves the modification or enhancement of the deactivated zeolite catalyst to restore its activity. This can be achieved by adding promoters, modifiers, or coatings to enhance catalytic activity or resistance to deactivation. For example, the addition of metal promoters or modifying agents such as phosphorus or boron has been found to enhance the activity and stability of zeolite catalysts in biomass gasification [171, 198]. Type selection of a metal catalyst impregnated with zeolite, such as nickel (Ni), cobalt (Co), and ruthenium (Ru), presents an alternative route to achieve thermal stability and prolong the catalyst's lifespan, making it a favorable choice [3, 179, 182]. The selection of zeolite type determines the ability of catalyst reactivity and stability, as the MFI of zeolite is better than ZSM-5 [199]. Catalyst rejuvenation can be an effective strategy to overcome deactivation and extend the lifespan [200, 201]. By employing these regeneration strategies, the performance and longevity of natural zeolite catalysts for biomass gasification can be improved. Optimizing the regeneration conditions, such as temperature, duration, and regeneration agents, is essential to achieve efficient removal of deactivating species while preserving the structural integrity and catalytic properties of the zeolite catalyst.

A wide array of innovative strategies is employed in the application of deactivation mitigation and regeneration methods for zeolites in biomass gasification as shown in Table 6, all with the overarching goal of improving operational efficiency and sustainability. Recent investigations have singled out hierarchical micro/mesopore-structured zeolites as particularly promising solutions for combating catalyst deactivation resulting from coke formation. Studies have demonstrated notable enhancements in catalytic activity associated with the utilization of these zeolite structures [16]. Core-shell zeolite structures are used in biomass gasification due to their stability during reducing and oxidizing treatment [202]. The core-shell interface

remains stable even though strong changes occur in the core due to the reduction of copper oxide to metallic copper particles [202]. Meanwhile, redox-based regeneration techniques have effectively reinstated as much as 98% of the original functionality of cobalt catalysts incorporating zeolites, employed in the Fischer-Tropsch synthesis [203], suggesting prospective suitability for biomass gasification endeavors.

Integration of Redox Metal Oxides in Zeolites Redox metal oxides are widely employed for biomass conversion due to their unique acid-base properties [204]. They are used for the dehydration of sugars to important furanic compounds [204]. Some additives like Ce, La, Mg, K, Mo, Zr, and Mn could improve the performance and stability of Ni-based catalyst, particularly Mo, Zr, and Mn led to the highest activity and stability during biomass gasification [205]. Integrating metal nanoparticles within these zeolites further enhances catalytic activity, providing additional active phases and improving overall performance [16]. Regeneration of zeolite catalysts, including those with Sn, can induce structural alterations that enhance their accessibility, hydrophobicity, reactivity, and stability, thereby improving their effectiveness in biomass conversion processes [206]. Moreover, oxidative treatments for catalyst regeneration, including molecular oxidation with oxygen and ozone, have been extensively reviewed, with studies highlighting their efficacy in recovering catalytic activity and extending catalyst lifespan [125, 170]. Innovative concepts such as integrating structured iron-based catalytic monoliths in fluidized bed gasifiers have shown preliminary effectiveness in gas conditioning and tar conversion [207]. Additionally, catalyst addition to biomass gasification processes, such as $\text{CeO}_2(111)$, has been explored to minimize coke formation and enhance gasification efficiency [208]. Mild regeneration methods involving air oxidation and H_2 reduction have been successfully applied, demonstrating their applicability to various catalysts and additional reactions involving oxygenated molecules [209]. Another novel approach involves regeneration by ZnCl_2 activation, effectively eliminating carbon deposition and recovering catalyst activity, thus prolonging operational life [210].

Furthermore, the application of metallic additives such as nickel and cobalt for the purpose of alleviating coke deposition on zeolite catalysts holds significant importance in biomass gasification processes. The incorporation of nickel into ZSM-5 zeolite has been proven to significantly diminish coke formation, thereby augmenting the stability of the catalyst [211, 212]. Nickel and cobalt exhibit strong hydrogenation abilities, converting coke precursors into gaseous products, thereby improving syngas quality by reducing CO_2 content and increasing CH_4 content. Moreover, core-shell zeolite structures, where the active zeolite core is encapsulated by an inert shell material, offer a promising approach. These structures, along with hierarchical porosity, facilitate better diffusion and mass transfer, leading to increased catalyst stability and efficiency in biomass

conversion [125]. A core-shell ZSM-5@silica catalyst has been developed, effectively mitigating zeolite deactivation by coke and extending its lifespan [213]. Additionally, the integration of redox metal oxides into zeolite catalysts has shown potential in reducing coke formation and maintaining the quality of gasification processes. Moreover, the incorporation of cerium oxide into HZSM-5 zeolite led to a marked reduction in coke deposition during the conversion of biomass-derived compounds. This effect can be ascribed to the robust redox properties exhibited by cerium oxide [9, 214]. Furthermore, regeneration techniques such as thermal regeneration, chemical washing, and steaming are essential for restoring the activity of deactivated zeolite catalysts. Heating spent HZSM-5 catalysts to 550°C for 5 hours effectively removed coke deposits, restoring catalytic activity [125]. Chemical washing involves dissolving coke deposits with a suitable solvent, partially restoring catalyst activity [56, 125]. Steaming entails treating spent zeolite catalysts with steam to remove coke deposits and restore catalytic activity [125, 215]. These novel approaches significantly impact biomass gasification processes by enhancing catalyst stability, maintaining activity over longer operational periods, increasing syngas selectivity, prolonging catalyst lifespan, improving product yields, process efficiency, and the overall quality of biomass gasification processes.

Potential Applications of Deactivated Zeolites in Other Fields

Despite their reduced catalytic activity, deactivated natural zeolites still possess unique properties that make them suitable for various applications beyond their primary use in biomass gasification. Deactivated zeolites can find potential applications in different fields by leveraging their inherent properties even after deactivation. Potential applications of deactivated natural zeolites include adsorbents for environmental remediation, construction materials, concrete additives, and nutrient delivery systems in agriculture.

The primary application of deactivated natural zeolites is as adsorbents for environmental remediation. Deactivated natural zeolites can be employed as adsorbents for environmental remediation. The porous structure of zeolites allows them to selectively adsorb pollutants such as heavy metals, organic contaminants, and dyes from wastewater or soil. Although their catalytic activity may be diminished, the high surface area and ion-exchange capacity of deactivated zeolites make them effective adsorbents. A study demonstrated high removal efficiencies of pollutants using deactivated zeolites, achieving over 90% removal of heavy metals from contaminated water. The use of deactivated zeolites in environmental remediation can contribute to pollution control and mitigation efforts [120].

Another potential application of deactivated natural zeolite catalysts is as construction materials and concrete additives. Deactivated natural zeolites can be incorporated as additives into construction materials and concrete. The high adsorption capacity and ion-exchange properties of

zeolites can provide benefits such as improved durability, reduced permeability, and enhanced moisture control in concrete. Deactivated zeolites can also contribute to the immobilization of harmful substances, such as heavy metals, in construction materials. The addition of deactivated zeolites to concrete can also enhance its mechanical strength and durability [216, 217]. Their incorporation as additives can lead to more sustainable and resilient construction.

Another intriguing potential application of deactivated natural zeolite catalysts is as nutrient delivery systems in agriculture. Deactivated natural zeolites can serve as nutrient delivery systems for agriculture. The porous structure of zeolites allows them to retain and slowly release essential plant nutrients, thereby promoting efficient nutrient uptake by crops. Deactivated zeolites can provide long-term nutrient availability, reduce fertilizer leaching, and improve nutrient-use efficiency. Studies have reported increased crop yields and improved nutrient retention in soils amended with deactivated zeolites [218]. A previous study demonstrated yield increases of up to 30% in certain crops using zeolite-based nutrient delivery systems [219, 220]. Their use in agriculture contributes to sustainable and environmentally friendly farming practices.

Another possible application of deactivated natural zeolite catalysts is as thermal insulation materials which can be utilized because of their low thermal conductivity. The porous structure of zeolites helps to trap air within their cavities, reducing heat transfer and improving insulation properties. The addition of deactivated zeolites to building materials can enhance thermal insulation efficiency [221]. Their application as thermal insulation materials can contribute to energy conservation and improve the building energy efficiency.

Therefore, the potential applications of deactivated natural zeolites in other fields highlight their versatility and value beyond their catalytic activity. These applications capitalize on the unique properties of zeolites, such as their adsorption capabilities, ion-exchange capacities, and porous structures. Further research and development efforts are required to explore and optimize the use of deactivated zeolites in these applications, promoting sustainable and innovative solutions for environmental remediation, construction, agriculture, and thermal insulation.

CONCLUSION

The investigation conducted a thorough exploration of the complex aspects surrounding zeolite catalyst deactivation. The utilization of zeolites in biomass gasification processes undergoes deactivation through various mechanisms such as abrasion, coke formation, and sintering, which significantly affect both synthetic and natural zeolites. Noteworthy findings indicate that the implementation of metal modifiers like nickel and cobalt can lead to reductions in coke yield of up to 60%. Additionally, innovative approaches, such as incorporating core-shell zeolite structures and integrating redox

metal oxides, demonstrated promising outcomes in preserving catalyst stability. Furthermore, regeneration methods like thermal treatment, chemical washing, and steaming showed considerable efficacy in removing coke, with efficiencies reaching up to 80%. Moreover, deactivated zeolites displayed remarkable versatility, exhibiting heavy metal removal rates exceeding 90% in environmental remediation applications and contributing to improvements in durability and permeability in construction materials. Despite these significant findings, our study has several limitations. First, the long-term stability of modified and regenerated zeolites under real-world operational conditions needs further investigation. Second, the economic feasibility of implementing these advanced modifications and regeneration methods at an industrial scale was not thoroughly evaluated. Finally, the environmental impact of the regeneration processes themselves warrants deeper exploration. Future research should focus on addressing these limitations by conducting extended field trials, comprehensive cost-benefit analyses, and environmental impact assessments. Additionally, exploring new metal modifiers, advanced regeneration techniques, and hybrid catalyst systems could further enhance the performance and sustainability of zeolite catalysts in biomass gasification. These findings highlight the potential of zeolite catalysts in sustainable biomass gasification processes and underscore the importance of ongoing research to optimize these strategies across diverse operational conditions, thereby advancing catalyst utilization and environmental sustainability.

ACKNOWLEDGEMENT

We express our sincere gratitude to the Rector of Universitas Sebelas Maret for their generous support in the form of funding grants (PKPTDN, Mandatory, PUT, and PTUPT) provided from 2021 to 2024.

REFERENCES

- [1] He M, Zhang K, Guan Y, Sun Y, Han B. Green carbon science: fundamental aspects. *Natl Sci Rev* 2023;nwad046. [\[Crossref\]](#)
- [2] Laidler KJ. A glossary of terms used in chemical kinetics, including reaction dynamics (IUPAC Recommendations 1996). *Pure Appl Chem* 1996;68:149–92. [\[Crossref\]](#)
- [3] Ranade V V., Joshi SS. *Catalysis and Catalytic Processes*. Elsevier Inc.; 2016. [\[Crossref\]](#)
- [4] Yan P, Wang H, Liao Y, Wang C. Zeolite catalysts for the valorization of biomass into platform compounds and biochemicals/biofuels: A review. *Renew Sustain Energy Rev* 2023;178:113219. [\[Crossref\]](#)
- [5] Zhang W, Taheri-Ledari R, Saeidirad M, Qazi FS, Kashtiaray A, Ganjali F, et al. Regulation of porosity in MOFs: A review on tunable scaffolds and related effects and advances in different applications. *J Environ Chem Eng* 2022;108836. [\[Crossref\]](#)
- [6] Kouser S, Hezam A, Khadri MJN, Khanum SA. A review on zeolite imidazole frameworks: Synthesis, properties, and applications. *J Porous Mater* 2022;29:663–81. [\[Crossref\]](#)
- [7] Qu H, Ma Y, Li B, Wang L. Hierarchical zeolites: Synthesis, structural control, and catalytic applications. *Emergent Mater* 2020;3:225–45. [\[Crossref\]](#)
- [8] Mashuri SIS, Ibrahim ML, Kasim MF, Mastuli MS, Rashid U, Abdullah AH, et al. Photocatalysis for organic wastewater treatment: From the basis to current challenges for society. *Catalysts* 2020;10:1–29. [\[Crossref\]](#)
- [9] Zhang H, bin Samsudin I, Jaenicke S, Chuah G-K (2022) Zeolites in catalysis: sustainable synthesis and its impact on properties and applications. *Catal Sci Technol* 12, 6024–39. [\[Crossref\]](#)
- [10] Verdoliva V, Saviano M, De Luca S. Zeolites as acid/basic solid catalysts: Recent synthetic developments. *Catalysts* 2019;9. [\[Crossref\]](#)
- [11] Nurliati G, Krisnandi YK, Sihombing R, Salimin Z. Studies of modification of zeolite by tandem acid-base treatments and its adsorptions performance towards thorium. *Atom Indones* 2015;41:87–95. [\[Crossref\]](#)
- [12] Ates A, Akgül G. Modification of natural zeolite with NaOH for removal of manganese in drinking water. *Powder Technol* 2016;287:285–91. [\[Crossref\]](#)
- [13] Fang Y, Yang F, He X, Zhu X (2019) Dealumination and desilication for Al-rich HZSM-5 zeolite via steam-alkaline treatment and its application in methanol aromatization. *Front Chem Sci Eng* 13, 543–53. [\[Crossref\]](#)
- [14] Yang S, Yu C, Yu L, Miao S, Zou M, Jin C, et al. Bridging dealumination and desilication for the synthesis of hierarchical MFI zeolites. *Angew Chemie - Int Ed* 2017;56:12553–6. [\[Crossref\]](#)
- [15] Ates A. Role of modification of natural zeolite in removal of manganese from aqueous solutions. *Powder Technol* 2014;264:86–95. [\[Crossref\]](#)
- [16] Soltanian S, Lee CL, Lam SS (2020) A review on the role of hierarchical zeolites in the production of transportation fuels through catalytic fast pyrolysis of biomass. *Biofuel Res J* 7, 1217–34. [\[Crossref\]](#)
- [17] Lin F, Xu M, Ramasamy KK, Li Z, Klinger JL, Schaidle JA, et al. Catalyst deactivation and its mitigation during catalytic conversions of biomass. *ACS Catal* 2022;12:13555–99. [\[Crossref\]](#)
- [18] Asadullah M, Tomishige K, Fujimoto K. A novel catalytic process for cellulose gasification to synthesis gas. *Catal Commun* 2001;2:63–8. [\[Crossref\]](#)
- [19] Tavares F, Mohamed HO, Kulkarni SR, Morlanés N, Castaño P. Decreasing the coking and deactivation of a reforming Ni-Ce/Al₂O₃ catalyst with intraparticle SiC in hydrogen production routes. *Fuel* 2023;337:127058. [\[Crossref\]](#)

- [20] Srinakruang J, Sato K, Vitidsant T, Fujimoto K. Highly efficient sulfur and coking resistance catalysts for tar gasification with steam. *Fuel* 2006;85:2419–26. [\[Crossref\]](#)
- [21] Puig-Gamero M, Lara-Díaz J, Valverde JL, Sanchez-Silva L, Sánchez P. Dolomite effect on steam co-gasification of olive pomace, coal and petcoke: TGA-MS analysis, reactivity and synergistic effect. *Fuel* 2018;234:142–50. [\[Crossref\]](#)
- [22] Sun Z, Toan S, Chen S, Xiang W, Fan M, Zhu M, et al. Biomass pyrolysis-gasification over Zr promoted CaO-HZSM-5 catalysts for hydrogen and bio-oil co-production with CO₂ capture. *Int J Hydrogen Energy* 2017;42:16031–44. [\[Crossref\]](#)
- [23] Yuan N, Tan K, Zhang X, Zhao A, Guo R. Synthesis and adsorption performance of ultra-low silica-to-alumina ratio and hierarchical porous ZSM-5 zeolites prepared from coal gasification fine slag. *Chemosphere* 2022;303:134839. [\[Crossref\]](#)
- [24] Valizadeh S, Jang SH, Hoon Rhee G, Lee J, Loke Show P, Ali Khan M, et al. Biohydrogen production from furniture waste via catalytic gasification in air over Ni-loaded Ultra-stable Y-type zeolite. *Chem Eng J* 2022;433:133793. [\[Crossref\]](#)
- [25] Yu L, Zhang R, Cao C, Liu L, Fang J, Jin H (2022) Hydrogen production from supercritical water gasification of lignin catalyzed by Ni supported on various zeolites. *Fuel* 319, 123744. [\[Crossref\]](#)
- [26] Mei Y, Zhang Q, Gao S, Xue Y, Wang Z. Zeolite preparation coupled with alkali recovery from catalytic gasification ash by one-step hydrothermal treatment. *Energy and Fuels* 2023;37:7911–8. [\[Crossref\]](#)
- [27] Porawati H, Kurniawan A, Yuliwati E. Effect of temperature on gasification of biomass using zeolit. *J. Phys. Conf. Ser.*, vol. 1845, 2021. [\[Crossref\]](#)
- [28] State RN, Volceanov A, Muley P, Boldor D. A review of catalysts used in microwave assisted pyrolysis and gasification. *Bioresour Technol* 2019;277:179–94. [\[Crossref\]](#)
- [29] Zhang Z, Liu L, Shen B, Wu C. Preparation, modification and development of Ni-based catalysts for catalytic reforming of tar produced from biomass gasification. *Renew Sustain Energy Rev* 2018;94:1086–109. [\[Crossref\]](#)
- [30] Kolodeznikov KY, Stepanov V V. Zeolites of Yakutia. *Stud. Surf. Sci. Catal.*, vol. 28, Elsevier; 1986, p. 93–100. [\[Crossref\]](#)
- [31] Weller MT, Dann SE. Hydrothermal synthesis of zeolites. *Curr Opin Solid State Mater Sci* 1998;3:137–43. [\[Crossref\]](#)
- [32] Jumaeri J, Santosa SJ, Sutarno, Kunarti ES. Synthesis of zeolite A from coal fly ash by alkali fusion and hydrothermal. *Adv Mater Res* 2014;1043:198–203. [\[Crossref\]](#)
- [33] Keith TEC, Staples LW. Zeolites in Eocene basaltic pillow lavas of the Siletz River volcanics, central Coast Range, Oregon. *Clays Clay Miner* 1985;33:135–44. [\[Crossref\]](#)
- [34] Lee Y Il. Chemistry and origin of zeolites in sandstones at DSDP Sites 445 and 446, Daito Ridge and Basin Province, northwest Pacific. *Chem Geol* 1988;67:261–73. [\[Crossref\]](#)
- [35] Caruso C, Nastro A. Synthesis of ZSM-5 and A zeolites on porous alumina substrates. *Stud. Surf. Sci. Catal.*, vol. 125, Elsevier; 1999, p. 93–100. [\[Crossref\]](#)
- [36] Zhu D, Wang L, Fan D, Yan N, Huang S, Xu S, et al. A bottom-up strategy for the synthesis of highly siliceous faujasite-type zeolite. *Adv Mater* 2020;32:2000272. [\[Crossref\]](#)
- [37] Król M, Florek P. Zeolites. MDPI, Basel; 2022.
- [38] Suyitno, Gravitiani E, Arifin Z, Muqoffa M, Hadi S. Feasibility of electric generation from municipal solid wastes by incineration and gasification. *Proc. 6th Int. Conf. Exhib. Sustain. Energy Adv. Mater. ICE-SEAM 2019*, 16–17 Oct. 2019, Surakarta, Indones., Springer; 2020, p. 485–91. [\[Crossref\]](#)
- [39] Imron R, Suyitno S, Ilyas AX, Faishal A, Budiono A, Yusuf M, et al. Producing hydrogen-rich syngas via microwave heating and co-gasification: a systematic review. *Biofuel Res J* 2022;9:1573–91. [\[Crossref\]](#)
- [40] Cortazar M, Santamaria L, Lopez G, Alvarez J, Zhang L, Wang R, et al. A comprehensive review of primary strategies for tar removal in biomass gasification. *Energy Convers Manag* 2023;276. [\[Crossref\]](#)
- [41] Susastriawan AAP, Purwanto Y, Sidharta BW, Siolimbona N. Thermal performance of cocoa pod cook stove. *J Therm Eng* 2021;10:188–95. [\[Crossref\]](#)
- [42] Rauch R, Hrbek J, Hofbauer H. Biomass gasification for synthesis gas production and applications of the syngas. *Adv Bioenergy Sustain Chall* 2016:73–91. [\[Crossref\]](#)
- [43] Mardiana S, Azhari NJ, Ilmi T, Kadja GTM. Hierarchical zeolite for biomass conversion to biofuel: A review. *Fuel* 2022;309:122119. [\[Crossref\]](#)
- [44] Antil S, Sachdeva G, Sharma A. Advancements and challenges in the fluidized bed gasification system: A comprehensive review. *J Therm Eng* 2023;9:233–46. [\[Crossref\]](#)
- [45] El-Rub ZA, Halawa D, Alqudah I, Nasr A, Naqvi M, Abu El-Rub Z, et al. Natural zeolite catalyst for tar removal in biomass gasification Systems: Kinetics and effectiveness evaluation. *Fuel* 2023;346:128393. [\[Crossref\]](#)
- [46] Inglezakis VJ, Zorpas AA. Handbook of natural zeolites. Bentham Science Publishers; 2012. [\[Crossref\]](#)
- [47] Waluyo J, Ruya PM, Hantoko D, Rizkiana J, Makertiharta I, Yan M, et al. Utilization of modified zeolite as catalyst for steam gasification of palm kernel shell. *Bull Chem React Eng Catal* 2021;16:623–31. [\[Crossref\]](#)
- [48] Hauserman WB. High-yield hydrogen production by catalytic gasification of coal or biomass. *Int J Hydrogen Energy* 1994;19:413–9. [\[Crossref\]](#)

- [49] Asadullah M, Ito SI, Kunimori K, Yamada M, Tomishige K. Energy efficient production of hydrogen and syngas from biomass: Development of low-temperature catalytic process for cellulose gasification. *Environ Sci Technol* 2022;**36**:4476–81. [\[Crossref\]](#)
- [50] Liu YL, Huang X, Ren J, Zhao X-Y, Cao J-P. Low-Temperature Reforming of Biomass Tar over Ni/ZSM-5 Catalysts: Unraveling the H₂-Rich Gas Production Pathways Using in Situ and Ex Situ Techniques. *Ind Eng Chem Res* 2022;**61**:5734–46. [\[Crossref\]](#)
- [51] Pan X, Jiao F, Miao D, Bao X. Oxide–zeolite-based composite catalyst concept that enables syngas chemistry beyond Fischer–Tropsch synthesis. *Chem Rev* 2021;**121**:6588–609. [\[Crossref\]](#)
- [52] Wang M, Kang J, Xiong X, Zhang F, Cheng K, Zhang Q, et al. Effect of zeolite topology on the hydrocarbon distribution over bifunctional ZnAlO/SAPO catalysts in syngas conversion. *Catal Today* 2021;**371**:85–92. [\[Crossref\]](#)
- [53] Gao N, Milandile MH, Quan C, Rundong L. Critical assessment of plasma tar reforming during biomass gasification: A review on advancement in plasma technology. *J Hazard Mater* 2022;**421**:126764. [\[Crossref\]](#)
- [54] Ren J, Cao J-P, Zhao X-Y, Yang F-L, Wei X-Y. Recent advances in syngas production from biomass catalytic gasification: A critical review on reactors, catalysts, catalytic mechanisms and mathematical models. *Renew Sustain Energy Rev* 2019;**116**:109426. [\[Crossref\]](#)
- [55] Ngo TNLT, Chiang K-Y, Liu C-F, Chang Y-H, Wan H-P. Hydrogen production enhancement using hot gas cleaning system combined with prepared Ni-based catalyst in biomass gasification. *Int J Hydrogen Energy* 2021;**46**:11269–83. [\[Crossref\]](#)
- [56] Waluyo J, Ruya PM, Hantoko D, Rizkiana J, Makertihartha I, Yan M, Susanto H (2021) Utilization of Modified Zeolite as Catalyst for Steam Gasification of Palm Kernel Shell. *Bull Chem React Eng Catal* **16**, 623–31. [\[Crossref\]](#)
- [57] Maryudi M, Aktawan A, Amelia S. Water scrubber and zeolite catalyst for clean syngas production on biomass gasification of bagasse in a downdraft system. *J Bahan Alam Terbarukan* 2022;**11**:92–9. [\[Crossref\]](#)
- [58] Liu P, Chen Z, Li X, Chen W, Li Y, Sun T, et al. Enhanced degradation of VOCs from biomass gasification catalyzed by Ni/HZSM-5 series catalyst. *J Environ Manage* 2023;**345**. [\[Crossref\]](#)
- [59] Singh A, Shivapuji AM, Dasappa S. VPSA process characterization for ISO quality green hydrogen generation using two practical multi-component biomass gasification feeds. *Sep Purif Technol* 2023;**315**:123667. [\[Crossref\]](#)
- [60] Zsinka V, Miskolczi N, Juzsakova T, Jakab M. Pyrolysis-gasification of biomass using nickel modified catalysts: The effect of the catalyst regeneration on the product properties. *J Energy Inst* 2022;**105**:16–24. [\[Crossref\]](#)
- [61] Dastyar W, Raheem A, Zhao M, Yuan W, Li H, Ting ZJ. Effects of ionic liquid-assisted pretreatment of heavy metal-contaminated biomass on the yield and composition of syngas production using noncatalytic and catalytic pyrolysis and gasification processes. *ACS Sustain Chem Eng* 2019;**7**:18303–12. [\[Crossref\]](#)
- [62] Kislov VM, Salganskii EA, Tsvetkov M V, Tsvetkova YY. Effect of catalysts on the yield of products formed in biomass gasification. *Russ J Appl Chem* 2017;**90**:716–20. [\[Crossref\]](#)
- [63] Laksmono N, Paraschiv M, Loubar K, Tazerout M. Biodiesel production from biomass gasification tar via thermal/catalytic cracking. *Fuel Process Technol* 2013;**106**:776–83. [\[Crossref\]](#)
- [64] Phillips SD, Tarud JK, Biddy MJ, Dutta A. Gasoline from woody biomass via thermochemical gasification, methanol synthesis, and methanol-to-gasoline technologies: A technoeconomic analysis. *Ind Eng Chem Res* 2011;**50**:11734–45. [\[Crossref\]](#)
- [65] Buchireddy PR, Bricka RM, Rodriguez J, Holmes W. Biomass gasification: catalytic removal of tars over zeolites and nickel supported zeolites. *Energy & Fuels* 2010;**24**:2707–15. [\[Crossref\]](#)
- [66] Thao Ngo TNL, Chiang KY. Hydrogen sulfide removal from simulated synthesis gas using a hot gas cleaning system. *J Environ Chem Eng* 2023;**11**:109592. [\[Crossref\]](#)
- [67] Hus J. Experimental verification of a pilot pyrolysis/split product gasification (PSPG) unit. *Energy* 2022;**244**:1–9. [\[Crossref\]](#)
- [68] Kim I-T, Ahn K-H, Jung J, Jeong Y, Shin D-C, Lee Y-E. Removal of tar contents derived from lignocellulosic biomass gasification facilities using MgAl-LDH@clinoptilolite. *Catalysts* 2021;**11**:1111. [\[Crossref\]](#)
- [69] Chipera SJ, Bish DL. Thermodynamic modeling of natural zeolite stability. Los Alamos National Lab. (LANL), Los Alamos, NM (United States); 1997.
- [70] H. Abbasi, F. Rahimpour, F. Pourfayaz AK. Evaluating integration of biomass gasification process with solid oxide fuel cell and torrefaction process. *J Therm Eng* 2019;**5**:230–9. [\[Crossref\]](#)
- [71] Güney OF, Koyun A. Experimental analysis and kinetic modelling for steam gasification of the Turkish Lignites. *J Therm Eng* 2020;**6**:204–13. [\[Crossref\]](#)
- [72] Acevedo-Paez JC, Arenas-Castiblanco E, Posso F, Alarcón E, Villa AL, Jahromi H, et al. Effect of calcium and potassium on activity of mordenite-supported nickel catalyst for hydrogen production from biomass gasification. *Int J Hydrogen Energy* 2023. [\[Crossref\]](#)

- [73] Huang X, Ma M, Li M, Shen W. Regulating the location of framework aluminium in mordenite for the carbonylation of dimethyl ether. *Catal Sci Technol* 2020;10:7280–90. [\[Crossref\]](#)
- [74] Woolcock PJ, Brown RC. A review of cleaning technologies for biomass-derived syngas. *Biomass and Bioenergy* 2013;52:54–84. [\[Crossref\]](#)
- [75] Mohamed DKB, Veksha A, Ha QLM, Chan WP, Lim T-T, Lisak G. Advanced Ni tar reforming catalysts resistant to syngas impurities: Current knowledge, research gaps and future prospects. *Fuel* 2022;318:123602. [\[Crossref\]](#)
- [76] Torres W, Pansare SS, Goodwin Jr JG. Hot gas removal of tars, ammonia, and hydrogen sulfide from biomass gasification gas. *Catal Rev* 2007;49:407–56. [\[Crossref\]](#)
- [77] Jae J, Tompsett GA, Foster AJ, Hammond KD, Auerbach SM, Lobo RF, et al. Investigation into the shape selectivity of zeolite catalysts for biomass conversion. *J Catal* 2011;279:257–68. [\[Crossref\]](#)
- [78] Mertens G, Snellings R, Balen K Van, Bicer-Simsir B, Verlooy P, Elsen J. Pozzolanic reactions of common natural zeolites with lime and parameters affecting their reactivity. *Cem Concr Res* 2009;39:233–40. [\[Crossref\]](#)
- [79] Saramok M, Inger M, Antoniuk-Jurak K, Szymaszek-Wawryca A, Samojeden B, Motak M. physicochemical features and NH₃-SCR catalytic performance of natural zeolite modified with iron—the effect of Fe loading. *Catalysts* 2022;12. [\[Crossref\]](#)
- [80] Lari GM, Dapsens PY, Scholz D, Mitchell S, Mondelli C, Pérez-Ramírez J. Deactivation mechanisms of tin-zeolites in biomass conversions. *Green Chem* 2016;18:1249–60. [\[Crossref\]](#)
- [81] Cadar O, Dinca Z, Senila M, Becze A, Todor F. Studies on the modification of some natural zeolite from NW Romania after acid and basic treatments. *Int Multidiscip Sci GeoConference SGEM* 2020;20:309–16. [\[Crossref\]](#)
- [82] Petranovskii V, Chaves-Rivas F, Espinoza MAH, Pestryakov A, Kolobova E. Potential uses of natural zeolites for the development of new materials: short review. *MATEC Web Conf.*, vol. 85, EDP Sciences; 2016, p. 1014. [\[Crossref\]](#)
- [83] Santi LP, Prakoso HT, Kalbuadi DN. Preliminary study of molecular sieve materials to alleviate problems faced by tropical peatland. *IOP Conf. Ser. Earth Environ. Sci.*, vol. 1025, IOP Publishing; 2022, p. 12027. [\[Crossref\]](#)
- [84] Froment GF (2008) Kinetic modeling of hydrocarbon processing and the effect of catalyst deactivation by coke formation. *Catal Rev* 50, 1–18. [\[Crossref\]](#)
- [85] Van Kooten WEJ, Krijnsen HC, Van Den Bleek CM, Calis HPA. Deactivation of zeolite catalysts used for NO_x removal. *Appl Catal B Environ* 2000;25:125–35. [\[Crossref\]](#)
- [86] Guan G, Kaewpanha M, Hao X, Abudula A. Catalytic steam reforming of biomass tar: Prospects and challenges. *Renew Sustain energy Rev* 2016;58:450–61. [\[Crossref\]](#)
- [87] Silaghi MC, Chizallet C, Raybaud P. Challenges on molecular aspects of dealumination and desilication of zeolites. *Microporous Mesoporous Mater* 2014;191:82–96. [\[Crossref\]](#)
- [88] Cruciani G. Zeolites upon heating: Factors governing their thermal stability and structural changes. *J Phys Chem Solids* 2006;67:1973–94. [\[Crossref\]](#)
- [89] Al-Shawabkeh AF, Al-Najdawi N, Olimat AN. High purity oxygen production by pressure vacuum swing adsorption using natural zeolite. *Results Eng* 2023;18:101119. [\[Crossref\]](#)
- [90] Chikati R. *Iron Supported on Clinoptilolite (Natural Zeolites) as a Low-Temperature Fischer-Tropsch Synthesis Catalyst*, University of the Witwatersrand, Faculty of Engineering and the Built.
- [91] Guisnet M, Magnoux P, Martin D. Roles of acidity and pore structure in the deactivation of zeolites by carbonaceous deposits. In: *Studies in Surface Science and Catalysis*, vol 111, Elsevier, pp 1–19.
- [92] Ates A, Hardacre C (2012) The effect of various treatment conditions on natural zeolites: Ion exchange, acidic, thermal and steam treatments. *J Colloid Interface Sci* 1997;372:130–40. [\[Crossref\]](#)
- [93] Hulea V, Huguet E, Cammarano C, Lacarriere A, Durand R, Leroi C, et al. Conversion of methyl mercaptan and methanol to hydrocarbons over solid acid catalysts – A comparative study. *Appl Catal B Environ* 2014;144:547–53. [\[Crossref\]](#)
- [94] Zhao X, Li J, Tian P, Wang L, Li X, Lin S, et al. Achieving a superlong lifetime in the zeolite-catalyzed MTO reaction under high pressure: synergistic effect of hydrogen and water. *ACS Catal* 2019;9:3017–25. [\[Crossref\]](#)
- [95] Bai J, Liu S, Xie S, Xu L, Lin L. Shape selectivity in methane dehydroaromatization over Mo/MCM-22 catalysts during a lifetime experiment. *Catal Letters* 2003;90:123–30. [\[Crossref\]](#)
- [96] Milina M, Mitchell S, Crivelli P, Cooke D, Pérez-Ramírez J. Mesopore quality determines the lifetime of hierarchically structured zeolite catalysts. *Nat Commun* 2014;5:3922. [\[Crossref\]](#)
- [97] Mitchell S, Boltz M, Liu J, Pérez-Ramírez J. Engineering of ZSM-5 zeolite crystals for enhanced lifetime in the production of light olefins via 2-methyl-2-butene cracking. *Catal Sci Technol* 2017;7:64–74. [\[Crossref\]](#)
- [98] Hwang A, Kumar M, Rimer JD, Bhan A. Implications of methanol disproportionation on catalyst lifetime for methanol-to-olefins conversion by HSSZ-13. *J Catal* 2017;346:154–60. [\[Crossref\]](#)
- [99] Zhou J, Gao M, Zhang J, Liu W, Zhang T, Li H, et al. Directed transforming of coke to active intermediates in methanol-to-olefins catalyst to boost light olefins selectivity. *Nat Commun* 2021;12:17. [\[Crossref\]](#)

- [100] Li Z. New micro and mesoporous materials for the reaction of methanol to olefins 2014.
- [101] Martínez A, Lopez C. The influence of ZSM-5 zeolite composition and crystal size on the in situ conversion of Fischer–Tropsch products over hybrid catalysts. *Appl Catal A Gen* 2005;294:251–9. [\[Crossref\]](#)
- [102] Martín AJ, Mitchell S, Mondelli C, Jaydev S, Pérez-Ramírez J (2022) Unifying views on catalyst deactivation. *Nat Catal* 5, 854–66. [\[Crossref\]](#)
- [103] Morales–Leal FJ, Ancheyta J, Torres–Mancera P, Alonso F. Experimental methodologies to perform accelerated deactivation studies of hydrotreating catalysts. *Fuel* 2023;332:126074. [\[Crossref\]](#)
- [104] Adananche DE, Aliyu A, Atta AY, El-Yakubu BJ. Residue fluid catalytic cracking: A review on the mitigation strategies of metal poisoning of RFCC catalyst using metal passivators/traps. *Fuel* 2023;343:127894. [\[Crossref\]](#)
- [105] Bai P, Etim UJ, Yan Z, Mintova S, Zhang Z, Zhong Z, et al. Fluid catalytic cracking technology: current status and recent discoveries on catalyst contamination. *Catal Rev* 2019;61:333–405. [\[Crossref\]](#)
- [106] Li N, Chen C, Wang B, Li S, Yang C, Chen X. Retardation effect of nitrogen compounds and condensed aromatics on shale oil catalytic cracking processing and their characterization. *Appl Petrochemical Res* 2015;5:285–95. [\[Crossref\]](#)
- [107] Skorupska M, Ilnicka A, Lukaszewicz JP. The effect of nitrogen species on the catalytic properties of N-doped graphene. *Sci Rep* 2021;11:1–11. [\[Crossref\]](#)
- [108] Bobkova T V., Doronin VP, Potapenko O V., Sorokina TP, Ostrovskii NM. Poisoning effect of nitrogen compounds on the transformation of model hydrocarbons and real feed under catalytic cracking conditions. *Catal Ind* 2014;6:218–22. [\[Crossref\]](#)
- [109] Kordala N, Wyszowski M. Zeolite properties, methods of synthesis, and selected applications. *Molecules* 2024;29. [\[Crossref\]](#)
- [110] Ruiz-Martínez J, Buurmans ILC, Knowles W V, Van Der Beek D, Bergwerff JA, Vogt ETC, Weckhuysen BM (2012) Microspectroscopic insight into the deactivation process of individual cracking catalyst particles with basic sulfur components. *Appl Catal A Gen* 419–420, 84–94. [\[Crossref\]](#)
- [111] Leflaive P, Lemberon JL, Pérot G, Mirgain C, Carriat JY, Colin JM (2002) On the origin of sulfur impurities in fluid catalytic cracking gasoline - Reactivity of thiophene derivatives and of their possible precursors under FCC conditions. *Appl Catal A Gen* 227, 201–15. [\[Crossref\]](#)
- [112] Arevalo RL, Aspera SM, Nakanishi H (2019) Sulfation of a PdO(101) methane oxidation catalyst: Mechanism revealed by first principles calculations. *Catal Sci Technol* 9, 232–40. [\[Crossref\]](#)
- [113] Li Y, Guo Q, Dai Z, Dong Y, Yu G, Wang F. Study of oxidation for gas mixture of H₂S and CH₄ in a non-premixed flame under oxygen deficient condition. *Appl Therm Eng* 2017;117:659–67. [\[Crossref\]](#)
- [114] Yang Z, Liu J, Zhang L, Zheng S, Guo M, Yan Y. Catalytic combustion of low-concentration coal bed methane over CuO/γ-Al₂O₃ catalyst: Effect of SO₂. *RSC Adv* 2014;4:39394–9. [\[Crossref\]](#)
- [115] Honkanen M, Wang J, Kärkkäinen M, Huuhtanen M, Jiang H, Kallinen K, et al. Regeneration of sulfur-poisoned Pd-based catalyst for natural gas oxidation. *J Catal* 2018;358:253–65. [\[Crossref\]](#)
- [116] Zhang Y, Glarborg P, Andersson MP, Johansen K, Torp TK, Jensen AD, et al. Influence of the support on rhodium speciation and catalytic activity of rhodium-based catalysts for total oxidation of methane. *Catal Sci Technol* 2020;10:6035–44. [\[Crossref\]](#)
- [117] Wallenstein D, Farmer D, Knoell J, Fougret CM, Brandt S. Progress in the deactivation of metals contaminated FCC catalysts by a novel catalyst metallation method. *Appl Catal A Gen* 2013;462:91–9. [\[Crossref\]](#)
- [118] Li D, Zhu Q, Bao Z, Jin L, Hu H. New insight and countermeasure for sulfur poisoning on nickel-based catalysts during dry reforming of methane. *Fuel* 2024;363:131045. [\[Crossref\]](#)
- [119] Lisi L, Cimino S. Poisoning of SCR catalysts by alkali and alkaline earth metals. *Catalysts* 2020;10:1475. [\[Crossref\]](#)
- [120] Belviso C. Zeolite for potential toxic metal uptake from contaminated soil: A brief review. *Processes* 2020;8:820. [\[Crossref\]](#)
- [121] Zhang Y, Dong J, Guo F, Shao Z, Wu J. Zeolite synthesized from coal fly ash produced by a gasification process for Ni²⁺ removal from water. *Minerals* 2018;8:116. [\[Crossref\]](#)
- [122] Xu M, Mukarakate C, Robichaud DJ, Nimlos MR, Richards RM, Trewyn BG. Elucidating zeolite deactivation mechanisms during biomass catalytic fast pyrolysis from model reactions and zeolite syntheses. *Top Catal* 2016;59:73–85. [\[Crossref\]](#)
- [123] Ihli J, Jacob RR, Holler M, Guizar-Sicairos M, Diaz A, Da Silva JC, et al. A three-dimensional view of structural changes caused by deactivation of fluid catalytic cracking catalysts. *Nat Commun* 2017;8:809. [\[Crossref\]](#)
- [124] Hunston C, Baudouin D, Tarik M, Kröcher O, Vogel F. Investigating active phase loss from supported ruthenium catalysts during supercritical water gasification. *Catal Sci Technol* 2021;11:7431–44. [\[Crossref\]](#)
- [125] Magyarová Z, Králik M, Soták T. Utilization of zeolite catalysts in biomass exploitation: a mini-review. *Monatshefte Für Chemie - Chem Mon* 2023;154:815–35. [\[Crossref\]](#)
- [126] Bingre R, Louis B, Nguyen P. An overview on zeolite shaping technology and solutions to overcome diffusion limitations. *Catalysts* 2018;8. [\[Crossref\]](#)

- [127] Xie Y, Zhang Y, He L, Jia CQ, Yao Q, Sun M, et al. Anti-deactivation of zeolite catalysts for residue fluid catalytic cracking. *Appl Catal A Gen* 2023;657:119159. [\[Crossref\]](#)
- [128] Sun Y, Wei L, Zhang Z, Zhang H, Li Y. Coke formation over zeolite catalysts in light alkanes aromatization and anti-carbon-deposition strategies and perspectives: a review. *Energy & Fuels* 2023;37:1657–77. [\[Crossref\]](#)
- [129] Nazarova GY, Ivashkina EN, Ivanchina ED, Mezkhova MY. A model of catalytic cracking: catalyst deactivation induced by feedstock and process variables. *Catalysts* 2022;12:98. [\[Crossref\]](#)
- [130] Hambali HU, Jalil AA, Abdulrasheed AA, Siang TJ, Gambo Y, Umar AA. Zeolite and clay based catalysts for CO₂ reforming of methane to syngas: a review. *Int J Hydrogen Energy* 2022;47:30759–87. [\[Crossref\]](#)
- [131] Samoilov NA. Mechanism of catalytic deactivation of zeolites in multicycle processes. *React Kinet Catal Lett* 1982;19:5–9. [\[Crossref\]](#)
- [132] Arcoya A, Seoane XL, Soria J. Effect of iron on the deactivation of Ni/clinoptilolite catalysts by thiophene poisoning. *J Chem Technol Biotechnol Int Res Process Environ Clean Technol* 1997;68:171–6. [\[Crossref\]](#)
- [133] Arcoya A, Seoane XL, Soria J. Sulfur resistance of nickel catalysts supported on K-clinoptilolite containing iron in ethylbenzene hydrogenation. *Stud Surf Sci Catal* 1993;75:2341–4. [\[Crossref\]](#)
- [134] Yaşyerli S, Ar İ, Dogu G, Dogu T. Removal of hydrogen sulfide by clinoptilolite in a fixed bed adsorber. *Chem Eng Process* 2002;41:785–92. [\[Crossref\]](#)
- [135] Daneshvar M, Falamaki C. Fixed-bed catalytic oxidative removal of dissolved iron by manganese oxide-coated clinoptilolite: enhanced activity in the presence of aqueous Mn²⁺ cations. *Environ Process* 2018;5:65–79. [\[Crossref\]](#)
- [136] Lee C-Y, Ha B-H. Deactivation of CuO/mordenite by the breakage of mordenite crystal through the H₂/O₂ cycle treatment. *Stud. Surf. Sci. Catal.*, vol. 126, Elsevier; 1999, p. 203–10. [\[Crossref\]](#)
- [137] Stöcker M, Holm KM. Improved deactivation profile observed for Pt/H-chabazites and Pt/H-Y zeolite. *React Kinet Catal Lett* 1990;41:271–5. [\[Crossref\]](#)
- [138] Yuan Y, Lee JS, Lobo RF. Ga⁺-chabazite zeolite: A highly selective catalyst for nonoxidative propane dehydrogenation. *J Am Chem Soc* 2022. [\[Crossref\]](#)
- [139] Cui Y, Xu Y, Lu J, Suzuki Y, Zhang Z-G. The effect of zeolite particle size on the activity of Mo/HZSM-5 in non-oxidative methane dehydroaromatization. *Appl Catal A Gen* 2011;393:348–58. [\[Crossref\]](#)
- [140] Gao X, Wen Y, Tan R, Huang H, Kawi S. A review of catalyst modifications for a highly active and stable hydrogen production from methane. *Int J Hydrogen Energy* 2023;48:6204–32. [\[Crossref\]](#)
- [141] Hutchings GJ, Themistocleous T, Copperthwaite RG. Methanol conversion to hydrocarbons using modified clinoptilolite catalysts: Investigation of catalyst lifetime and reactivation. *Appl Catal* 1988;43:133–40. [\[Crossref\]](#)
- [142] Miro EE, Costa L, Dereppe J-M, Petunchi JO. H-mordenite deactivation during the SCR of NO_x. Adsorption and diffusion of probe molecules on fresh and deactivated catalysts. *Stud. Surf. Sci. Catal.*, vol. 111, Elsevier; 1997, p. 231–8. [\[Crossref\]](#)
- [143] Ribotta A, Lezcano M, Kurgansky M V, Miroè E, Lombardo EA, Petunchi JO, et al. Kinetics, acid sites and deactivation of H-mordenite during the SCR of NO_x with CH₄. *Catal Letters* 1997;49:77–85. [\[Crossref\]](#)
- [144] Kunal P, Toops TJ, Kidder MK, Lance MJ. Deactivation trends of Pd/SSZ-13 under the simultaneous presence of NO, CO, hydrocarbons and water for passive NO_x adsorption. *Appl Catal B-Environmental* 2021;299:120591. [\[Crossref\]](#)
- [145] Aziz MAA, Jalil AA, Wongsakulphasatch S, Vo D-VN. Understanding the role of surface basic sites of catalysts in CO₂ activation in dry reforming of methane: a short review. *Catal Sci Technol* 2020;10:35–45. [\[Crossref\]](#)
- [146] Orlyk S, Kyriienko P, Kapran A, Chedryk V, Balakin D, Gurgul J, et al. CO₂-assisted dehydrogenation of propane to propene over Zn-BEA zeolites: Impact of acid–base characteristics on catalytic performance. *Catalysts* 2023;13:681. [\[Crossref\]](#)
- [147] Orlyk SM, Kapran AY, Chedryk VI, Nychiporuk YM, Kyriienko PI, Dzwigaj S. Effect of surface acidity/basicity of Cr/Zn–BEA zeolite catalysts on performance in CO₂-PDH process. *ChemistrySelect* 2024;9:e202304801. [\[Crossref\]](#)
- [148] Yuan B, Zhu T, Han Y, Zhang X, Wang M, Li C. Deactivation mechanism and anti-deactivation measures of metal catalyst in the dry reforming of methane: A review. *Atmosphere (Basel)* 2023;14:770. [\[Crossref\]](#)
- [149] Xing S, Turner S, Fu D, van Vreeswijk S, Liu Y, Xiao J, et al. Silicalite-1 layer secures the bifunctional nature of a CO₂ hydrogenation catalyst. *JACS Au* 2023;3:1029–38. [\[Crossref\]](#)
- [150] Ivanov DP, Kharitonov AS, Piryutko L V. Phenol oxidation by nitrous oxide: The role of zeolite catalyst acidity. *Catal Ind* 2015;7:275–81. [\[Crossref\]](#)
- [151] Graca I, Comparot J-D, Laforge S, Magnoux P, Lopes JM, Ribeiro ME, et al. Influence of phenol addition on the H-ZSM-5 zeolite catalytic properties during methylcyclohexane transformation. *Energy & Fuels* 2009;23:4224–30. [\[Crossref\]](#)
- [152] Xu X, Zhang Y, Li X, Xia X, Jiang H, Toghan A. Comparative study on the catalytic behaviors of zeolites with different diffusion limitation in ethane aromatization. *Microporous Mesoporous Mater* 2021;315:110926. [\[Crossref\]](#)

- [153] Cheng Y, Hoard J, Lambert C, Kwak JH, Peden CHF. NMR studies of Cu/zeolite SCR catalysts hydrothermally aged with urea. *Catal Today* 2008;136:34–9. [\[Crossref\]](#)
- [154] Bach-Oller A, Furusjö E, Umeki K. On the role of potassium as a tar and soot inhibitor in biomass gasification. *Appl Energy* 2019;254:113488. [\[Crossref\]](#)
- [155] Barrientos J, Montes V, Boutonnet M, Järås S. Further insights into the effect of sulfur on the activity and selectivity of cobalt-based Fischer–Tropsch catalysts. *Catal Today* 2016;275:119–26. [\[Crossref\]](#)
- [156] Querini CA, Roa E. Deactivation of solid acid catalysts during isobutane alkylation with C4 olefins. *Appl Catal A Gen* 1997;163:199–215. [\[Crossref\]](#)
- [157] Wang L, Li D, Koike M, Watanabe H, Xu Y, Nakagawa Y, et al. Catalytic performance and characterization of Ni–Co catalysts for the steam reforming of biomass tar to synthesis gas. *Fuel* 2013;112:654–61. [\[Crossref\]](#)
- [158] Huang H, Tang X, Haas M. In-situ continuous coke deposit removal by catalytic steam gasification for fuel-cooled thermal management. *J Eng Gas Turbines Power-Transactions Asme* 2012;134:101502. [\[Crossref\]](#)
- [159] Hui L. Function of fluidized bed reactor in reforming of methane with CO₂ to syngas in the presence of oxygen. *Ind Catal* 2008.
- [160] Beltramini JN, Datta R. Gasification of Carbon Deposited on Mono and Bi-metallic Reforming Catalysts. *Chemeca 88 Aust. Bicenten. Int. Conf. Process Ind. Prepr. Pap., Australia: Barton, ACT: Institution of Engineers; 1988.*
- [161] Lachén J, Herguido J, Peña JÁ. High purity hydrogen from biogas via steam iron process: Preventing reactor clogging by interspersed coke combustions. *Renew Energy* 2020;151:619–26. [\[Crossref\]](#)
- [162] Das S, Pérez-Ramírez J, Gong J, Dewangan N, Hidajat K, Gates BC, et al. Core–shell structured catalysts for thermocatalytic, photocatalytic, and electrocatalytic conversion of CO₂. *Chem Soc Rev* 2020;49:2937–3004. [\[Crossref\]](#)
- [163] Kawi S, Ashok J, Dewangan N, Pati S, Junmei C. Recent advances in catalyst technology for biomass tar model reforming: Thermal, plasma, and membrane reactors. *Waste and Biomass Valorization* 2022;13:1–30. [\[Crossref\]](#)
- [164] Doronin VP, Sorokina TP, Lipin P V, Potapenko O V, Korotkova N V, Gordenko VI. Development and introduction of zeolite containing catalysts for cracking with controlled contents of rare earth elements. *Catal Ind* 2015;7:12–6. [\[Crossref\]](#)
- [165] Hosseinpour M, Ahmadi SJ, Fatemi S. Successive co-operation of supercritical water and silica-supported iron oxide nanoparticles in upgrading of heavy petroleum residue: Suppression of coke deposition over catalyst. *J Supercrit Fluids* 2015;100:70–8. [\[Crossref\]](#)
- [166] Anggoro DD, Buchori L, Putra MF, Le Monde BU. Regeneration method for spent FCC catalysts: Brief. *J Res Chem* 2023;4:49–53. [\[Crossref\]](#)
- [167] Valle B, Aramburu B, Olazar M, Bilbao J, Gayubo AG. Steam reforming of raw bio-oil over Ni/La₂O₃-AAI₂O₃: Influence of temperature on product yields and catalyst deactivation. *Fuel* 2018;216:463–74. [\[Crossref\]](#)
- [168] Barbera K, Sørensen S, Bordiga S, Skibsted J, Fordsmand H, Beato P, et al. Role of internal coke for deactivation of ZSM-5 catalysts after low temperature removal of coke with NO₂. *Catal Sci Technol* 2012;2:1196–206. [\[Crossref\]](#)
- [169] Cao E, Zheng Y, Zhang H, Wang J, Li Y, Zhu T, et al. In-situ regenerable Cu/Zeolite adsorbent with excellent H₂S adsorption capacity for blast furnace gas. *Sep Purif Technol* 2024;336:126305. [\[Crossref\]](#)
- [170] Daligaux V, Richard R, Manero MH (2021) Deactivation and regeneration of zeolite catalysts used in pyrolysis of plastic wastes—a process and analytical review. *Catalysts* 11. doi:10.3390/catal11070770. [\[Crossref\]](#)
- [171] Galadima A, Muraza O. Stability improvement of zeolite catalysts under hydrothermal conditions for their potential applications in biomass valorization and crude oil upgrading. *Microporous Mesoporous Mater* 2017;249:42–54. [\[Crossref\]](#)
- [172] Sun P, Chen J, Zai S, Gao S, Weng X, Wu Z. Regeneration mechanism of a deactivated zeolite-supported catalyst for the combustion of chlorinated volatile organic compounds. *Catal Sci Technol* 2021;11:923–33. [\[Crossref\]](#)
- [173] Horne PA, Nugranad N, Williams PT. The influence of steam on the zeolite catalytic upgrading of biomass pyrolysis oils. *Dev Thermochem Biomass Convers Vol 1/Volume 2* 1997:648–56. [\[Crossref\]](#)
- [174] Díaz M, Epelde E, Valecillos J, Izaddoust S, Aguayo AT, Bilbao J (2021) Coke deactivation and regeneration of HZSM-5 zeolite catalysts in the oligomerization of 1-butene. *Appl Catal B Environ* 291. [\[Crossref\]](#)
- [175] Cao E, Zheng Y, Zhang H, Wang J, Li Y, Zhu T, Zhang Z, Xu G, Cui Y (2024) In-situ regenerable Cu/Zeolite adsorbent with excellent H₂S adsorption capacity for blast furnace gas. *Sep Purif Technol* 336, 126305. [\[Crossref\]](#)
- [176] Leardini L, Martucci A, Braschi I, Blasioli S, Quartieri S. Regeneration of high-silica zeolites after sulfamethoxazole antibiotic adsorption: A combined in situ high-temperature synchrotron X-ray powder diffraction and thermal degradation study. *Mineral Mag* 2014;78:1141–59. [\[Crossref\]](#)
- [177] Zhang Y, Williams PT. Carbon nanotubes and hydrogen production from the pyrolysis catalysis or catalytic-steam reforming of waste tyres. *J Anal Appl Pyrolysis* 2016;122:490–501. [\[Crossref\]](#)

- [178] Lee D-G, Kim J-H, Lee C-H. Adsorption and thermal regeneration of acetone and toluene vapors in dealuminated Y-zeolite bed. *Sep Purif Technol* 2011;77:312–24. [\[Crossref\]](#)
- [179] Hussain I, Jalil AA, Alhooshani K, Alasiri H, Malaibari Z, Hassan NS, et al. CO methanation over highly active and coke-resistant ruthenium-doped fibrous mordenite zeolite catalyst for synthetic natural gas (SNG) production. *J Energy Inst* 2023;108:101230. [\[Crossref\]](#)
- [180] Zhang Q, Zhao X, Zheng Y, Xiao Y, Li J, Liu F, et al. Facile synthesis of regenerable NaY zeolite adsorbent relying on alcohol-terminated compounds for efficient trace sulfur dioxide capture. *Chem Eng J* 2023;475:146265. [\[Crossref\]](#)
- [181] Zhang W, Zhou Z, An Y, Du S, Ruan D, Zhao C, et al. Optimization for zeolite regeneration and nitrogen removal performance of a hypochlorite-chloride regenerant. *Chemosphere* 2017;178:565–72. [\[Crossref\]](#)
- [182] Wu SL, Kuo JH, Wey MY. Hydrogen promotion by Co/SiO₂@HZSM-5 core-shell catalyst for syngas from plastic waste gasification: The combination of functional materials. *Int J Hydrogen Energy* 2019;44:13480–9. [\[Crossref\]](#)
- [183] Kim S, Sasmaz E, Lauterbach J. Effect of Pt and Gd on coke formation and regeneration during JP-8 cracking over ZSM-5 catalysts. *Appl Catal B Environ* 2015;168–169:212–9. [\[Crossref\]](#)
- [184] Kumar R, Ohtani S, Tsunogi N. Direct air capture on amine-impregnated FAU zeolites: Exploring for high adsorption capacity and low-temperature regeneration. *Microporous Mesoporous Mater* 2023;360:112714. [\[Crossref\]](#)
- [185] Andrunik M, Skalny M, Gajewska M, Marzec M, Bajda T. Comparison of pesticide adsorption efficiencies of zeolites and zeolite-carbon composites and their regeneration possibilities. *Heliyon* 2023;9:e20572. [\[Crossref\]](#)
- [186] Küntzel J, Ham R, Melin T. Regeneration of hydrophobic zeolites with steam. *Chem Eng Technol* 1999;22:991–4. [\[Crossref\]](#)
- [187] Schulz H, Wei M. Deactivation and thermal regeneration of zeolite HZSM-5 for methanol conversion at low temperature (260–290°C). *Microporous Mesoporous Mater* 1999;29:205–18. [\[Crossref\]](#)
- [188] Rodeghero E, Martucci A, Cruciani G, Sarti E, Cavazzini A, Costa V, et al. Detailed investigation of thermal regeneration of high-silica ZSM-5 zeolite through in situ synchrotron X-ray powder diffraction and adsorption studies. *J Phys Chem C* 2017;121:17958–68. [\[Crossref\]](#)
- [189] Santiago-Colón ÁN, Gounder R. Structural changes to molybdenum and Brønsted acid sites in MFI zeolites during methane dehydroaromatization reaction-regeneration cycles. *J Catal* 2024;430. [\[Crossref\]](#)
- [190] Kariim I, Swai H, Kivevele T. Bio-oil upgrading over ZSM-5 catalyst: A review of catalyst performance and deactivation. *Int J Energy Res* 2023;2023. [\[Crossref\]](#)
- [191] Lutz W. Zeolite Y: Synthesis, modification, and properties—a case revisited. *Adv Mater Sci Eng* 2014;2014:724248. [\[Crossref\]](#)
- [192] Hita I, Mohamed HO, Attada Y, Zambrano N, Zhang W, Ramírez A, et al. Direct analysis at temporal and molecular level of deactivating coke species formed on zeolite catalysts with diverse pore topologies. *Catal Sci Technol* 2023;13:1288–300. [\[Crossref\]](#)
- [193] Muhammad I, Makwashi N, Ahmed TG, Manos G, Zhao D. A mechanistic model on catalyst deactivation by coke formation in a CSTR reactor. *Processes* 2023;11. [\[Crossref\]](#)
- [194] Bukhtiyarova M V, Echevskii G V. Coke formation on zeolites Y and their deactivation model. *Pet Chem* 2020;60:532–9. [\[Crossref\]](#)
- [195] Liu D, Slocombe M, AlKinany H, AlMegren J, Wang J, Arden A, et al. Advances in the study of coke formation over zeolite catalysts in the methanol-to-hydrocarbon process. *Appl Petrochemical Res* 2016;6:209–15. [\[Crossref\]](#)
- [196] Gu Y, Zhu Q, Liu Z, Fu C, Wu J, Zhu Q, et al. Nitrogen reduction reaction energy and pathways in metal-zeolites: deep learning and explainable machine learning with local acidity and hydrogen bonding features. *J Mater Chem A* 2022;10:14976–88. [\[Crossref\]](#)
- [197] Liu M, Miao C, Wu Z. Recent advances in the synthesis, characterization, and catalytic consequence of metal species confined within zeolite for hydrogen-related reactions. *Ind Chem Mater* 2024. [\[Crossref\]](#)
- [198] Luo HY, Lewis JD, Román-Leshkov Y. Lewis acid zeolites for biomass conversion: Perspectives and challenges on reactivity, synthesis, and stability. *Annu Rev Chem Biomol Eng* 2016;7:663–92. [\[Crossref\]](#)
- [199] Shan J, Li Z, Zhu S, Liu H, Li J, Wang J, et al. Nanosheet MFI Zeolites for gas phase glycerol dehydration to acrolein. *Catalysts* 2019;9:13–27. [\[Crossref\]](#)
- [200] Bartholomew CH, Argyle MD. Advances in catalyst deactivation and regeneration. *Catalysts* 2015;5:949–54. [\[Crossref\]](#)
- [201] Ammendola P, Chirone R, Ruoppolo G, Russo G. Regeneration strategies of deactivated catalysts for thermo-catalytic decomposition process in a fluidized bed reactor. *Combust Sci Technol* 2008;180:869–82. [\[Crossref\]](#)
- [202] Baier S, Damsgaard CD, Klumpp M, Reinhardt J, Sheppard T, Balogh Z, et al. Stability of a bifunctional Cu-based core@zeolite shell catalyst for dimethyl ether synthesis under redox conditions studied by environmental transmission electron microscopy and in situ X-ray ptychography. *Microsc Microanal* 2017;23:501–12. [\[Crossref\]](#)

- [203] Gorshkov AS, Sineva L V, Gryaznov KO, Asalieva EY, Mordkovich VZ. Deactivation and regeneration of a zeolite-containing cobalt catalyst in a Fischer-Tropsch synthesis reactor. *Catal Ind* 2023;15:152–64. [\[Crossref\]](#)
- [204] Padovan D, Nakajima K, Hensen EJM. Metal oxide catalysts for the valorization of biomass-derived sugars BT - Crystalline metal oxide catalysts. In: Ueda W, editor., Singapore: Springer Nature Singapore; 2022, p. 325–47. [\[Crossref\]](#)
- [205] Costa P, Pinto F, André RN, Marques P. Integration of gasification and solid oxide fuel cells (SOFCs) for combined heat and power (CHP). *Processes* 2021;9. [\[Crossref\]](#)
- [206] Botti L, Padovan D, Navar R, Tolborg S, Martinez-Espin JS, Hammond C. Thermal regeneration of Sn-containing silicates and consequences for biomass upgrading: From regeneration to preactivation. *ACS Catal* 2020;10:11545–55. [\[Crossref\]](#)
- [207] Ruoppolo G, Landi G. Towards biomass gasification enhanced by structured iron-based catalysts. *Fuels* 2021;2:546–55. [\[Crossref\]](#)
- [208] Rubinsin NJ, Timmiati SN, Lim KL, Isahak WNRW, Karim NA. Gasification reaction on CeO₂(111) and effects on the structural and electronic properties of adsorption molecules. *Chem Eng Technol* 2024..
- [209] Huo J, Pham HN, Cheng Y, Lin H-H, Roling LT, Datye AK, Shanks BH (2020) Deactivation and regeneration of carbon supported Pt and Ru catalysts in aqueous phase hydrogenation of 2-pentanone. *Catal Sci & Technol* 10, 3047–56. [\[Crossref\]](#)
- [210] Lu F, Wang Q, Zhu M, Dai B (2023) Deactivation and Regeneration of Nitrogen Doped Carbon Catalyst for Acetylene Hydrochlorination. *Molecules* 28. doi:10.3390/molecules28030956. [\[Crossref\]](#)
- [211] Yung MM, Starace AK, Mukarakate C, Crow AM, Leshnov MA, Magrini KA. Biomass catalytic pyrolysis on Ni/ZSM-5: effects of nickel pretreatment and loading. *Energy & Fuels* 2016;30:5259–68. [\[Crossref\]](#)
- [212] Xu Z, Park ED. Recent advances in coke management for dry reforming of methane over Ni-based catalysts. *Catalysts* 2024;14. [\[Crossref\]](#)
- [213] Luan H, Wu Q, Zhang J, Wang Y, Meng X, Xiao F-S. Sustainable synthesis of core-shell structured ZSM-5@silicalite-1 zeolite. *Chem Res Chinese Univ* 2022;38:136–40. [\[Crossref\]](#)
- [214] Xu H, Wu P. New progress in zeolite synthesis and catalysis. *Natl Sci Rev* 2022;9:nwac045. [\[Crossref\]](#)
- [215] Dyer AC, Nahil MA, Williams PT. Biomass:polystyrene co-pyrolysis coupled with metal-modified zeolite catalysis for liquid fuel and chemical production. *J Mater Cycles Waste Manag* 2022;24:477–90. [\[Crossref\]](#)
- [216] Najimi M, Sobhani J, Ahmadi B, Shekarchi M. An experimental study on durability properties of concrete containing zeolite as a highly reactive natural pozzolan. *Constr Build Mater* 2012;35:1023–33. [\[Crossref\]](#)
- [217] Khoshroo M, Javid AAS, Katebi A. Effect of chloride treatment curing condition on the mechanical properties and durability of concrete containing zeolite and micro-nano-bubble water. *Constr Build Mater* 2018;177:417–27. [\[Crossref\]](#)
- [218] Colombani N, Di Giuseppe D, Faccini B, Ferretti G, Mastrocicco M, Coltorti M. Estimated water savings in an agricultural field amended with natural zeolites. *Environ Process* 2016;3:617–28. [\[Crossref\]](#)
- [219] de Campos Bernardi AC, Oliviera PPA, de Melo Monte MB, Souza-Barros F. Brazilian sedimentary zeolite use in agriculture. *Microporous Mesoporous Mater* 2013;167:16–21. [\[Crossref\]](#)
- [220] Ramesh V, Jyothi JS, Shibli SMA. Effect of zeolites on soil quality, plant growth and nutrient uptake efficiency in sweet potato (*Ipomoea batatas* L.). *J Root Crop* 2015;41:25–31.
- [221] Borchardt L, Michels N-L, Nowak T, Mitchell S, Perez-Ramirez J. Structuring zeolite bodies for enhanced heat-transfer properties. *Microporous Mesoporous Mater* 2015;208:196–202. [\[Crossref\]](#)



Review Article

Factors affecting and methods of reducing thermal stratification in cryogenic storage tanks of launch vehicles: review article

Krish V. RAIBOLE¹ , Puskaraj D SONAWWANAY^{1,*}

¹Department of Mechanical Engineering, Dr. Vishwanath Karad MIT World Peace University, 411038 Pune, India

ARTICLE INFO

Article history

Received: 22 June 2024

Revised: 23 October 2024

Accepted: 27 October 2024

Keywords:

Cryogenic Propellants; Launch Vehicles; Liquid Rocket Engines; Thermal Insulation; Thermal Stratification

ABSTRACT

Contemporary launch vehicles of the past few decades primarily implement liquid rocket engines, which in turn employ cryogenic propellants stored in sub-zero conditions in highly sophisticated cryogenic storage tanks. These tanks are usually deprived of any thermal insulation in order to prioritize the payload capacity, and thus they are prone to a substantial amount of heat in-flux that leads to a rise in the temperature of the cryogenic liquid, leading to thermal stratification. This paper presents a comprehensive review of the various factors affecting the rate of thermal stratification in cryogenic propellant storage tanks, along with numerous experimental and numerical techniques developed for controlling or mitigating this stratification through geometrical modifications, varying surface properties, and bubbling of gases through the bulk liquid. Out of the techniques reviewed, simple geometrical modifications showed substantial results, with ribs reducing stratification by up to 30%. On the other hand, complex techniques like bubbling of gases destratified the bulk liquid within 25 to 35 s. A special focus has also been placed on reviewing the numerical modelling and simulations of this phenomenon, particularly those developed in recent years.

Cite this article as: Raibole Krish V, Sonawwanay Puskaraj D. Factors affecting and methods of reducing thermal stratification in cryogenic storage tanks of launch vehicles: review article. J Ther Eng 2025;11(5):1585–1599.

INTRODUCTION

Nearly every sophisticated launch vehicle of the 20th century employs cryogenic propulsion systems [1]. Therefore, understanding the complications involved in such systems is of utmost importance. One such difficulty arises in the storage of such low temperature liquids. Storage and transportation of such low temperature cryogenic liquids has always been one of the primary challenges in its application [2,3]. Cryogenic liquids, with temperatures in the range of -187 °C

to -210 °C, are primarily stored inside cylindrical tanks that receive heat from the surroundings, causing a subsequent change in density and temperature variance between the bulk and liquid adjacent to the walls [4–6]. This leads to the formation of convective currents, wherein warm layers, comparatively less dense, rise up and accumulate near the liquid vapour boundary, resulting in the formation of a temperature rise along the height of the tank as illustrated in Fig. 1 [7,8]. The accumulated liquid near the interface is called ‘thermal

*Corresponding author.

*E-mail address: puskaraj.sonawwanay@mitwpu.edu.in

This paper was recommended for publication in revised form by Editor-in-Chief Ahmet Selim Dalkılıç



stratified liquid', and this phenomenon is called 'thermal stratification'. Apart from thermal stratification occurring due to convection currents developed in the bulk liquid, heat flux from the hotter ullage can also augment thermal stratification. This phenomenon has also been reported to persist in launch vehicles in both, sea-level and micro-gravity levels [9–11]. Furthermore, the pressure of the vapor phase, known as the ullage (Refer Fig. 1), also dictates the degree of stratification [12,13]. Fig. 1 depicts the phenomenon of thermal stratification in a cryogenic storage tank, wherein, under the action of buoyancy force, natural upward convection currents are developed, which result in the formation of boundary layers along the tank walls [14,15]. T_S , T_B & T_U stand for the temperature of the stratified layer, bulk liquid and ullage, respectively, and their typical ranges in a cryogenic storage tank are shown in Fig.1.

Thermal stratification of the fuel, typically LH_2 , is a highly unfavourable phenomenon that affects the functioning of the cryogenic engine, even if the temperature difference is only of a few degrees [16–18]. Any degree of stratification, if present, affects the system negatively [19]. It results in an increase in the pressure of the vessel, conforming to the temperature of the warmer upper layers of the fuel (LH_2). To maintain the required tank pressure, the vapor formed due to stratification has to be vented out. In the case of LH_2 as the fuel, its stratification would lead to diminished lock-up times, implying that the time required for the vaporized LH_2 gas pressure to reach the tank's pressure limit would be reduced; and the warmer

stratified column of LH_2 would source cavitation inside the LH_2 pumps [20,21]. Subsequently, a plethora of techniques have been implemented in the past to counter stratification, such as: baffles, fins, and ribs on the tank's inner wall [22–24], mixers [25–27], bubbling of gases [28], etc. These techniques have been discussed in detail in further sections.

Owing to the importance of comprehending thermal stratification and a lack of detailed review articles on the same, the present study provides a comprehensive review of research spanning over 60 years involving a wide array of techniques employed to mitigate thermal stratification in cryogenic liquids. The present study explores numerous destratification techniques employed which have not been comprehensively reviewed before. The scope of this study involves delineating various factors that influence thermal stratification in cryogenic liquids, followed by the different numerical and experimental approaches adopted by researchers in the past half-century. The scope of this study is not only limited to the stratification dynamics involved in the storage tanks of launch vehicles, but is also applicable to the transportation and storage of cryogenic liquids in the thermal, power and energy sector. Furthermore, experimental techniques employed to mitigate stratification have also been extensively reviewed. Special care has also been taken to include recent trends observed in studies involving thermal stratification. The present study, will not only help researchers and readers to get an in-depth picture of the works conducted on the thermal stratification in the last half a century, but also aid in highlighting research efforts necessary to be taken in the future.

FACTORS AFFECTING THERMAL STRATIFICATION

The following section enumerates various real-world parameters that significantly affect the degree of stratification in storage tanks of LVs. Owing to their importance, only the parameters which are experienced by a LV during its mission, have been explained in detail below.

Effect of Rotation of Launching Vehicle

After lift-off, just after the LV clears the launch tower, a rolling motion of the LV about its vertical axis is initiated through thrust vectoring or, in the case of sounding rockets, by the use of canted fins, which produce aerodynamic forces. These forces are applied at the centre of pressure of the rocket, which is at some distance from the rocket's centre of gravity, resulting in a torque (or moment) about the major axes, as seen in Fig. 2. This torque causes the rocket to rotate about the axis, initiating a roll manoeuvre [29,30].

The rotation, in turn, has an effect on the motion of the fluid inside the tanks. It is assumed that the liquid has a solid body rotation and takes a paraboloid shape, resulting in an increased contact surface area between the liquid and the tank walls, as seen in Fig. 3 [31–33]. Fig. 3 represents a similar case wherein the gravitational force is perfectly aligned with the rocket, acting vertically downward. It is

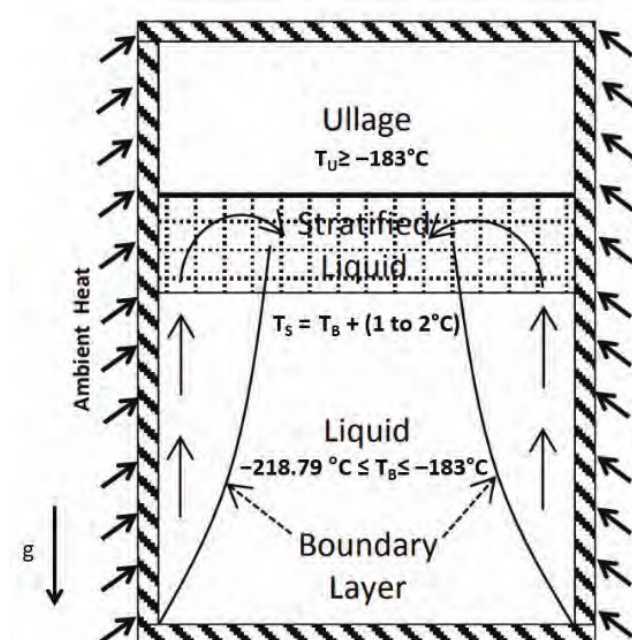


Figure 1. Representation of liquid stratification phenomenon inside a cryogenic tank [From Agarwal et al. [14], with permission from IOPscience.]

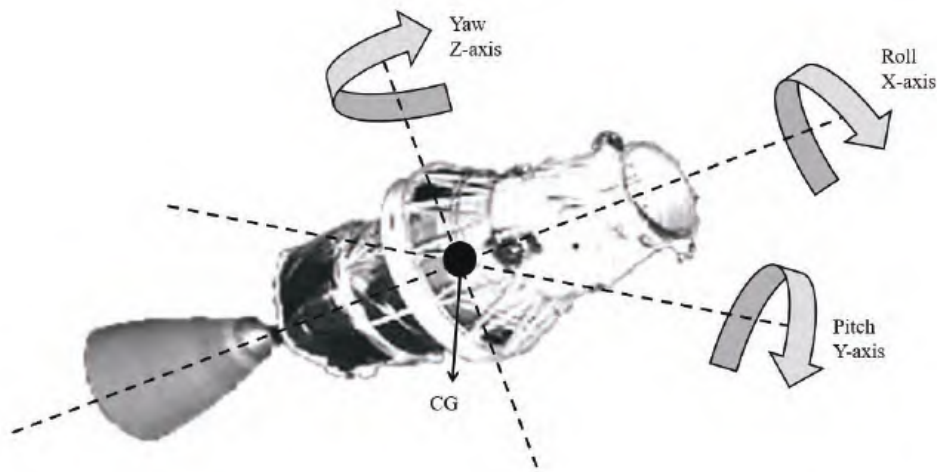


Figure 2. Principal axis of rotation of the LV [From Oliveira et al. [29], with permission from NASA NTRS]

evident from Fig. 3 that the liquid at the center of rotation dips down by a height equal to the height of the liquid that rises at the rim, given by $(h/2)$, where h represents the distance between the lowest and highest points of liquid at the surface. The liquid within the tank is observed to have a similar significant dishing effect [14]. Fig. 4 depicts a combined model of rotation and stratification, illustrating the effect of different gravity levels on the shape attained by the liquid fuel (LH_2) surface. This displays how a parabolic surface of the liquid results in the liquid gaining height along the tank wall, resulting in a higher net heat transfer from the wall to the liquid as illustrated in Fig. 3 & 4. At a spin rate of $\text{GD} = 1^\circ/\text{sec}$ and at reduced gravity levels of up to $g/g_0 = 10^{-5}$ the liquid is observed to be parabolic in shape, as seen in Fig. 4 [34]. The resultant tank temperature always tends to be higher because of the larger ΔT , with increasing rotation. Rotation has a large impact on stratification as it is observed to reduce the time required for stratification of

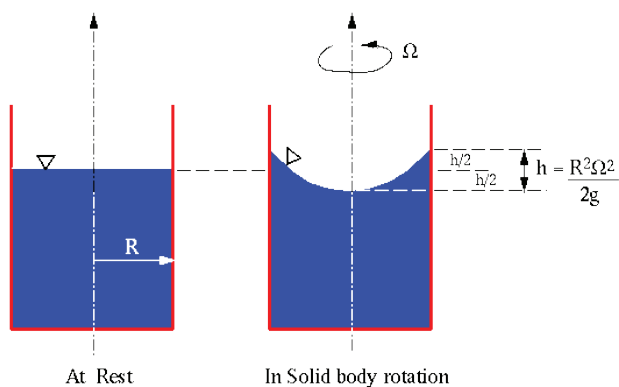


Figure 3. Paraboloid of revolution liquid free surface [From Cimbal et al. [83], with permission Penn State University.]

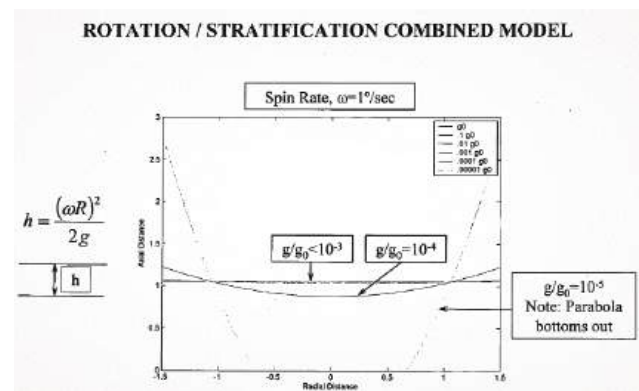


Figure 4. Rotation/Stratification combined model [From Oliveira et al. [29], with permission from NASA NTRS.].

LH_2 stored in a square tank of 3 m in diameter at 20% fill level at 16 K and 206843 Pa with a heat flux of 10 W/m^2 by 30 to 60 minutes during the 4 hr coast phase of the LV, which is rotating at $\text{GD} = 1^\circ/\text{sec}$ in $g/g_0 = 10^{-4}$. Overall, the effect of rotation on stratification for the boundary conditions mentioned above is observed as follows:

- Rotation reduces period to stratification by 15%.
- Rotation intensifies stratification temperature by 1.0 K [29,30,32,33].

Effect of Insulation Thickness

The insulation thickness of a cryogenic tank wall is a critical specification that influences the entire mission of the LV [35]. Excess insulation thickness results in extra weight added to the LV, hindering its efficiency. On the other hand, insulation thickness that is lower than what is required, results in excess heat flux into the liquid from outside the tank [36,37]. This, in turn, increases the stratification rate

of the fluid as seen in Fig.5 [38]. The optimum insulation thicknesses of several types of materials, like Extruded polystyrene (EXS), Expanded polystyrene (EPS), and rock wool, have been determined in the past and show promising results in limiting the heat in-flux from the side walls of cylindrical tanks [39,40].

Thermal stratification in the liquid and stratified mass growth inside a LH_2 storage tank in ambient conditions, which is of diameter 4 m, 7 m in length, and a wall thickness of 4 mm, for varied thickness of insulation, can be observed in Fig. 5, wherein the temperature of the pressurant gas used was 50 K. The thickness of the foam-based insulation varies from 10 mm to 40 mm, as 10 mm, 20 mm, 30 mm and 40 mm whereas the liquid fill level was kept constant at 87% of the total tank height [38]. The heat in-leak for tanks of different insulation thicknesses is observed in Fig.6. Heat

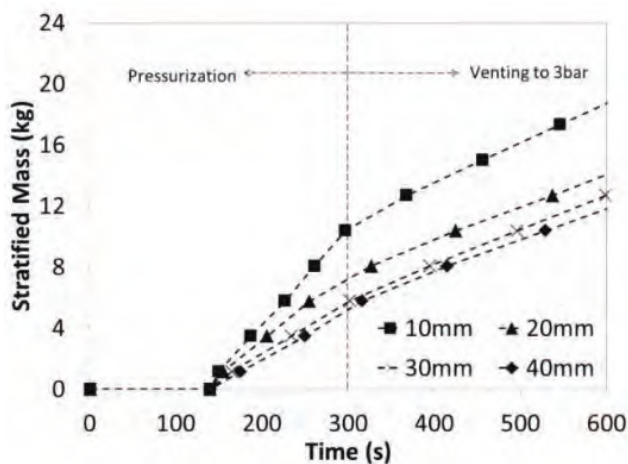


Figure 5. Stratified mass (kg) Vs. Time (s) [From Joseph et al. [38], with permission from Elsevier.].

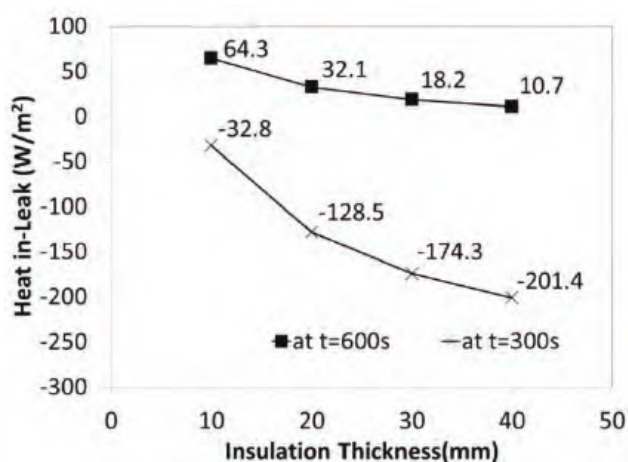


Figure 6. Heat in-leak (W/m^2) Vs. Insulation Thickness (mm) [From Joseph et al. [38], with permission from Elsevier.].

in-leak is plotted for two different times, $t = 300$ s, which covers the time taken for pressurisation, and $t = 600$ s, which covers an extended time of 300 s after pressurisation is completed.

It is evident from Fig. 6 that the tank with the highest insulation thickness of 40 mm records a heat in-leak of only $10.7 W/m^2$. Ullage temperature, as reported for all cases of insulation thickness, was in the range of 21 to 45 K before pressurisation. Thus, the warm pressurant gas entering the tank at 50 K leads to an increase in the ullage gas temperature. This results in the ullage gas temperature being higher than the wall temperature, culminating in a net heat loss to the tank wall. This is exhibited in Fig. 6, wherein the heat in-leak for $t = 300$ s is negative [38]. Lesser thickness of tank insulation leads to higher stratified mass due to higher liquid heat in-leak from the surroundings, triggering a payload penalty in propellant tanks used in launch vehicles as observed for a thickness of 10 mm, which resulted in a heat in-leak of $64.3 W/m^2$, as shown in Fig. 6 [38,41].

Effect of Pressurization

Pressurization occurs in two stages, the first being wherein the tank is pressurised to reach the required tank pressure. While the second stage involves maintaining the required tank pressure with the use of pressurant gas. Similarly, it is observed that throughout pressurization the stratified mass increases at a quicker rate compared to that during the fixed pressure scenario post pressurization (Fig. 5) [42].

This is due to the fact that during the pressurization phase, there is a continuous increase in the liquid-vapor interface temperature corresponding to the saturation temperature at that pressure, which is a result of the ullage temperature being higher during pressurisation [43]. During this phase, the stirring effect produced by the pressurant gas results in an enhanced forced convective heat transfer

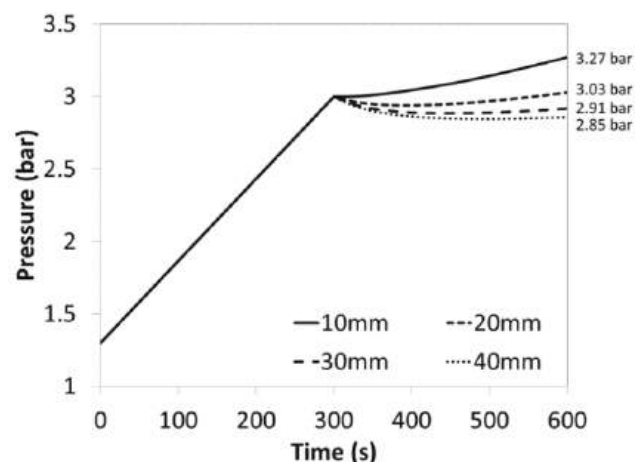


Figure 7. Pressure rise for different insulation thicknesses during and post pressurisation [From Joseph et al. [38], with permission from Elsevier.].

coefficient between the ullage and the liquid-vapor interface. Due to this, there is an increment in the conductive heat flux from the liquid-vapor interface to the bulk liquid inside the tank, as illustrated in Fig. 7. Apart from this, there is an additional temperature rise due to the ambient heat in-leak into the tank in spite of the insulation layer [44–51].

Fig. 7 depicts how the ullage pressure rises during pressurisation ($t = 300$ s) and after pressurisation is completed ($t = 600$ s). The cooling of ullage gas after pressurisation accounts for the pressure drop after $t = 300$ s, as seen in Fig. 7. A subsequent rise in pressure is observed due to heat in flux from the tank wall to the ullage, which dominates its cooling rate [38]. Hence, for lower insulation thicknesses, there is a greater heat in flux, leading to a higher pressure rise than thicker insulations as illustrated in Fig. 7. Hence, it is detected that tank pressure has a noteworthy role in the rate of thermal stratification or stratified mass evolution. Higher tank pressure causes more mass of the liquid to be stratified, and vice versa.

Similarly, in 2019, Vishnu et al. [52] evaluated the development of thermal stratification, experimentally and numerically, and its effects on tank self-pressurization, in a 10 L stainless steel tank of 1400 mm height and 108 mm in diameter, filled with liquid Nitrogen up to a height of 660 mm. The tank wall consisted of a total of 30 layers of fiberglass paper and foils of aluminium placed alternatively. An outer vessel of 400 mm in diameter served as a vacuum heat shield to minimize the heat in-leak. Stratification was measured using PT100 temperature sensors, which were placed along the height of the tank. The experimentation was carried out under two tank conditions, namely venting and non-venting. Venting implied that the vent valve was kept open all throughout the experiment, whereas in the non-venting case, the valve remains shut, leading to

the tank's self-pressurization. For the venting case, negligible thermal stratification was observed even after a time of 2000 seconds, with a maximum temperature difference of 1.03 K between the uppermost and the bottommost layer of the liquid. On the other hand, the non-venting case showed a temperature difference equivalent to 6 K, which can be attributed to the rise in pressure of the ullage. It was also reported that the rate of thermal stratification at the gas-liquid interface was greater than the stratification rate of the bottommost layer. This is the result of an ascending convection current formed within the bulk liquid due to the effect of buoyancy.

For the non-venting case, as the tank pressure was increased from 1 to 4 bar, stratification was observed to increase throughout the liquid as evident from Fig. 8. The interface temperatures for the pressures of 1, 2, 3 and 4 bar, were reported to be 79.8 K, 81.3 K, 83.3 K and 87.4 K, respectively, as seen in Fig. 8 [52]. A stratification parameter (λ) was established in an attempt to quantify the degree of stratification. λ is given by Eq. (1):

$$\lambda = \frac{T_8 - T_1}{T_0} \quad (1)$$

Wherein, T_8 and T_1 stand for the temperature readings recorded by the eighth sensor and first sensor from the tank's bottom, respectively [52]. In total, 16 sensors are placed along the height, with eight lying at the liquid-gas interface. For the venting case, this parameter stayed near constant and averaged at a value of 0.015 after a duration of 25 min. Whereas, in the case of the non-venting condition, λ increased from 0.0275 at $t=0$ min and reached a value of 0.045 after 25 min. Furthermore, Vishnu et al. [52] also developed a numerical model to further understand the complexities of the thermodynamics involved.

NUMERICAL MODELLING

As the numerical modelling schemes and techniques improve over time, research in the domain of aerospace and thermodynamics has gradually inclined more and more towards incorporating these numerical simulations in order to better understand the underlying complexities involved. A similar trend has been observed in evaluating the thermal stratification occurring in cryogenic tanks [53].

Various mixing techniques, such as sprinklers [54], rotating tank lids [55], porous structures [56–58] and nozzle jets [59], have been experimentally and numerically investigated in the past and have exhibited encouraging results in disrupting the natural convective flows in the bulk liquid. Similarly, in 2024, Brodnick et al. [60] numerically modelled and validated an axial jet mixer in order to attain a homogeneous temperature across the cryogenic propellant. The aim of using a jet mixer within the propellant tank is to induce a swirling current within the bulk fluid, which results in enhanced forced convection leading to a more homogenous temperature of the propellant along with a

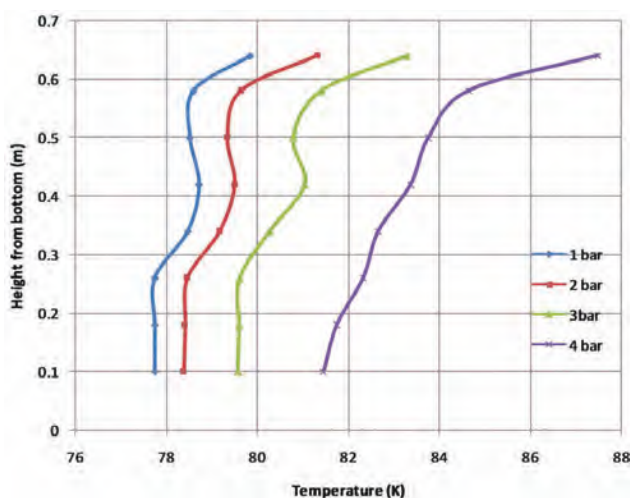


Figure 8. Development of stratification by varying tank pressure from 1 to 4 bar [From Vishnu et al. [52], with permission from Taylor & Francis.]

reduced ullage pressure. Fig.9 (a) shows the tank design, and Fig.9 (b) illustrates the computational domain (1.2 major-to-minor axis ratio with a major diameter of 2.2 m) incorporating the axial jet mixer. The jet mixer was modelled such that its tip had a rounded lip design as illustrated in Fig.9 (a). A fill level of 86% of the tank's total volume was used in this study [60].

Loci/STREAM was the CFD code employed by Brodnick et al. [60] and served as a pressure-based solver for the simulations. Instead of applying the VoF (Volume of Fluid) method, an interface model known as the sharp interface model, specially designed for the Loci/STREAM solver was implemented. A total of 119,700 cells were used to discretize the domain while solving equations with 1st and 2nd order accuracy in time and space, respectively, with a time step size equal to 0.1 s. Buoyancy within the liquid phase was modelled by employing the Boussinesq approximation. For modelling the turbulence, the 2 equation $k-\omega$ SST model was chosen. A heat flux of 4.2 W/m² was provided to the tank walls as an initial boundary condition. The effect of jet flow rate was studied by varying the flow rate from 1.82 m³/hr to 3.47 m³/hr [60].

Fig. 10 demonstrates the destratification results obtained for the jet mass flow rate of 3.47 m³/hr. From Fig.10 it can be clearly observed that, after a time of 8 min has passed, due to the mixing induced by the jet, thermal stratification within the liquid is virtually nullified. As anticipated, the case with the higher jet flow rate provided better and more promising results than the low flow rate case. From the initial ullage pressure of 186.1 kPa, the higher jet mass flow rate took nearly 8.3 min to drop the ullage pressure to a pressure of 128.4 kPa as represented in Fig. 10. Whereas, the lower jet mass flow rate took more than 20 min to accomplish the same pressure drop [60].

Similarly, in 2024, Raj et al. [61] numerically analyzed the rate of thermal destratification by bubbling of gases in liquid storage tanks with the help of a newly developed OpenFOAM solver, interThermalDestratificationFoam, that was specially designed to solve thermal stratification

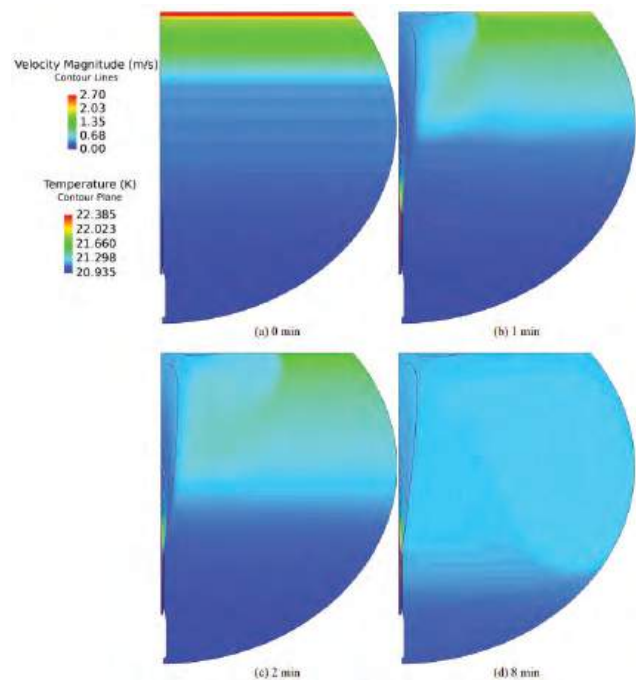


Figure 10. Temperature contour of the propellant fill level at a jet mass flow rate of 3.47 m³/hr [From Brodnick et al. [60], with permission from NASA NTRS.]

problems. In this study, primarily two cases were compared, both in 2D and 3D: a tank with one orifice and a tank with two orifices. To obtain a comprehensive understanding of the decay in stratification and its effect on gas bubble dynamics, fluid properties such as interfacial tension force, liquid viscosity and density were varied as shown in Table 1.

Fig. 11 shows the 2D computational domain with a singular orifice used by Raj et al. [61]. The rectangular tank measured 100 mm by 50 mm with a fill level up to a height of 85 mm. The orifice had a diameter of 0.3 mm and in the case of 2 orifices, they were placed 12.5 mm apart from each other. In order to satisfy the CFL (Courant-Frederick-Levy) criteria,

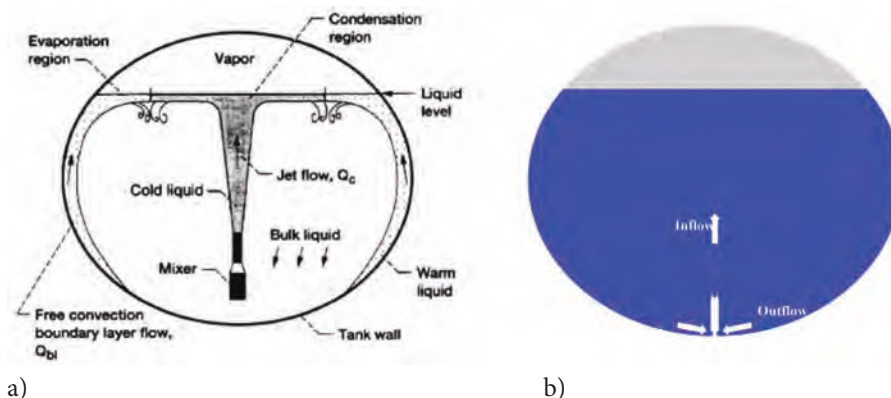


Figure 9. Cryogenic tank design with the axial jet mixer [From Brodnick et al. [60], with permission from NASA NTRS.]

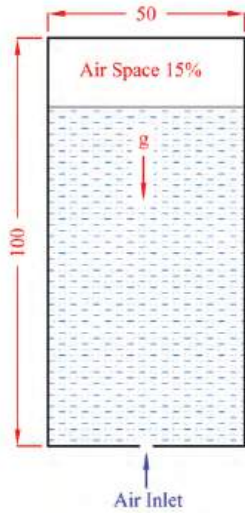


Figure 11. 2D Computational domain [From Raj et al. [61], with permission from Elsevier.]

the time step for the simulations was adjusted and reduced to 0.15 ms for the 2D simulations, whereas for the 3D simulations, a time step of 0.1 ms was implemented. When one parameter was changed, the other parameters were kept constant for simplicity purposes as tabulated in Table 1.

For the 3D domain, a cylindrical tank (diameter = 50 mm) of 162 mm height, with liquid filled up to a height of 152 mm was modelled with an orifice diameter of 2 mm. Results for the 2D case, indicated an increase in the bubble diameter and the bubble's detachment time with respect to reducing surface tension values. After a time of $t = 0.44$ s, $\sigma = 0.0728$ N/m showed 5 distinct detached bubbles, whereas for the case of $\sigma = 0.0482$ N/m, 6 bubbles were reported. A similar result was observed for dynamic viscosity, wherein bubble diameter and detachment time increased with an increase in the liquid's viscosity. With increasing liquid density, the rate of bubble detachment was also reported to rise [61,62].

To quantify thermal destratification efficiency in the 3D case, a thermal stratification decay coefficient (T_{sd}) is defined. T_{sd} is given by Eq. (2):

$$T_{sd} = \frac{|T_{fin} - T_{ini}|}{t_{ho}} \quad (2)$$

T_{fin} represents the resulting temperatures of the liquid layers post bubbling of gases, whereas T_{ini} represents the initial temperature of the liquid layers and t_{ho} is the time required in seconds to achieve a homogeneous temperature within the liquid. Consequently, as anticipated, the value of T_{sd} and T_{sdavg} for the inlet gas velocity of 0.8 m/s was the highest, and the time required to achieve a uniform temperature distribution in the liquid was 24.95 s. This is illustrated in Fig. 12 below. T_{sdavg} is defined as the spatially averaged value of T_{sd} at every instant for every case [61].

Similarly, Fig.13 depicts the gradual development of destratification within the bulk liquid with progressing time for the single inlet case and an inlet gas velocity of 0.6 m/s. A time of 27.73 s is required to attain a uniform temperature distribution in the liquid when an inlet gas velocity of 0.6 m/s is provided [61].

This investigation conducted by Raj et al. [61] on the effects of various parameters on the rate of thermal

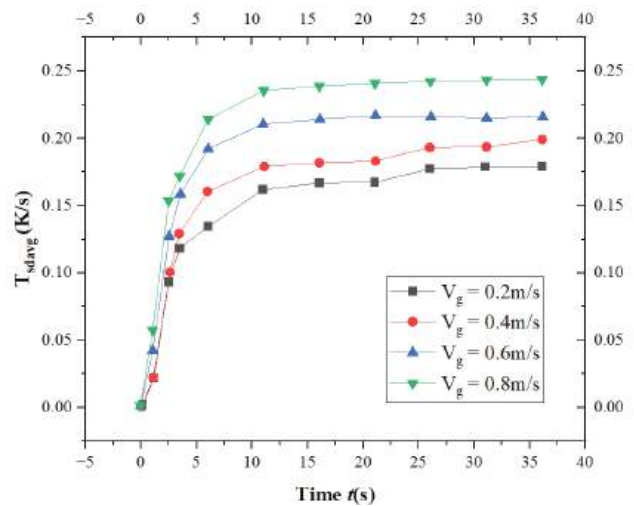


Figure 12. Variation of T_{sdavg} with time for different gas inlet velocities [From Raj et al. [61], with permission from Elsevier.].

Table 1. Variation of parameters for the 2D and 3D case [From Raj et al. [61], with permission from Elsevier.]

Parameter	Value 1	Value 2	Value 3	Value 4
2-Dimensional case				
Surface tension (N/m)	0.0728	0.0693	0.0538	0.0482
Bulk liquid density (kg/m ³)	998.2	1500	2000	2500
Viscosity (Pa-s)	3.494×10^{-6}	7.203×10^{-6}	1.1698×10^{-5}	1.501×10^{-5}
3-Dimensional case				
Inlet gas velocity V_g (m/s)	0.2	0.4	0.6	0.8

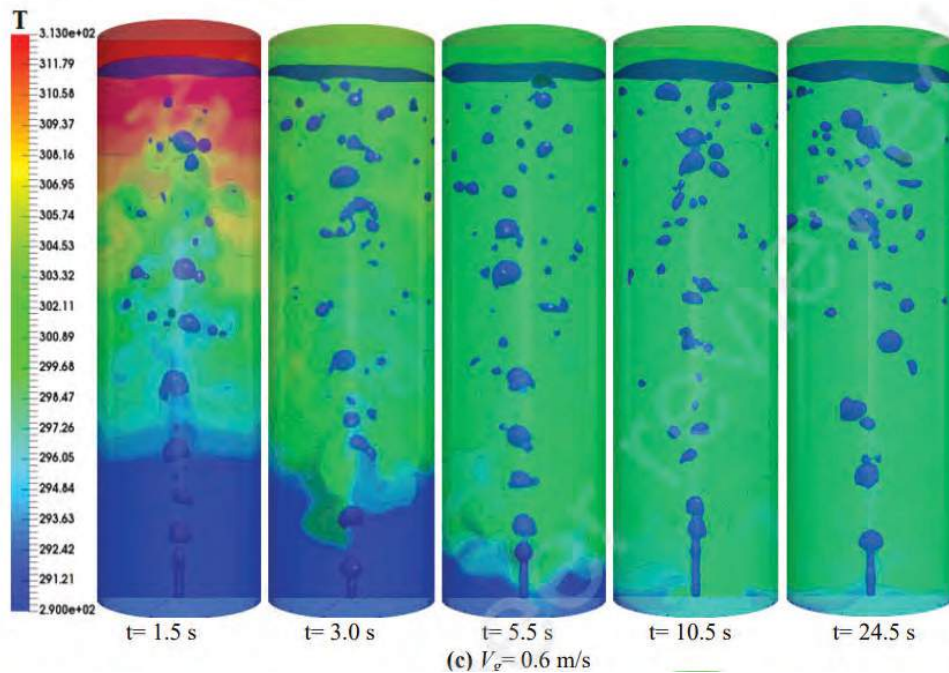


Figure 13. Thermal destratification as a result of gas bubbling for $v=0.6\text{m/s}$ [From Raj et al. [61], with permission from Elsevier.].

destratification serves as an intermediate step in designing an efficient cryogenic tank with minimum stratification.

EXPERIMENTAL STUDIES

Methods of Reducing Thermal Stratification

A number of techniques effective in mitigating stratification have been discussed in detail in the following sections. These techniques either involve modifying the tank's geometry or introducing external elements, such as a mixer or bubbling inert gasses through the bulk liquid. The scope of the following sections, involves delving into promising experimental techniques that have been employed in the recent past to deter stratification. The methods have been explained in detail below, as follows:

Surface Roughness Elements

Surface roughness of the tank's inner walls is a key element in determining the amount of stratification that might occur within the liquid. Micro-surface irregularities have little to no effect on the rate of stratification, although more prominent features like ribs and isogrid walls have a significant effect on stratification. Ribs on the inner walls of the tank are an effective way of reducing stratification, as it enhances the mixing of the fluid layers within the tank.

Fig. 14 shows a cross section of a cylindrical tank of height 'H' and diameter 'D'. Roughness elements are introduced on the cylindrical wall of the tank in the shape of ribs having rectangular cross-section of height 'h' and spaced apart by distance 'p' [20]. Stratification mainly occurs in

three stages, which are as follows; Stage 1: Formation of a boundary layer along the wall, leading to lateral intervention of the warmer liquid adjacent to the upper region. Stage 2: Series of intricate interactions occur amid the thermal boundary layer and the bulk. Stage 3: Final stage comprises the development of a stratified column of the liquid

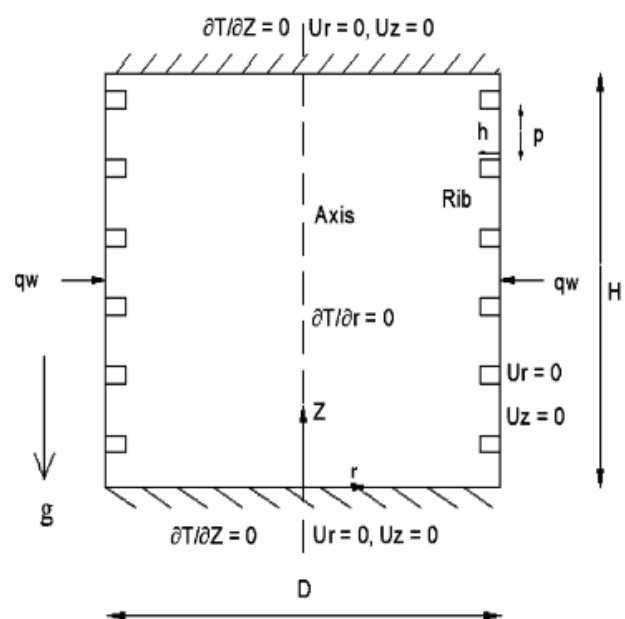


Figure 14. Physical illustration of a ribbed cylinder-shaped tank [From Khurana et al. [20], with permission from Elsevier.]

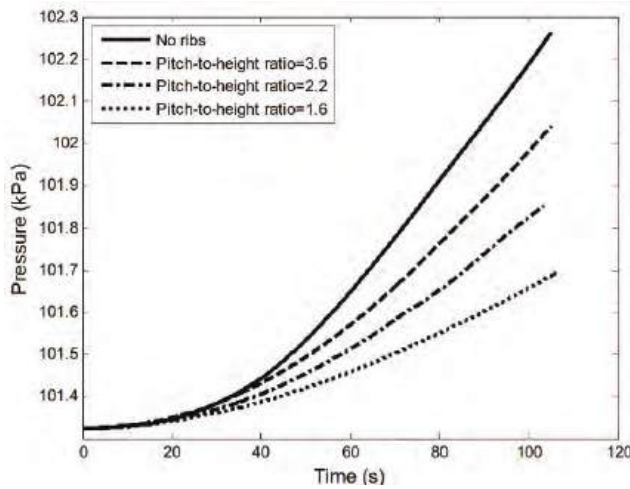


Figure 15. A comparison of pressure rise with time for tank with different rib configurations [From Fu et al. [63], with permission from Elsevier.]

inside the tank, resulting in a temperature gradient along the height of the tank. It is observed that for tanks having ribbed surfaces as shown in Fig. 14, the time essential for the formation of the thermal boundary layer [Stage 1] is higher as compared to tanks with smooth walls. The time required further increases with increase in the ratio of the height of the rib to the spacing amongst two ribs (h/p) as shown in Fig. 15. Although it does not have a major effect on the degree of stratification [20].

Fig.15 above shows the evolution of pressure with time for a tank with ribs with 3 different pitch-to-height ratios of 3.6, 2.2 & 1.6, compared to a tank with a smooth inner wall [63]. It is evident that the rate of pressure increase is minimum for a pitch-to-height ratio of 1.6, while it is the highest for a tank with no ribs. Hence, as discussed in the previous section (Section 2.3), a lower rate of pressure rise is preferred for lesser stratification. Fig.16 below shows the nomenclature used to define different surfaces of the ribs along with their respective wall heat transfer coefficients.

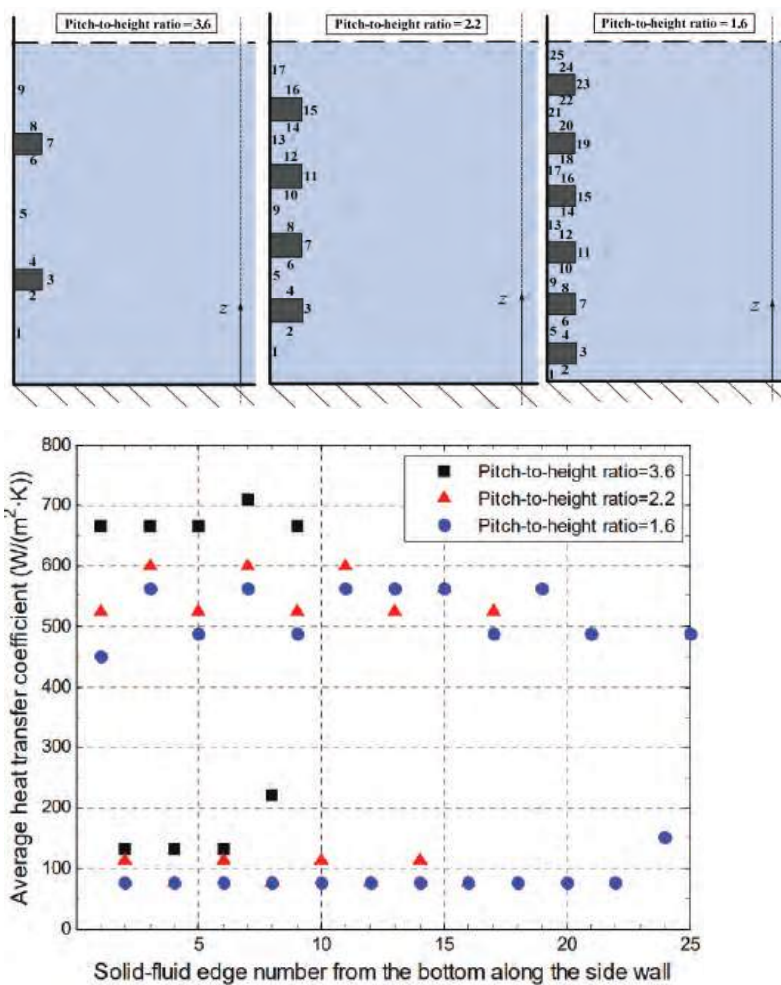


Figure 16. A comparison of the heat transfer co-efficients of different surfaces of the rib and tank wall [From Fu et al. [63], with permission from Elsevier.]

As the number of ribs are increased, so does the number of solid-liquid interfaces within the tank. For pitch-to-height ratios of 1.6, 2.2 and 3.6 there are 25, 17 and 9 surfaces, respectively, as shown in Fig. 16. It is evident that the upper and lower surfaces of each rib have a lower heat transfer coefficient, typically observed to be in the range of 50 to 160 W/m²K as shown in Fig. 16. This is due to the fact that the ribs serve as an obstruction to flow velocity, leading to the thickening of the boundary layer around its surfaces [63].

Overall, the presence of ribs on the internal wall drastically reduces the degree of stratification by almost as much as 30%. Such ribs essentially slow down the convection currents formed within the liquid and act as dampers for all stages of stratification [20,63,64]. Consequently, very recently, ribs of different configurations like V-shaped ribs [65,66], micro-ribs [67,68] and cylindrical ribs [69–72] have also been extensively studied, owing to their enhanced heat transfer qualities.

Modifying Geometry of The Tank

The introduction of a circular or parabolical bulkhead above the total height of the cryogenic tank helps disrupt the natural convection flow of the liquid along the periphery of the tank. Although this method is successful in reducing the stratification rate significantly, it is not preferred due to the complexities involved in the design phase. Another promising method involves the introduction of baffles on the inner walls of the tank [57,58,72–79]. Fig.17 illustrates the use of such baffles along the tank walls.

The tank in Fig.17 is cylindrical in shape, with a diameter of 201 mm and a height of 213 mm respectively. It has two annular plane baffles fitted along its internal wall. The effect of such baffles was studied through a two-dimensional

Volume of Fluid (VOF) simulation conducted in reduced gravity environments of $10^{-5}g_0$, $10^{-3}g_0$, $10^{-1}g_0$, respectively. 'd' is defined as the linear distance amid two neighboring baffles, which was varied from 0 to 60 mm, wherein 0 mm signified a single baffle. The two baffles are positioned symmetrically along the center of the tank. Symbols θ and δ are defined as the angles amongst the baffle and the tank wall, as shown in Fig. 17, which were $\theta_1=105^\circ$ and $\theta_2=75^\circ$ for one case and vice versa for the other; and the gap between the baffle and the wall, varied as 5 mm and 10 mm; respectively. d, θ and δ were varied in order to optimise the effect of baffles on suppressing flow velocity and pressure. The effects of different fill levels (30% to 70%) were also investigated and it was reported that for fill levels between 40% and 60% wherein the baffles are fully immersed in the liquid phase, the pressure rising rate in the simulated tank is considerably lower, than in a tank without baffles [44].

As gravity reduces from $10^{-1}g_0$ to $10^{-5}g_0$, the convection current below the baffles is augmented. A very similar phenomenon was observed in the vapor phase of the ullage. Subsequently, the temperature stratification in the fluid deteriorates. This can be attributed to the suppression of hot-spots which cause the local superheating of the liquid with a reduction in gravity [44].

It was observed that to improve the suppression of sloshing and pressurization within the tank, a large distance of 60 mm between the baffles was preferred over 40 mm. Fig.18 illustrates the variation in the pressure rise for baffles with varying distance between adjoining baffles. 60 mm baffle distance showed a 47% lower pressure rise than for the case of 40 mm as shown in Fig.18 above. Installation of such baffles also leads to disruption of the natural convection flow in the tank [44,71,74].

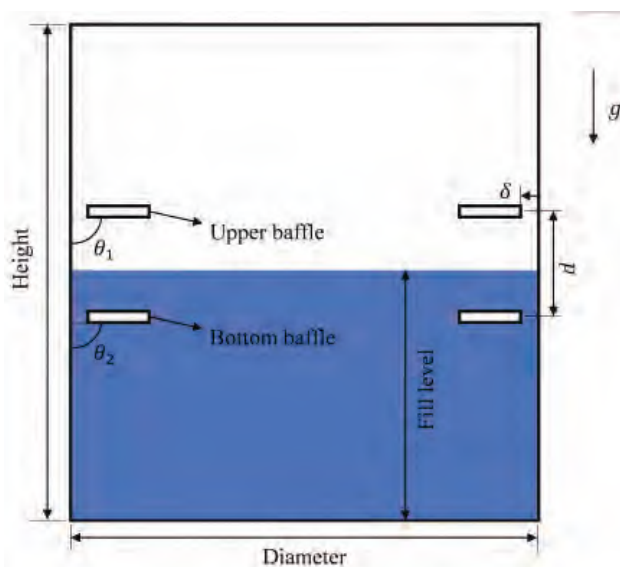


Figure 17. Tank with baffles on the inner walls [From Zuo et al. [44], with permission from Elsevier.]

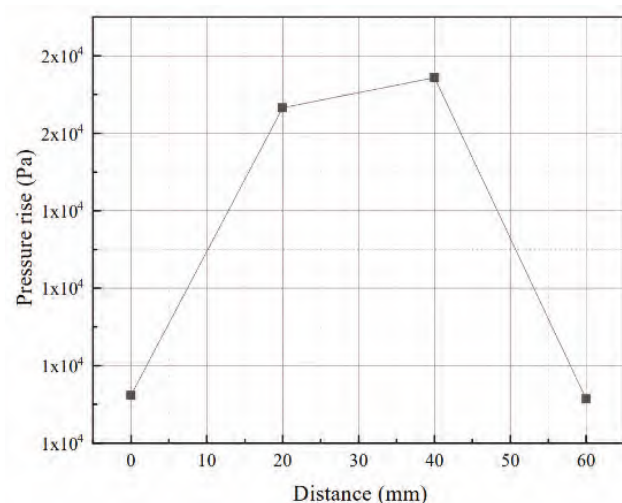


Figure 18. Pressure rise Vs. distance between the adjoining baffles [From Zuo et al. [44], with permission from Elsevier.].

Introduction to cold Helium

This method involves the introduction of cold helium from the bottom up into the tank, through the tor collector. The cold helium in the form of a bubble floats up and blends the heated and cold strata of liquid oxygen (LO_2). This results in an overall reduction in the temperature of the uppermost layer of the propellant or oxidizer [4,75]. This method, even if effective, has a few disadvantages, such as, it aims at reducing the effect caused by heat in-flux and does not deal with the cause of its occurrence. Another disadvantage is that for the storage of the cold helium and for the collector, space and energy are required, which might burden the LV [80–82]. Bubbling of cold Helium (He) is one of the simplest methods to sub-cool the cryogenic fluid in the storage tanks and rocket propulsion applications [28]. To study the effects of bubbling cold helium in cryogenic liquid on the destratification time, a cylindrical tank made from aluminium alloy AA2219, and filled with LN_2 , of thickness 5 mm, diameter 400 mm and a height of 915 mm was used for the study [28]. The tank is filled with LN_2 at a temperature of 88 K, up to a height of 755 mm. The tank is pressurized at a constant pressure of 3.0 bar. Prior to bubbling, the tank is kept still for 1000 s, providing enough time for stratification. Bubbling of the gas is done for a duration of 300 s. The effects of different helium mass flow rates (0.1 g s^{-1} , 0.2 g s^{-1} and 0.4 g s^{-1}) on destratification times were analysed. In this study, destratification time is defined as the time taken to drop the fluid temperatures below the stratification limit temperature of 83 K and further to a steady value. It was observed that a total of 6s were required to destratify the LN_2 with GHe flow rate of 0.1 g s^{-1} . By increasing the bubbling gas flow rate by four times, it resulted in a 66.6% reduction in the destratification time. For the range of GHe mass flow rates considered for this study, it was found that the destratification time decreases as GHe mass flow rate increases [28].

CONCLUSION

Thermal stratification, if not controlled, can have severe effects on the functioning of the LVs. Although it is very essential to determine in every case the amount of stratification that can be accepted, its excess reduction can lead to inefficient use of resources, thereby hindering the overall efficiency of the LV.

A few of the important conclusions drawn are as follows:

1. Any degree of thermal stratification, even if it's in the range of 1-2 °C, deteriorates the functioning of the LV.
2. Rotation of the LV tends to slow down the rate of thermal stratification. For a LH_2 tank at 20% fill level, rotating at $1^\circ/\text{sec}$ in $g/g_0 = 10^{-4}$, under a heat flux of 10 W/m^2 shows a 30 to 60 min reduction in time taken for stratification.
3. An insulation thickness of 40 mm showed the best results, as it had minimum heat in-leak into the storage tank. 40 mm insulation thickness led to a heat in-leak

of only 10.7 W/m^2 , whereas for the 10mm case a heat in-leak of 64.3 W/m^2 was observed.

4. During the pressurisation phase of the tank, there is a subsequent rise in thermal stratification. Consequently, it has also been reported that with an increase in the ullage pressure from 1 to 4 bar, the liquid-gas interface temperature of liquid Nitrogen rises by up to 9.5%.
5. Novel parameters like λ and T_{sd} have been derived in an attempt to quantify the degree of thermal stratification. T_{sdavg} exhibited a value of 0.20 for a $V_g = 0.4 \text{ m/s}$, whereas for 0.8 m/s , T_{sdavg} increased to 0.25, conforming to the observations made, wherein a higher V_g showed better destratification times.
6. Ribs with a pitch-to-height ratio of 1.6 showed promising results among the cases studied. These ribs reduced the degree of stratification by about 30%.
7. Annular baffles along the inner wall of the cryogenic tank also showed promising results. A gap of 60 mm between 2 baffles was deemed optimum and showed a 47% lower pressure rise than that of the 40 mm case.
8. Bubbling of cold Helium gas within the cryogenic tank from its bottom at a rate of 0.1 g s^{-1} took only 6 s of time to bring down the temperature of LN_2 below 83 K. Similarly, simulating bubbling of an inert gas at a velocity of 0.8 m/s from a single orifice, in an 86% filled tank, destratified the tank in 24.95 s.
9. Numerical modelling of an axial jet mixer with a mass flow rate of $3.47 \text{ m}^3/\text{hr}$ yielded encouraging results as it was successful in mitigating thermal stratification under a time of 8 min within liquid oxygen.

The works reviewed in this article are not only applicable for storage tanks of spacecrafts, but also extend to cryogenic storage tanks used in the thermal industry and thus address a wide array of applications. In the near future, research efforts should be directed and focused more towards reproducing real world conditions in experimental set-ups, which are experienced by the cryogenic storage tanks of LV's during its mission. Replicating such circumstances will substantially improve our understanding of thermal stratification. Furthermore, the combined effects of one or more destratification techniques must also be evaluated in the future.

NOMENCLATURE

D	diameter (m)
DC	drain valve
g	gravitational acceleration (m/s^2)
g/g_0	relative gravity
h	axial distance (m)
H	tank of height (m)
K	kelvin
LRE	liquid rocket engine
LV	launch vehicles
T_U	Ullage temperature (°C)
T_B	Bulk liquid temperature (°C)

T_s	Stratified layer temperature (°C)
CG	Centre of Gravity
LH_2	liquid hydrogen
LO_2	liquid oxygen
LN_2	liquid nitrogen
GHe	Gaseous Helium
P	spacing between ribs (m)
q_w	wall heat flux (W/m ²)
R	tank radius (m)
ΔT	temperature difference
EXS	extruded polystyrene
EPS	expanded polystyrene
d	linear distance between adjoining baffles (m)
VOF	Volume Of Fluid
T	temperature (K)
Q_C	jet flow rate
Q_{BL}	boundary layer flow rate
CFL	Courant Frederick-Levy criteria
T_{sd}	thermal stratification decay coefficient (K/s)
T_{sdavg}	average thermal stratification decay coefficient (K/s)
T_{fin}	ultimate temperature of thermally stratified liquid layer (K)
T_{ini}	original temperature of the thermally stratified liquid layer (K)
t_{ho}	time required for attaining homogeneous temperature distribution (s)
V_g	inlet gas velocity (m/s)
Greek symbols	
Ω	angular velocity (°/s)
$\dot{G}$	spin rate (°/s)
θ_1	upper baffle tilt angle (°)
θ_2	lower baffle tilt angle (°)
δ	distance between baffle and tank wall (m)
λ	thermal stratification parameter

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

REFERENCES

- [1] Palerm S, Bonhomme C, Guelou Y, Chopinet JN, Danous P. The future of cryogenic propulsion. *Acta Astronaut* 2015;112:166–173. [\[CrossRef\]](#)
- [2] Fesmire JE, Tomsik TM, Bonner T, Oliveira JM, Conyers HJ, Johnson WL, et al. Integrated heat exchanger design for a cryogenic storage tank. *AIP Conf Proc* 2014;1573:1365–1372. [\[CrossRef\]](#)
- [3] Qiu Y, Yang H, Tong L, Wang L. Research progress of cryogenic materials for storage and transportation of liquid hydrogen. *Metals* 2021;11:1101. [\[CrossRef\]](#)
- [4] Mitikov YO, Ivanenko IS, Pauk OL. New way of eliminating the temperature stratification of liquid oxygen in the tanks of rocket propulsion units. *J Aeronaut Aerospace Eng* 2017;6:4.
- [5] Duan Z, Sun H, Cheng C, Tang W, Xue H. A moving-boundary based dynamic model for predicting the transient free convection and thermal stratification in liquefied gas storage tank. *Int J Therm Sci* 2021;160:106690. [\[CrossRef\]](#)
- [6] Collins Q, Farrington R, Gowie R, Kelly A, Nuzio S, Polus N, et al. Design of thermally efficient cryogenic tanks for spacecraft. In: 2024 Regional Student Conferences 2024;85787. [\[CrossRef\]](#)
- [7] Choi SW, Lee WI, Kim HS. Numerical analysis of convective flow and thermal stratification in a cryogenic storage tank. *Numer Heat Transf A Appl* 2017;71:402–422. [\[CrossRef\]](#)
- [8] Feldman Y, Colonius T, Pauken MT, Hall JL, Jones JA. Simulation and cryogenic experiments of natural convection for the Titan Montgolfiere. *AIAA J* 2012;50:2483–2491. [\[CrossRef\]](#)
- [9] Zuo ZQ, Jiang WB, Huang YH. Effect of baffles on pressurization and thermal stratification in cryogenic tanks under micro-gravity. *Cryogenics* 2018;96:116–124. [\[CrossRef\]](#)
- [10] Liu Z, Li Y, Zhou G. Study on thermal stratification in liquid hydrogen tank under different gravity levels. *Int J Hydrogen Energy* 2018;43:9369–9378. [\[CrossRef\]](#)
- [11] Pietrzyk J, Hepworth H. Analysis of thermal stratification in cryogen storage tanks under quiescent, microgravity conditions. In: 27th Joint Propulsion Conference 1991;2327. [\[CrossRef\]](#)

- [12] Liu Z, Li Y, Jin Y. Pressurization performance and temperature stratification in cryogenic final stage propellant tank. *Appl Therm Eng* 2016;106:211–220. [\[CrossRef\]](#)
- [13] Xavier M, Raj RE, Narayanan V. Thermal stratification in LH2 tank of cryogenic propulsion stage tested in ISRO facility. *IOP Conf Ser Mater Sci Eng* 2017;171:012063. [\[CrossRef\]](#)
- [14] Agrawal G, Joseph J, Agarwal D, Pisharady JC, Kumar SS. Mathematical modelling of thermal stratification in a cryogenic propellant tank. *IOP Conf Ser Mater Sci Eng* 2017;171:01204. [\[CrossRef\]](#)
- [15] Gupta S, Sharma PK, Kumar S, Tiwari CM. Influence of buoyancy forces in MHD non-Newtonian convective nanofluid utilizing Buongiorno's model induced by 3D exponential sheet. *J Therm Eng* 2024;10:1107–1119. [\[CrossRef\]](#)
- [16] Kumar SP, Prasad BVSSS, Venkatarathnam G, Ramamurthi K, Murthy SS. Influence of surface evaporation on stratification in liquid hydrogen tanks of different aspect ratios. *Int J Hydrogen Energy* 2007;32:1954–1960. [\[CrossRef\]](#)
- [17] Robbins JH, Rogers AC Jr. An analysis on predicting thermal stratification in liquid hydrogen. *J Spacecr Rockets* 1966;3:40–45. [\[CrossRef\]](#)
- [18] Bailey TE, Fearn RF. Analytical and experimental determination of liquid-hydrogen temperature stratification. In: *Advances in Cryogenic Engineering* 1964;9:254–264. [\[CrossRef\]](#)
- [19] Fan SC, Chu JC, Scott LE. Thermal stratification in closed cryogenic containers. In: *Advances in Cryogenic Engineering* 1969;14:249–257. [\[CrossRef\]](#)
- [20] Khurana TK, Prasad BVSSS, Ramamurthi K, Murthy SS. Thermal stratification in ribbed liquid hydrogen storage tanks. *Int J Hydrogen Energy* 2006;31:2299–2309. [\[CrossRef\]](#)
- [21] Yilmaz C, Çetin TH, Öztürkmen B, Kanoglu M. Thermodynamic performance analysis of gas liquefaction cycles for cryogenic applications. *J Therm Eng* 2019;5:62–75. [\[CrossRef\]](#)
- [22] Vishnu SB, Kuzhiveli BT. Effect of roughness elements on the evolution of thermal stratification in a cryogenic propellant tank. In: Kazi SN, editor. *Low-Temperature Technologies and Applications*. London: IntechOpen; 2021. [\[CrossRef\]](#)
- [23] Ryali L, Stautner W, Mariappan D. Impact of internal baffle designs on liquid hydrogen sloshing in cryogenic aircraft fuel tanks. *IOP Conf Ser Mater Sci Eng* 2024;1301:012068. [\[CrossRef\]](#)
- [24] Babaelahi M, Babazadeh MA, Saadatfar M. New design for the cold part of heat pipes using functionally graded material in heat sink with variable thickness fins: An analytical approach. *J Therm Eng* 2024;10:1323–1334. [\[CrossRef\]](#)
- [25] Kumar K, Singh S. Investigating thermal stratification in a vertical hot water storage tank under multiple transient operations. *Energy Rep* 2021;7:7186–7199. [\[CrossRef\]](#)
- [26] Poth LJ, Van Hook JR. Control of the thermodynamic state of space-stored cryogenics by jet mixing. *J Spacecr Rockets* 1972;9:332–336. [\[CrossRef\]](#)
- [27] VanOverbeke T. Modeling of pressure reduction due to axial-jet mixers in cryogenic tanks. In: *41st Aerospace Sciences Meeting and Exhibit* 2003;998. [\[CrossRef\]](#)
- [28] Puthettu Muraleedharan S, Joseph J, Chollackal A, Peter J, Agarwal DK. Experimental investigation of thermal stratification in cryogenic tank subjected to multi-species bubbling. *J Therm Anal Calorim* 2023;148:2949–2959. [\[CrossRef\]](#)
- [29] Oliveira J, Kirk DR, Schallhorn PA, Piquero JL, Campbell M, Chase S. An improved model of cryogenic propellant stratification in a rotating, reduced gravity environment. In: *NASA Thermal and Fluids Analysis Workshop (TFAWS) Conference* 2007;1–20.
- [30] Liu Z, Wang L, Jin Y, Li Y. Development of thermal stratification in a rotating cryogenic liquid hydrogen tank. *Int J Hydrogen Energy* 2015;40:15067–15077. [\[CrossRef\]](#)
- [31] Oliveira J, Kirk DR, Schallhorn PA. Analytical model for cryogenic stratification in a rotating and reduced-gravity environment. *J Spacecr Rockets* 2009;46:459–465. [\[CrossRef\]](#)
- [32] Saunders KD, Beardsley RC. An experimental study of the spin-up of a thermally stratified rotating fluid. *Geophys Astrophys Fluid Dyn* 1975;7:1–27. [\[CrossRef\]](#)
- [33] Liu Z, Wang L, Jin Y, Li Y. Development of thermal stratification in a rotating cryogenic liquid hydrogen tank. *Int J Hydrogen Energy* 2015;40:15067–15077. [\[CrossRef\]](#)
- [34] Zhang M, Liu Q, Tao Y, He N, et al. Research on interfacial flow and thermal stratification of cryogenic liquid nitrogen in variable gravity. *Chin J Space Sci* 2024;44:846–862. [\[CrossRef\]](#)
- [35] Onosovskii EV, Stolper LM, Kulik NA, Kostritskii VY. Selection of the optimum type and thickness of insulation for cryogenic tanks. *Chem Pet Eng* 1984;20:309–313. [\[CrossRef\]](#)
- [36] Agrawal G, Joseph J, Agarwal DK, Kumar SS. Effect of insulation thickness on evolution of pressure and temperature in a cryogenic tank. In: *Proc 23rd Natl Heat Mass Transfer Conf and 1st Int ISHMT-ASTFE Heat Mass Transfer Conf*. 2015:17–20.
- [37] Mzad H, Haouam A. Optimization approach of insulation thickness of non-vacuum cryogenic storage tank. *Prog Supercond Cryog* 2020;22:17–23.
- [38] Joseph J, Agrawal G, Agarwal DK, Pisharady JC, Kumar SS. Effect of insulation thickness on pressure evolution and thermal stratification in a cryogenic tank. *Appl Therm Eng* 2017;111:1629–1639. [\[CrossRef\]](#)
- [39] Kürekci NA. Optimum insulation thickness for cold storage walls: Case study for Turkey. *J Therm Eng* 2020;6:873–887. [\[CrossRef\]](#)

- [40] Shahid M, Karimi MN, Mishra AK. Optimum insulation thickness for external building walls for different climate zone in India. *J Therm Eng* 2024;10:1198–1211. [\[CrossRef\]](#)
- [41] Kang M, Kim J, You H, Chang D. Experimental investigation of thermal stratification in cryogenic tanks. *Exp Therm Fluid Sci* 2018;96:371–382. [\[CrossRef\]](#)
- [42] Scheufler H, Gerstmann J. Heat and mass transfer in a cryogenic tank in case of active-pressurization. *Cryogenics* 2022;121:103391. [\[CrossRef\]](#)
- [43] Schmidt AF, Purcell JR, Wilson WA, Smith RV. An experimental study concerning the pressurization and stratification of liquid hydrogen. In: *Advances in Cryogenic Engineering* 1960;5:487–497. [\[CrossRef\]](#)
- [44] Zuo ZQ, Jiang WB, Huang YH. Effect of baffles on pressurization and thermal stratification in cryogenic tanks under micro-gravity. *Cryogenics* 2018;96:116–124. [\[CrossRef\]](#)
- [45] Chin JH, Harper EY, Hines FL, Hurd SE, Vliet GC. Analytical and experimental study of stratification and liquid-ullage coupling. NASA Report No. LMSC-2-05-65-1. 1965.
- [46] Sunena S. Better propulsion system for next generation space travel. *J Aeronaut Aerospace Eng* 2021;10:262.
- [47] Moran ME. Cryogenic fluid storage technology development: Recent and planned efforts at NASA. NASA Report No. E-16816. 2009.
- [48] Hermesen R, Zandbergen B. Pressurization system for a cryogenic propellant tank in a pressure-fed high-altitude rocket. In: *7th European Conference for Aeronautics and Aerospace Sciences* 2017;1–13.
- [49] Kassemi M, Kartuzova O. Effect of interfacial turbulence and accommodation coefficient on CFD predictions of pressurization and pressure control in cryogenic storage tank. *Cryogenics* 2016;74:138–153. [\[CrossRef\]](#)
- [50] Means JD, Ulrich RD. Transient convective heat transfer during and after gas injection into containers. *J Heat Transf* 1975;97:282–287. [\[CrossRef\]](#)
- [51] Raval P, Ramani B. Heat transfer enhancement techniques using different inserts in absorber tube of parabolic trough solar collector: A review. *J Therm Eng* 2024;10:1068–1091. [\[CrossRef\]](#)
- [52] SB V, Bhowmick S, Kuzhiveli BT. Experimental and numerical investigation of stratification and self pressurization in a high pressure liquid nitrogen storage tank. *Energy Sources Part A* 2022;44:2580–2594. [\[CrossRef\]](#)
- [53] Wang J, Webley PA, Hughes TJ. Thermodynamic modelling of low fill levels in cryogenic storage tanks for application to liquid hydrogen maritime transport. *Appl Therm Eng* 2024;256:124054. [\[CrossRef\]](#)
- [54] Fard MA, Barkdoll B. Simple device to improve mixing in storage tanks. In: *World Environmental and Water Resources Congress* 2019;2019:528–535. [\[CrossRef\]](#)
- [55] Dash S, Singh N. Effects of partial heating of top rotating lid with axial temperature gradient on vortex breakdown in case of axisymmetric stratified lid driven swirling flow. *J Therm Eng* 2016;2:883–896. [\[CrossRef\]](#)
- [56] Shad M, Kamran MA. Investigation of the thermal performance of cryogenic regenerator as a porous structure. *J Therm Eng* 2016;2:962–970. [\[CrossRef\]](#)
- [57] Paul A, Nath JM, Das TK. An investigation of the MHD Cu-Al₂O₃/H₂O hybrid-nanofluid in a porous medium across a vertically stretching cylinder incorporating thermal stratification impact. *J Therm Eng* 2023;9:799. [\[CrossRef\]](#)
- [58] Bayareh M. Study of the effect of the porous plates on the tank bottom on the boiling process. *J Therm Eng* 2019;5:149–156. [\[CrossRef\]](#)
- [59] Perona JJ, Hylton TD, Youngblood EL, Cummins RL. Jet mixing of liquids in long horizontal cylindrical tanks. *Ind Eng Chem Res* 1998;37:1478–1482. [\[CrossRef\]](#)
- [60] Brodnick JM, Williams BR, Reske EJ. Validation of a computational fluid dynamics model of axial jet mixing for cryogenic propellant tank pressure control. In: *AIAA SciTech Forum and Exposition* 2024;1–12.
- [61] Raj S, Bibin KS, Roy KER, Prasad B, Jayakumar JS. Analysis of thermal stratification decay in liquid storage tanks using OpenFOAM. 2024;1–81. [\[CrossRef\]](#)
- [62] Lan E, Shi S. Coupled modeling and simulation of tank self-pressurization and thermal stratification. *Int J Heat Mass Transf* 2024;232:125885. [\[CrossRef\]](#)
- [63] Fu J, Sundén B, Chen X. Influence of wall ribs on the thermal stratification and self-pressurization in a cryogenic liquid tank. *Appl Therm Eng* 2014;73:1421–1431. [\[CrossRef\]](#)
- [64] Akinshilo A. Analytical decomposition solutions for heat transfer on straight fins with temperature dependent thermal conductivity and internal heat generation. *J Therm Eng* 2019;5:76–92. [\[CrossRef\]](#)
- [65] Elmouazen H, Zhang X, Gibreel M, Ali M. Heat transfer enhancement of hydrogen rocket engine chamber wall by using V-shape rib. *Int J Hydrogen Energy* 2022;47:9775–9790. [\[CrossRef\]](#)
- [66] Attou Y, Bouhafs M, Feddal A. Numerical analysis of turbulent flow and heat transfer enhancement using V-shaped grooves mounted on the rotary kiln's outer walls. *J Therm Eng* 2023;10:350–359. [\[CrossRef\]](#)
- [67] Li X, Zhang S, Qin J, Bao W. Regulation mechanism of microribs on heat transfer process of cracking reactive flow with strong thermal stratification. *J Heat Transf* 2020;142:114501. [\[CrossRef\]](#)
- [68] Madani K. Numerical investigation of cooling a ribbed microchannel using nanofluid. *J Therm Eng* 2018;4:2408–2422. [\[CrossRef\]](#)

- [69] Gibreel M, Zhang X, Elmouazen H. Investigation of heat transfer enhancement and thermal stratification phenomenon on supercritical hydrogen fuel flow. *Therm Sci Eng Prog* 2024;53:102759. [\[CrossRef\]](#)
- [70] Yildirim C. Theoretical investigation of a solar air heater roughened by ribs and grooves. *J Therm Eng* 2018;4:1702–1712. [\[CrossRef\]](#)
- [71] Noor D, Mirmanto H, Sarsetiyanto J, Soedjono D. Flow behavior and thermal separation mechanism on vortex tube. *J Therm Eng* 2021;7:1090–1099. [\[CrossRef\]](#)
- [72] Ahmadi Nadooshan A, Jahanbakhshi A, Bayareh M. Geometry effects on thermohydraulic behavior of fluid flow in a square enclosure with an inner circular tube. *J Therm Eng* 2019;5:138–148. [\[CrossRef\]](#)
- [73] Grayson G, Lopez A, Chandler F, Hastings L, Tucker S. Cryogenic tank modeling for the Saturn AS-203 experiment. In: 42nd AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit 2006;5258. [\[CrossRef\]](#)
- [74] Tunc G, Wagner H, Bayazitoglu Y. Space shuttle upgrade liquid oxygen tank thermal stratification. In: 35th AIAA Thermophysics Conference 2001;3082. [\[CrossRef\]](#)
- [75] Liu Z, Yang Y, Liu Y, Li Y. Effect of gas injection mass flow rates on the thermal behavior in a cryogenic fuel storage tank. *Int J Hydrogen Energy* 2022;47:14703–14713. [\[CrossRef\]](#)
- [76] Dec JE, Hwang W. Characterizing the development of thermal stratification in an HCCI engine using planar-imaging thermometry. *SAE Int J Engines* 2009;2:421–438. [\[CrossRef\]](#)
- [77] Daigle MJ, Smelyanskiy VN, Boschee J, Foygel M. Temperature stratification in a cryogenic fuel tank. *J Thermophys Heat Transf* 2013;27:116–126. [\[CrossRef\]](#)
- [78] Walls LK, Kirk D, deLuis K, Habermusch MS. Experimental and numerical investigation of reduced gravity fluid slosh dynamics for the characterization of cryogenic launch and space vehicle propellants. In: *Cryogenic Engineering Conference* 2011;136. [\[CrossRef\]](#)
- [79] Fakiri F, Rahmoun K. Unsteady numerical simulation of turbulent forced convection in a rectangular pipe provided with waved porous baffles. *J Therm Eng* 2017;3:1466–1477. [\[CrossRef\]](#)
- [80] Schallhorn P, Campbell D, Chase S, Piquero J, Fortenberry C, Li X, et al. Upper stage tank thermodynamic modeling using SINDA/FLUINT. In: 42nd AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit 2008;5051.
- [81] Bolshinskiy LG, Hedayat A, Hastings LJ, Moder JP, Schnell AR, Sutherlin SG. TankSIM: A cryogenic tank performance prediction program. In: *Space Cryogenics Workshop* 2015;M15-4803.
- [82] Jurns JM, Tomsik TM, Greene WD. Testing of densified liquid hydrogen stratification in a scale model propellant tank. In: 1999 *Cryogenic Engineering and International Cryogenic Materials Conference* 2001;TM-2001-209391.
- [83] Cimbala JM. Fluids in rigid body motion. Available at: https://www.me.psu.edu/cimbala/Learning/Fluid/Rigid_body/rigid_body.htm. Accessed July 16, 2025.