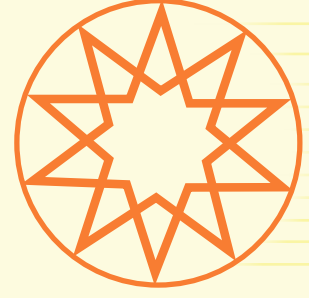


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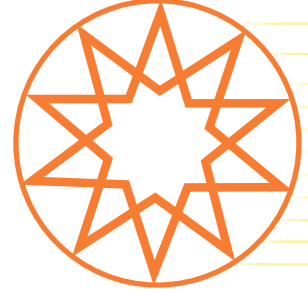
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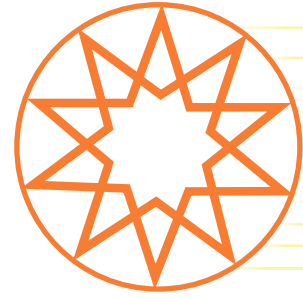
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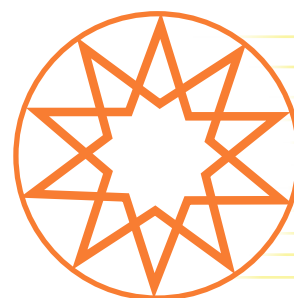
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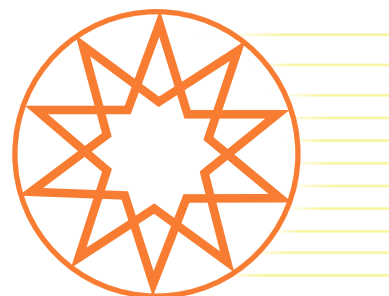
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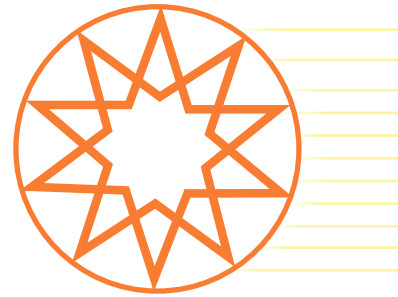


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Research Article

Thermal performance optimization of wickless heat pipes using flash evaporation for distillation

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ABSTRACT

This study presents a numerical and experimental investigation of the effect of flash evaporation on the thermal performance of a wickless heat pipe used in desalination applications. Moreover, this study examines the factors that affect flashing efficiency, such as the feed water mass flow rate, inlet temperature, and cooling water flow rate. High-temperature liquid in the range of 373K–393K becomes superheated when injected through a jet nozzle into a wickless heat pipe. This study focuses on using a jet nozzle type with a small diameter (0.4 mm). The steady value of the mass flow rate of the water is 0.00138 kg/s. Computational Fluid Dynamics (CFD) was used to simulate the two-phase flow and heat transfer processes in a heat pipe, which involves evaporation, condensation, and phase change. This study is unique in that it employs a non-homogeneous multiphase model to capture these processes. The results indicate that increasing the inlet temperature of liquid water leads to a rise in flashing vapor to a certain extent, thereby increasing the condensate flow rate. The maximum condensate flow rate was observed at 388K. The findings suggest that flashing efficiency improves as the condensate flow rate and inlet temperature increase. The optimum flash efficiency was found to be 80% at 393K. A strong agreement was observed between the experimental data, numerical temperature profiles, and the heat pipe's thermal performance.

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INTRODUCTION

Water is an essential resource for human survival, commonly sourced from lakes, reservoirs, and groundwater. Over the past decades, numerous methods and equipment have been developed to produce fresh water

from these sources [1]. However, scientists and experts remained concerned about sewage pollution in rivers and lakes. Numerous devices and methods have been built in recent decades with the intention of creating fresh water [2]. Desalination technologies are used for various

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purposes to extract fresh water from the limitless source of seawater; however, majority of these methods require a significant amount of energy [3]. Relevant and reliable heat dissipation technologies must be developed to advance distillation technologies and address these challenges; distillation devices must be designed to ensure safety, dependability, and system longevity [4,5]. Flash evaporation is one of the recent methods used in desalination [6]. When the surrounding pressure abruptly drops below the saturation pressure corresponding to the liquid temperature, flash evaporation occurs. Accordingly, the liquid becomes superheated. This notion means that the temperature of the liquid is higher than its saturation point, resulting in a highly thermodynamic unstable condition known as liquid superheat. Thereafter, sensible heat is rapidly converted into latent heat as the liquid vaporizes to reach a new thermodynamic equilibrium state [7]. This study utilizes a wickless heat pipe for flash evaporation desalination. Heat pipes have long been recognized as efficient devices for transferring heat from a source to a sink. Moreover, heat pipes have been widely used in various applications, such as solar energy harvesting, waste heat recovery, space vehicles, and in cooling electronic and electrical components [8]. A heat pipe is a sealed, narrow tube with closed ends. After the pipe is evacuated, it is filled with a specific amount of liquid. One end of the pipe is connected to a heat sink, while the other end is exposed to a heat source. The condenser is the sink end, and the evaporator is the source end. These two ends are separated by the intermediate section of the pipe, known as the adiabatic area. The fluid in the heat pipe absorbs heat as it vaporizes in the evaporator and releases it as it condenses in the condenser [9]. The wick-form lining of traditional heat pipes helps return the condensate to the evaporator through the capillary effect. Wickless heat pipes, which rely solely on gravity instead of capillarity, will be introduced later. The orientation is constrained by positioning the condenser above the evaporator. Wickless heat pipes are also referred to as thermosiphons [10]. The thermal performance of heat pipes is primarily determined by their working fluids. Given that water has the highest latent heat and a good boiling temperature, it is widely recommended for various applications; however, other fluids should also be carefully evaluated [11,12].

Several researchers have investigated different designs and applications of wickless heat pipes. Mameli et al. [13] found that the utilization of multiple heaters in a closed-loop thermosyphon enhances its performance. Their experiments with different heater placements demonstrated that asymmetrical arrangements improved fluid circulation and ensured stable operation. The thermosyphon using FC-72 as the working fluid exhibited more efficient heat dissipation than traditional methods, indicating its potential for electronic cooling and thermal management. Alamnar et al. [14] developed a new CFD model to examine how varying tilt angles (10°, 30°, 50°,

70°, and 90°) and fill ratios (25%, 35%, 65%, 80%, and 100%) affect thermosiphon performance. ANSYS Fluent was used to conduct a CFD simulation of two-phase flow inside a thermosiphon heat pipe (THPs) to investigate the temperature distribution and the effect of thermal resistance on thermal performance. The CFD results showed that the evaporator temperature increased for low tilt angles and low fill rates. The lowest heat resistance was observed at a fill ratio of 65% and a tilt angle of 90°. Comparison with experimental results showed remarkable consistency, with maximum variances for thermal resistance and temperature distribution of 8.1% and 4.2%, respectively. Marc et al. [15] numerically and experimentally examined a counter-current two-phase thermosyphon with cascading pools. The pools were utilized to cascade the working fluid to the evaporator end cap. They predicted and validated the experimental outcomes of the novel design using a numerical model and a control volume technique. The results showed that the thermosyphon stages were suitable for the study and for evaluating the effect of design changes. Naresh et al. [16] conducted an experimental study on heat transmission in thermosiphons using six fins with a regular cross-sectional area across the condenser portion. This study utilized three fill ratios (20%, 50%, and 80%) and power levels ranging from 50 W to 275 W, with water and acetone as the charging media. The test results showed that the use of these fins enhanced condensation and improved the thermal performance of the device by 17% in terms of sink and source temperature reduction and 35.48% in terms of thermal conductivities at low heat inputs. Naruka et al. [17] conducted an experimental investigation using three distinct working fluids (water, methanol, and acetone) to estimate the thermal performance of THPs. The experiment used four different tilt angles (0°, 30°, 45°, and 60°) and four input heat levels (10, 15, 20, and 25 W). The results showed that the THP with acetone achieved 31.8% higher efficiency compared with water, with the optimal tilt angle for thermal efficiency being 45°. Heat pipes charged with methanol and acetone efficiently performed in this application, with minimal heat dissipation.

Gorecki et al. [18] in this paper, a finned heat pipe heat exchanger utilized as a recuperator in small air conditioning systems with airflow rates of 300–500 m³/h is modeled, designed, and experimentally validated. R404A was found to be the ideal working fluid at a 20% filling ratio after a thermal model was created utilizing known correlations. A staggered configuration of 20 finned heat pipe rows was suggested by parametric analysis utilizing the model, which produced a steady efficacy of about 60%. While maintaining a pressure drop below 150 Pa, additional optimization through the use of a brute-force technique increased effectiveness to 66%. With a relative error of less than 10%, the experimental data closely matched the predictions of the model.

Alaa et al. [19] studied the thermal performance of a thermosyphon heat pipe (THP) influenced by a novel tube packing (TP). The THP pipe measured 600 mm in length with an inner diameter of 17.4 mm and was made of copper. The experimental results demonstrated that the TP-THP exhibited significantly enhanced heat transfer efficiency, reducing transit time by 31% and thermal resistance by 17%–62%. Samah et al. [20] compared two types of wickless heat pipes: loop type (LT) and straight type (ST). Both systems featured identical evaporator and condenser geometries. The evaporator consisted of a rectangular box with a heating element, and the condenser was a vertical tube cooled by a water jacket. The test was performed with 10 input powers and four fill ratios. The results showed that the LT was more efficient than the ST, exhibiting lower wall temperatures and thermal resistance. The 50% fill ratio consistently resulted in the lowest wall temperatures across all input powers. Hohne [21] conducted a numerical analysis of the thermal performance of a heat pipe using computational fluid dynamics (CFD) to continuously model the two-phase flow of steam and water and heat transfer. The study's uniqueness lies in the use of a homogeneous multiphase model to represent the evaporation, condensation, and phase change processes and the implementation of source terms influenced by the Lee phase change model. In this study, the evaporation and condensation processes inside a heat pipe were validated and compared with earlier research findings from the literature. The evaluation of the heat-pipe's thermal performance and the comparison of the CFD temperature profiles with experimental data indicated a reasonable level of agreement. The heating power should not exceed 1000 W for the type of heat pipe design studied, which was constructed using copper. Ayad and Qusay [22] conducted a numerical education on the thermal performance of a wickless heat pipe (thermosyphon) using 2D CFD with ANSYS Fluent. They studied the supply of temperature and the thermal resistance along the thermosyphon under different working circumstances, considering tilt angle, fill rate, and heat input as key parameters. The results indicated that the influence of filling ratios on the mean wall temperature of the evaporator is greater at high heat inputs than at low heat addition. Ali et al. [23] experimentally examined the thermal performance of heat pipes by observing the influence of the wick and working fluids in a vertical orientation. Water, methanol, ethanol, and various binary combinations with mixing ratios of 50%, 30%, and 70% were used as the working fluids. The findings indicated that the thermal resistance of heat pipes charged with pure and binary mixtures of working fluids reduced with increasing heat inputs. In recent years, several studies have focused on enhancing flash evaporation processes. Wang et al. [24] The performance of a thermosiphon heat pipe with two evaporating surfaces, one corrugated and the other smooth, was studied. The results showed that the corrugated pipe achieved better thermal performance, recording the lowest total thermal resistance of 0.0243

K/W. Numerical simulations (CFD) also showed that the corrugations in the pipe promote fluid boiling within the grooves, improving heat transfer efficiency in the region above the fluid level. Compared to conventional pipes, the corrugated design also offers higher stability and reliability.

The thermal performance of a wickless heat pipe used for electronic cooling with a flat evaporator (100×100×30 mm) is investigated numerically by Samah et al. [25]. EES software was used to assess various fill ratios (15%, 25%, 50%, 85%), input powers (10–100 W), and cooling water flow rates (0.0083–0.033 kg/s). With minimum evaporator and condenser thermal resistances of 0.07°C/W and 0.14°C/W, respectively, the results indicated that the best performance happened at a 50% fill ratio. 0.016 kg/s was the ideal water flow rate.

El-Fiqi et al. [26] conducted an experimental study on the flash vaporization of superheated liquid jets. The study used tap water at low pressures and investigated the effects of superheat degree, inlet temperature, and feed flow rate on flash vaporization. The amount of flashed vapor was determined through condensation measurements and compared with the calculated values. The relationship between superheat degree and flashed vapor was evaluated. The flashing efficiency was measured and compared with previous results. Akram and Radwan [27] developed an experimental setup to investigate the flash evaporation of upward water flow in a 1.8 m vertical pipe. The authors investigated the effects of hydrogen bubble injection and water electrolysis on the flash evaporation process. The findings indicated that both methods increased steam quality and reduced non-equilibrium temperature differences. The experimental results were compared with the theoretical predictions based on a two-phase flow model. Lapka et al. [28] investigated the effect of nozzle shape on flash boiling. The mixture model was used to simulate two-phase flow, and the Zwart–Gerber–Belamri model was utilized for phase change. Simulations were conducted for different nozzle diameters (0.62, 0.72, and 0.82 mm), pressures (5–7 bar), and undercoolings (1–50 K). The results indicated that the nozzle shape had no influence on the mass flow rate of flashing water. Fathinia et al. [29] investigated a low-temperature thermal desalination flash evaporation system. The researchers found that the undefined temperature follows an exponential decay curve, aligning well with the temperature distributions observed in various flash evaporation scenarios. Moreover, increasing the spray flow rate results in high flash evaporation, while decreasing the jet flow rate enhances evaporation. John et al. [30] presented a novel model for flash evaporation, a challenging process due to its high-pressure and high-temperature conditions. The model, developed using Open FOAM, utilizes a dual-diameter approach and a critical nucleation-based mass transfer limit. This model demonstrates excellent agreement with the experimental data for area-averaged pressure and volume fractions. Inter-momentum forces are crucial for accurate momentum transfer prediction in

bubble-dominant flows. Although radial volume fraction predictions show room for improvement, sensitivity studies reveal the model's responsiveness to parameter changes. Cai et al. [31] carried out an experimental investigation to examine the thermal behavior of spray flash evaporation at high pressures and temperatures, emphasizing the impact of superheat level and liquid mass flow rate on vapor production. Similarly, Ji et al. [32] conducted a numerical study on the internal flow properties of a nozzle with S-shaped vanes, demonstrating the effects of geometric changes on the flow distribution and flashing process. Furthermore, a droplet-based model was used in a different work by Cai et al. [33] to describe the dynamics of flash evaporation and evaluate how droplet size and inlet temperature affect flash efficiency. The present study, which uses a wickless heat pipe and a fine-diameter jet nozzle, is based on these recent contributions, which highlight the significance of nozzle shape, fluid conditions, and modeling approaches in comprehending and optimizing flash evaporation.

Despite numerous studies on wickless heat pipe modeling and flash evaporation in distillation, the novelty of this study lies in its integrated numerical and experimental investigation of flash boiling within a wickless heat pipe using a micro-scale jet nozzle. This study aims to utilize a non-homogeneous multiphase model of a wickless heat pipe with flash evaporation induced by a 0.4 mm micro-scale jet nozzle to simulate evaporation, condensation, and phase change phenomena. This setup enables high-intensity evaporation under elevated pressures and temperatures, which has not been comprehensively explored in prior literature. Unlike conventional approaches, this study utilizes a validated 3D non-homogeneous multiphase CFD model to simulate phase-change behavior and compares these results with experimental data. This study highlights how inlet temperature, cooling flow rate, and feed flow rate influence flashing efficiency. This advancement marks a measurable enhancement over prior work and supports the system's potential for scalable, high-performance thermal desalination applications.

NUMERICAL METHODOLOGY

Physical Model

Pressurized water is heated at the supply water tank by an electrical heater, and then the hot water flows through the jet nozzle. After the hot water passes into the jet nozzle, an expansion of the flow occurs as a result of the internal energy being converted into kinetic energy. A sudden change in pressure occurs, and the temperature decreases as a result of flash evaporation in the throat nozzle area, and thus the flow speed increases. The amount of liquid converted to vapor flow passed through the heat pipe; the liquid-vapor mixture reached the condenser section, where the vapor formed was re-condensed to liquid for reuse. Various flow parameters, including outlet nozzle

temperature, saturation temperature and pressure, velocity, thermal resistance, flash efficiency, heat transfer coefficient, and evaporation rate, were measured. Excellent comparisons are provided by the experimental data to validate the recently created methodology. The liquid-vapor mixture in the condensing portion was cooled by cold water, which considerably lowers its temperature when compared to the input. The mass transfer rate downstream of the throat appears to have been impacted by this. In this work, the temperature of water entering the test section is increased in order to increase the flash evaporation in the system.

Computational Model

The thermal performance of a wickless heat pipe with flash evaporation induced by a jet nozzle is investigated through 3D numerical simulations with ANSYS Fluent 22.2. Water is used as the working fluid, circulating within a cooper double-pipe heat pipe. A numerical simulation of the computational domain in (Fig. 1). The two-phase flow is simulated using the volume of fluid (VOF) approach, which accurately tracks the vapor-liquid boundary. The following presumptions are used:

1. The layered annular topology of the flow domain allows for axisymmetric modeling.
2. The flow field is considered transient and three dimensions.
3. Liquid condensate is regarded as the secondary phase, while water vapor is treated as the primary phase.
4. Although thermal energy exchange incorporates flash evaporation effects, the two-phase flow is regarded as incompressible.
5. Non-slip boundary conditions are used at the heat pipe wall.
6. Surface tension and wall roughness are disregarded.
7. Heat conduction governs the copper wall, which is regarded as a solid domain.
8. The inlet boundary is defined as a pressure-inlet and the exit as a pressure-outlet to allow flashing to naturally develop.
9. Symmetry: A symmetry plane is utilized along the model's longitudinal axis, based on the system's geometric and thermal symmetry, to save computational costs.

Governing Equations

The governing equations for the computational domain must be solved to conduct a numerical simulation of the thermal performance of the heat pipe. In this study, the VOF assumption is applied to the fluid and solid domains. The 3D governing fluid flow, heat/mass transfer, momentum, and energy equations of this study are used for the solid and fluid domains, according to previous assumptions. The solid domain (copper pipe) is governed by the energy equation. Meanwhile, the water-vapor flow is governed by continuity, momentum, and energy equations. The examined situation is solved using these equations to determine the transient and steady-state velocities and temperatures [34]:

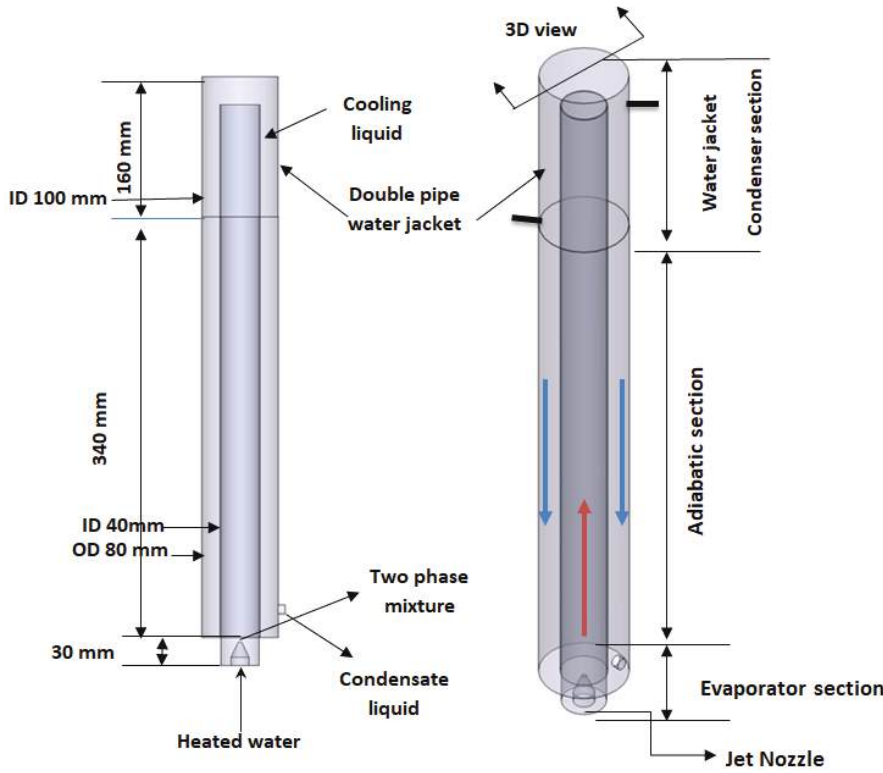


Figure 1. Straight heat pipe (SHP) model geometry and dimensions.

Continuity Equation

$$\nabla \cdot (\alpha_l \rho_l v) = -\frac{\partial}{\partial t} (\alpha_l \rho_l) + S_m \quad (1)$$

$$\rho = \alpha_l \rho_l + (1 - \alpha_l) \rho_v \quad (2)$$

Momentum Equation

$$F_{CSF} = 2\sigma \frac{\alpha_l \rho_l C_v \nabla \alpha_v + \alpha_v \rho_v C_l \nabla \alpha_l}{\rho_l + \rho_v} \quad (3)$$

The following momentum equation is utilized in the VOF model to account for the influence of F_{CSF} forces:

$$\frac{\partial}{\partial t} (\rho \vec{V}) + \nabla \cdot (\rho \vec{V} \vec{V}) = \vec{g} - \nabla p + \nabla \cdot [\mu (\nabla \cdot \vec{V} + (\nabla \cdot \vec{V})^T)] + \vec{F}_{CSF} \quad (4)$$

The dynamic viscosity μ is computed as follows, taking into account a mass average:

$$\mu = \alpha_l \mu_l + (1 - \alpha_l) \mu_v \quad (5)$$

Energy Equation

The energy equation is presented as follows:

$$\frac{\partial}{\partial t} (\rho E) + \nabla \cdot (\vec{V} \rho E) = \nabla \cdot (k \nabla T) + \nabla \cdot (p \vec{V}) + S_E \quad (6)$$

Energy source parameter S_E is added to calculate the heat transfer during the condensation and evaporation processes.

$$E = \frac{\alpha_l \rho_l E_l + \alpha_v \rho_v E_v}{\alpha_l \rho_l + \alpha_v \rho_v} \quad (7)$$

$$E = C_{v,l} (T - T_{sat}) \quad (8)$$

$$E = C_{v,v} (T - T_{sat}) \quad (9)$$

Boundary Conditions

Boundary conditions are defined in the current numerical simulation to accurately depict the flash evaporation and condensation processes inside the wickless heat pipe and closely mimic the experimental configuration.

- Inlet: The gauge pressure and saturation temperature of the incoming liquid water are defined using a pressure-inlet boundary condition. The incoming fluid is assumed to be entirely in the liquid phase (zero vapor volume fraction). The inlet pressure is adjusted between 1 bar and 2.5 bars to examine its influence on flashing behavior. This configuration enables the simulation to accurately depict the beginning of flashing due to pressure loss inside the nozzle instead of imposing a set mass flow rate.

Table 1. The boundary conditions used in the CFD simulation

Name	Description
Constant inlet temperature	373, 378, 383, 388, and 393 K
Constant inlet water cooling system temperature	293, 293.5, 300, 300.5, and 300 K
Constant heat transfer coefficient of the condenser	4937, 4582, 4355, 4076, and 3874 W/m ² .k

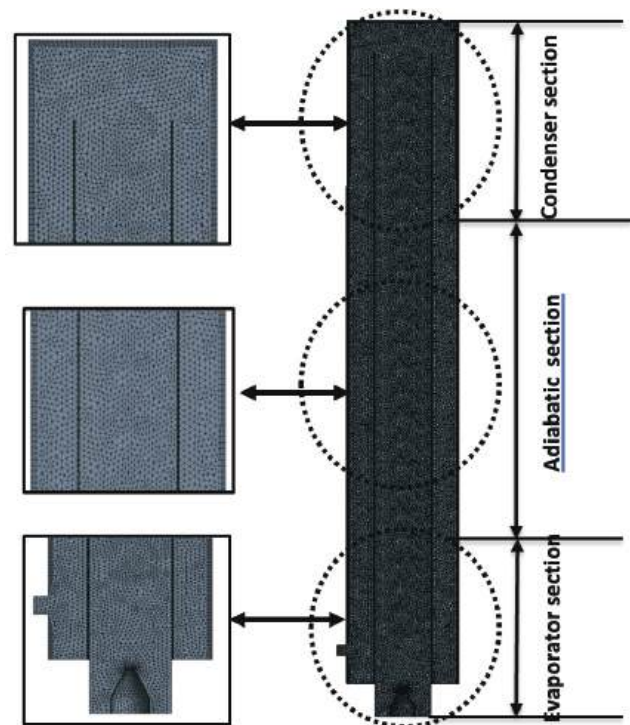
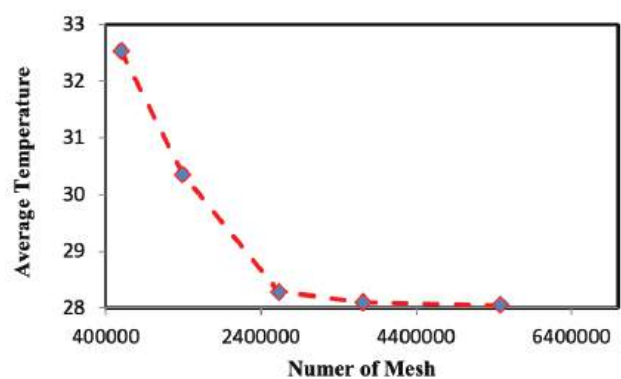
- **Outlet:** A pressure-outlet boundary condition is assigned at the condenser exit, assuming ambient atmospheric pressure. This condition enables free outflow of vapor and liquid without artificially constraining the phase composition.
- **Walls:** Convective heat transfer conditions are imposed on the condenser section walls, with a heat transfer coefficient derived from the experimental cooling water flow data. In all adiabatic sections of the pipe, no heat-flux (adiabatic) boundary conditions are observed.

The chosen boundary conditions reflect realistic experimental parameters, including inlet water temperature, pressure, and average cooling water temperature. A summary of all applied boundary conditions and their values is presented in Table 1.

Mesh Independence

The computational grids for the wickless heat pipe simulations with a jet nozzle are generated using tetrahedral elements via ANSYS Meshing (Fig. 2). This approach facilitates high-quality mesh generation for complex 3D geometries and significantly reduces computational cost. Global and local sizing functions are applied to control mesh density near walls and regions of interest. Mesh quality is evaluated based on orthogonal quality, aspect ratio, and skewness, in accordance with Fluent meshing best practices [34–36]. The orthogonal quality scale ranges from zero to one, with zero denoting a mesh of low quality. A minimum orthogonal quality of ≥ 0.01 must be maintained. The aspect ratio's value is important for wall functions and must be sufficiently low to accurately capture flow behavior near the walls. Lastly, the skewness rate, which is inversely related to the solution accuracy, must be kept as low as possible to minimize numerical error [37]. The orthogonal quality of the meshes used in this study is 0.924, with a minimal value of 0.39. The aspect ratio is 7.596, and the skewness rate is 0.19. These values confirm compliance with the recommended criteria for high mesh fidelity. Inflation layers are implemented along the wall surfaces to accurately capture near-wall behavior in the evaporator and condenser zones. Moreover, the y^+ parameter is monitored throughout the domain and remained below five near all critical heat transfer surfaces, ensuring that the mesh resolution is sufficient to resolve boundary layer effects without the need for wall functions. A mesh independence study is conducted using five mesh densities: 586,600, 1,386,654, 2,619,052, 3,708,894, and 5,488,044 cells. The simulation

results showed convergence beyond 2.6 million elements. The average temperature variation in the cooling water jackets between 2.6 million and 3.7 million grids is less than 0.16, confirming mesh independence (Fig. 3).

**Figure 2.** Mesh of the SHP.**Figure 3.** Mesh independence with a cooling water jacket

Numerical Solution

The numerical simulation is carried out using the ANSYS Fluent 22.1 solver and a pressure-based, transient formulation to represent the unsteady behavior of the two-phase flow and heat transfer process. The governing equations are discretized using second-order upwind schemes for convective terms and central differencing for diffusive terms. The interface between the liquid and vapor phases is tracked using the VOF model with a compressive interface capturing scheme. The simulation used an adaptive time-stepping approach, and the results are monitored until a quasi-steady state is reached. The convergence criteria are set to 10^{-3} for the continuity, momentum, and volume fraction equations and 10^{-6} for the energy equation. The simulations are continued until $t = 90$ s, by which point all residuals had met their convergence thresholds for all cases considered.

EXPERIMENTAL SETUP

The experimental apparatus used in this study is illustrated in Figure 4. This apparatus consists of five main components: the test section (a wickless heat pipe with an integrated jet nozzle), a supply water tank, a condensate collection tank, connecting pipes, and measurement

instruments. All fittings and joints are thoroughly sealed and tested to ensure leak-free operation and accurate data acquisition. The test section consists of a double-pipe made of acrylic (thermal conductivity $k = 0.18 \text{ W/m}\cdot\text{K}$) [38], with the evaporator positioned at the lower section and the condenser at the upper end to facilitate gravitational return of the condensate. Flash evaporation occurs at the nozzle outlet due to a sudden pressure drop. The generated vapor flows through an acrylic pipe (inner diameter: 36 mm, outer diameter: 40 mm, and length: 500 mm) to the condenser, where it is cooled by a surrounding water jacket (length: 200 mm, inner diameter: 96 mm, and outer diameter: 100 mm). Thereafter, the condensate returns to the collection tank via a return pipe (ID: 76 mm, OD: 80 mm, and length: 500 mm) located below the second pipe. All components are insulated with a glass wool to minimize heat losses. The jet nozzle, positioned at the entrance of the evaporator, is made of aluminum and has a 0.4 mm inlet diameter, 12.5 mm outlet diameter, and 30 mm length (Fig. 5). The chosen 0.4 mm nozzle size ensures controlled flash evaporation and is selected to match the operating pressure range of the system. The working temperature range (373–393 K) is selected to replicate low-grade heat conditions typically available in practical desalination setups. The supply tank (300 mm diameter, 350 mm height, and 3 mm



Figure 4. Complete experimental testing device: (1) Heat pipe (2) Nozzle (3) Condenser (4) Water jacket (5) Connecting pipe (6) Water tank (7) Control panel (8) Electrical heater (9) Digital pressure valve (10) Pressure valve (11) Pressure safety valve (12) Control valve (13) Data logger (14) Charging storage (15) Clamp meter (16) Condensate storage.

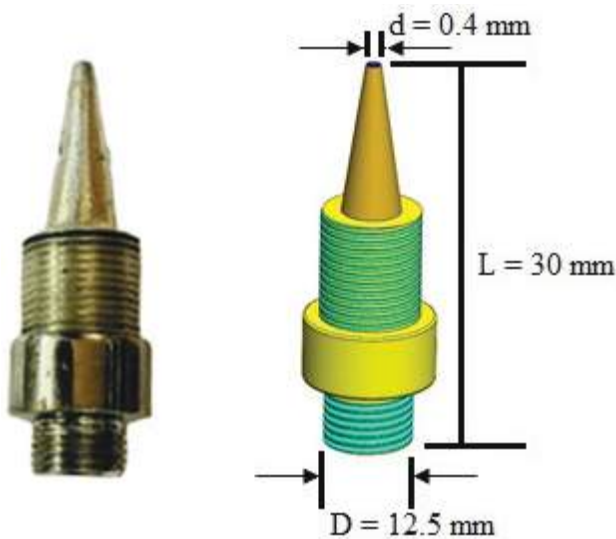


Figure 5. Jet nozzle diagram.

wall thickness) is fabricated from 6061 aluminum alloy. The tank can hold up to 14 kg of water and safely operate at pressures up to 10 bar. Moreover, the tank is equipped with a 4 kW electrical heater (adjustable between 0 W and 4000 W), powered by a 220 V AC source. Pressure regulation and thermal safety are ensured by a brass pressure relief valve, a thermostat-controlled contactor, and a pressure switch that disconnects the power supply when the pressure exceeds a predefined limit. A 220 V AC power source

supplies electricity to the heater. The heater's power supply can be adjusted between 0 W and 4000 W. The electrical heater is made of aluminum alloy, measures 200 mm long, and has a 4 kW capacity. The heater is securely attached to the left side of the water tank and powered by an input supply. Three control valves (6.25 mm diameter) manage fluid flow: one for filling distilled water, one for air discharge, and one for controlling flow to the test section. An additional valve is placed at the tank bottom for drainage. The water flow from the tank to the test section is directed through a copper pipe (12.5 mm diameter and 500 mm length) equipped with a flow-regulating valve. Pressure measurements are conducted using an analog pressure gauge and a digital pressure controller. The gauge, mounted on the tank, provides reference system pressure. Meanwhile, the digital controller—linked to a contactor—monitors and regulates system pressure by activating a cutoff mechanism in case of excessive pressure. The average value from both readings is used in the data analysis.

A temperature thermostat is used to measure the temperature of the fluid in the tank and is connected with a contactor to control the heater. If the temperature exceeds a predetermined limit, then the contactor is activated to shut off the heater, preventing overheating. A clamp meter is used to measure current and voltage. Thereafter, the power is computed by multiplying the voltage by the current. K-type thermocouples that had been calibrated (absolute inaccuracy of $\pm 1^\circ\text{C}$) are used for all temperature readings. Eight data points for temperature measurements

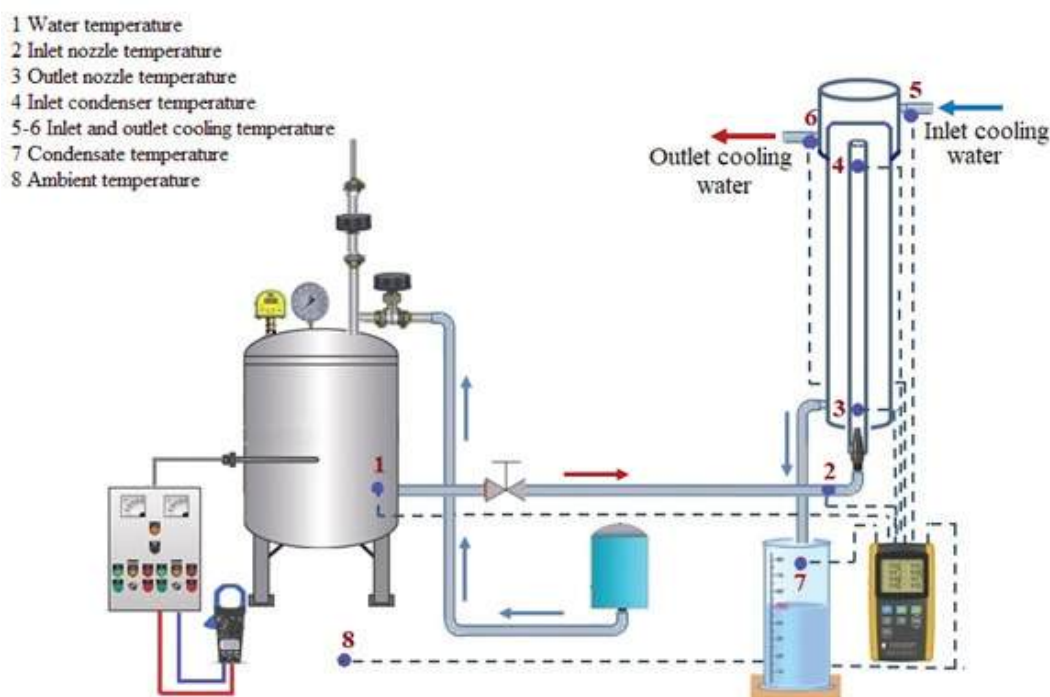


Figure 6. Schematic of the experimental test.

are carefully selected (Fig. 6). Thermocouples (no. 1) are connected to determine the liquid's temperature in the water tank. The temperature of the liquid inside the evaporator with the nozzle is measured by the thermocouple (no. 2). The evaporator output temperature is measured by one K-type (no. 3) thermocouple. The temperature of the vapor inside the condenser is measured by the thermocouple (no. 4). The fiberglass-insulated wires of the thermocouples are visible only at the junction area. The temperature of the cooling water jacket's input and output is measured by two thermocouples (nos. 5–6). Meanwhile, the condensate temperature is measured by the thermocouple (no. 7). The ambient temperature is measured using the thermocouple (no. 8). The heat pipe's temperature is tracked and recorded using a data logger. The data logger type (BTM-4208SD), which contains 12 channels, is compatible with various types of thermocouples (K, J, T, E, R, and S). The thermocouples are connected to the data logger through a special module. The system's internal pressure is examined using two pressure devices: a pressure gauge and a digital pressure control. The pressure gauge is placed on the boiler and is used to measure the system's initial pressure and verify the digital pressure readings. The second device is a digital pressure control that uses a digital pump switch to measure and automatically control system pressure, ensuring that the switch is disconnected via an electrical contractor when the pressure exceeds the specified limit. The average value of two readings is used.

TESTING PROCEDURE

The experimental procedure is initiated by filling the water tank with distilled water via a liquid-level column located on the upper right side of the tank. The corresponding valve is opened to allow water to enter the evacuated tank, and the process is repeated as necessary until the desired water level is reached, as indicated on the level gauge. Once the tank is filled, all valves are closed, and the system is inspected to ensure that no pressure fluctuations or air leakage occurred. After the charging process, the heater is activated by switching on the control panel and engaging the voltage stabilizer. The heating power is gradually increased to the required level using a manual control system. The internal water temperature is regulated using

a thermostat connected to a contactor, which disconnects the power supply once the set point is reached. Once the water reaches the target temperature (within the range of 373–393 K), the control valve between the water tank and the test section is opened, allowing hot, pressurized water to flow into the heat pipe and pass through the jet nozzle. Flash evaporation occurred due to the sudden pressure drop at the nozzle, generating steam that is directed toward the condenser. Furthermore, the cooling water is circulated through the condenser jacket, and the inlet temperature of the cooling water is recorded. As the steam condensed within the condenser, the resulting liquid is collected in a calibrated condensate tank for mass flow rate measurement. Data acquisition is initiated at this point, with temperature readings monitored through the connected data logger. The system is allowed to run for 15–30 min until a steady-state condition is achieved, as indicated by stable temperature and pressure readings. Once steady-state is established, experimental measurements are continuously recorded for three hours. All temperature sensors are calibrated K-type thermocouples ($\pm 1^\circ\text{C}$ accuracy) to ensure data accuracy, while the power input is measured with a clamp meter. Uncertainty in temperature and pressure readings is considered in the overall data analysis to enhance the reliability of the results.

UNCERTAINTY ANALYSIS

Uncertainty analysis was performed for all important measurements and derived values to guarantee the accuracy of the experimental and numerical results. The estimated uncertainties of the instruments used in the experiment are summarized in the table 2. The uncertainty in derived values including heat input, heat rejected, thermal resistance, and flash efficiency was calculated using error propagation.

The estimated uncertainties in computed quantities, based on uncertainty propagation, are:

- Heat rejected (Q_{cond}): $\pm 3.5\%$
- Thermal resistance (R_{cond}): $\pm 4.0\%$
- Flash efficiency (η_{flash}): $\pm 4.2\%$
- CFD prediction (numerical): $\pm 2.0\%$

These degrees of uncertainty validate the accuracy of the simulation and measurements and fall within reasonable engineering bounds.

Table 2. The uncertainty of parameters

Measured Quantity	Instrument / Method	Uncertainty
Temperature (water, vapor)	K-type Thermocouple	$\pm 1^\circ\text{C}$
Flow rate (cooling water)	Rotameter	$\pm 1\%$
Condensate mass	Digital Balance	$\pm 0.001\text{ kg}$
System Pressure	Bourdon gauge + Digital Controller	$\pm 0.5\%$
Voltage & Current	Clamp Meter	$\pm 2\%$

EVALUATION PARAMETER

The performance of the flash evaporation system is evaluated based on several key thermal and flow parameters derived from experimental measurements:

- Inlet Temperature (T_{in}) is determined as the average temperature recorded by thermocouples positioned along the central section of the pipe leading to the nozzle.
- Saturation Temperature (T_{sat}) is obtained from standard water saturation tables, corresponding to the measured system pressure.
- Superheat Degree (ΔT) represents the temperature difference between the inlet fluid and its saturation temperature. Superheat is a critical factor in flash evaporation, as it determines the onset of phase change [39]:

$$\Delta T = T_i - T_{sat} \quad (10)$$

- Actual heat (ΔT) is calculated based on the difference between the inlet and the outlet temperatures of the liquid passing through the evaporator [40]:

$$\Delta T = T_i - T_o \quad (11)$$

- The power provided to the system (Q_{evap}) is calculated by multiplying the measured voltage and current supplied to the heater. The heat rejected by the condenser is determined using the following equation [41]:

$$Q_{cond} = \dot{m}_w (c_{pw} T_{w2} - T_{w1}) \quad (12)$$

where $\dot{m}_w$ is the mass flow rate of the cooling water, and T_{w1} , T_{w2} are the inlet and outlet temperatures of the condenser cooling water.

- Condenser thermal resistance (R_{cond}) is the temperature difference across the condenser divided by the heat rejected:

$$R_{cond} = \frac{T_{sat} - T_{wc}}{Q_{cond}} \quad (13)$$

- Liquid mass flow rate ($\dot{m}_l$) refers to the volumetric flow rate of water entering the test section from the supply tank per unit time.
- Condensation mass flow rate ($\dot{m}_{cond}$) represents the rate at which vapor is condensed and collected in the condensate tank. This factor is considered equal to the vapor mass flow rate from the nozzle under steady conditions.
- Flashing Efficiency (η) is the ratio of actual heat transferred during flashing to the maximum possible energy based on the degree of superheat, assumptions of constant C_p and non-equilibrium conditions.
- This factor is calculated using the following expression [42]:

$$\eta = \frac{\Delta T_{actual}}{\Delta T_{sup}} = \frac{T_i - T_o}{T_i - T_{sat}} \quad (14)$$

These evaluation parameters provide insights into the thermal behavior, energy conversion efficiency, and overall performance of the wickless heat pipe system under flash evaporation conditions.

RESULTS AND DISCUSSION

Numerical Results

Water evaporation under flashing conditions is investigated using numerical approaches. A parametric analysis is conducted to evaluate the sensitivity of the system to various inlet conditions. A selection of findings along the heat pipe with a jet nozzle is presented to illustrate the challenges of selecting appropriate intake settings. Measurements of inlet mass flow rate, inlet temperature, inlet pressure, and the inlet and outlet temperature and pressure of the condenser water jacket serve as an input to the CFD simulation. Numerical results are obtained for the velocity, temperature, and pressure profiles. The inlet temperature of water values is determined by five values, which are 373, 378, 383, 388, and 393 K. Figure 7(a-e) presents the numerical results of the temperature distribution along the vertical wickless heat pipe under rapid evaporation conditions. The figure shows the temperature gradually decreases, reaching its lowest value at the nozzle outlet. When the steam expanded at the throating section of the nozzle, the temperature decreases to below the saturation temperature. A decrease in energy absorbed by the droplets during evaporation is seen in the later profile, which displays a more realistic difference. The ambient temperature is reached. Because of the evaporation, the rise is first somewhat abrupt. Because droplets evaporate at a slow rate, there isn't an abrupt reduction in temperature; instead, it happens gradually over time. Figure 8(a-e) show the data simulation of the temperature distribution at evaporator section of straight heat pipe. Note that the figure represent the temperature distribution at different values of the inlet temperature (373K, 378K, 383K, 388K). As inlet temperature increases, the temperature gradient in the section increases, resulting in a wider distribution of high-temperature regions, as shown in the colors in the figure. The figure show that the highest temperature is near the heating area (bottom), and gradually decreases towards the top. As the inlet temperature increases, the heat distribution becomes more uniform in some regions, indicating improved heat transfer through the system. The red-colored regions increase as inlet temperature increases, indicating an increase in the total heat energy transferred through the system.

The velocity distribution in the vertical wickless heat pipe is quantitatively examined to investigate the vapor flow behavior under flash evaporation conditions. Velocity distribution for the operating condition of (373K, 378K, 383K, 388K, and 393K at 0.00138 kg/s) at the longitudinal section of the thermosyphon heat pipe (THP) is plotted in Figures 9 and 10(a-e). The larger number was computed under the

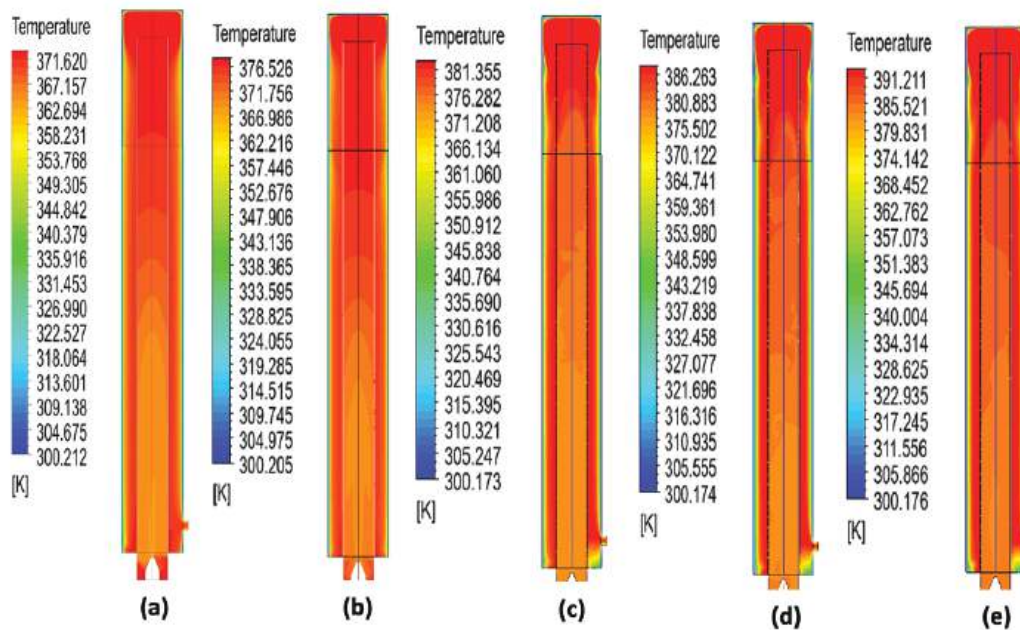


Figure 7. Numerical results of the temperature distribution for (a) 373K (b) 378 K (c) 383 K (d) 388 K (e) 393 K at inlet mass flow rate 0.00138kg/s.

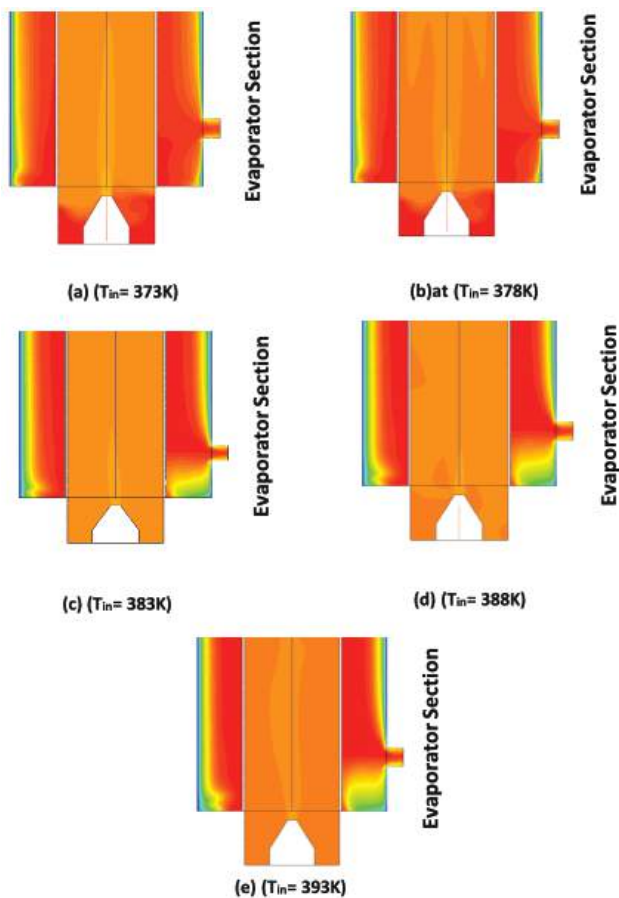


Figure 8. Numerical results of the temperature distribution of evaporator section for the SHP for (a) 373K (b) 378 K (c) 383 K (d) 388 K (e) 393 K at inlet mass flow rate 0.00138kg/s.

assumption that the nozzle exit velocity was determined by the maximal kinetic energy from the liquid cooling at flashing. On the other hand, the lower value was based on the idea that the storage pressure determines the velocity at which the spilled fluid emerges. Both velocity profiles exhibit a similar downward tendency. In this instance, the expected outcomes simply show that the measured value is near the input velocity, which was determined using the liquid's stagnation pressure. This assumption holds true when the volume of liquid inside the jet before flashing is modest compared to the surface area of the jet originating from the leak site. Flashing usually begins at the jet's corners for small leaks. In contrast, the jet's core spreads as a liquid, and this notion makes sense in these circumstances. The temperature profile is also significantly impacted by the beginning or intake velocity value. Therefore, to ensure the correct assumption is made when calculating velocity at the model intake, the inlet velocity can also be used to compare expected temperature profiles.

The two-phase flow patterns inside the SHP are characterized by a consistent behavior, where the steam predominantly flows upward near the inner surface of the evaporator section, while the condensate flows downward along the condenser side. This pattern, supported by three dimension numerical simulations of steam and liquid mass fractions, reflects a typical counter flow structure within the SHP. The steam and condensate streams remain confined near the wall. When these steams meet, the high-momentum stream influences the direction or momentum of the opposing flow. The flow patterns in the longitudinal and transverse sections of the straight heat

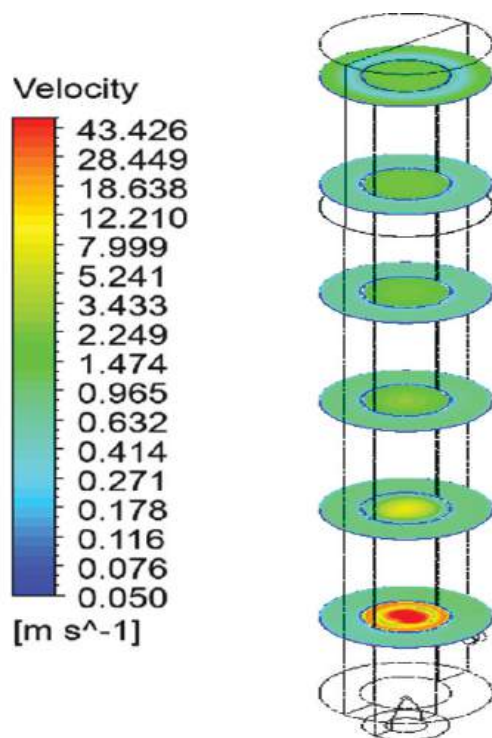


Figure 9. Numerical results of the velocity distribution for five inlet temperature.

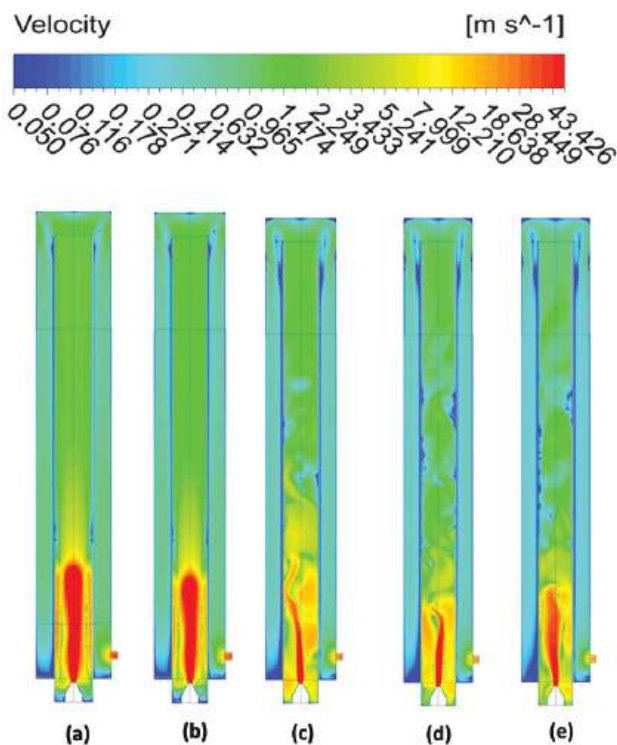


Figure 10. Numerical results of the velocity distribution for (a) 373K (b) 378 K (c) 383 K (d) 388 K (e) 393 K at inlet mass flow rate 0.00138kg/s.

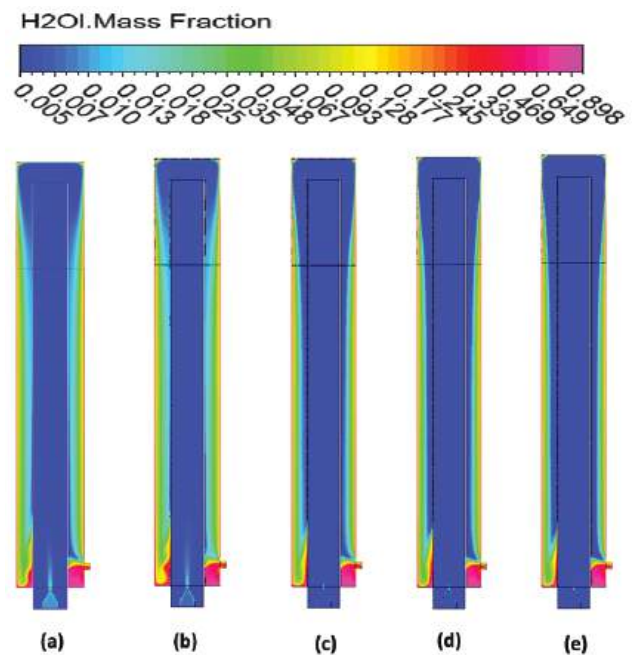


Figure 11. Numerical results of the liquid mass fraction distribution for (a) 373K (b) 378 K (c) 383 K (d) 388 K (e) 393 K at inlet mass flow rate 0.00138kg/s.

pipe are shown in Figure 11(a-e) and Figure 12(a-e) for operating conditions with inlet temperatures of 373K, 378K, 383K, 388K, and 393K, and an inlet liquid mass flow rate of 0.00138 kg/s as a sample. By comparing the mass fractions along the straight heat pipe with those in the jet nozzle, the model can be further evaluated. The generated steam continues to flow upward near the inner surface of the evaporator section of the wickless heat pipe. The condensate continues to flow downward near the inner surface of the condenser section of the straight heat pipe. The figures illustrate the distribution of water vapor concentration inside the tube. The highest vapor concentration is near the inlet area, indicated in red, and it gradually decreases towards the top, as indicated in blue. The blue areas indicate low water vapor concentration, suggesting that most of the vapor has condensed or separated from the mixture, which is consistent with thermal distillation mechanisms or heat exchange inside the thermal heat pipe. According to [1], the re-condensation of vapor is responsible for the mass fraction drop. In this investigation, the highest liquid mass fraction is 0.898.

Comparison Between Numerical and Experimental Data

Effect of the inlet temperature

In this case, the study's numerical results and experimental measurements are compared. Figure 13 presents the comparison between the outlet vapor temperatures from the jet nozzle obtained numerically and experimentally at a mass flow rate of 0.00138 kg/s across five inlet temperature

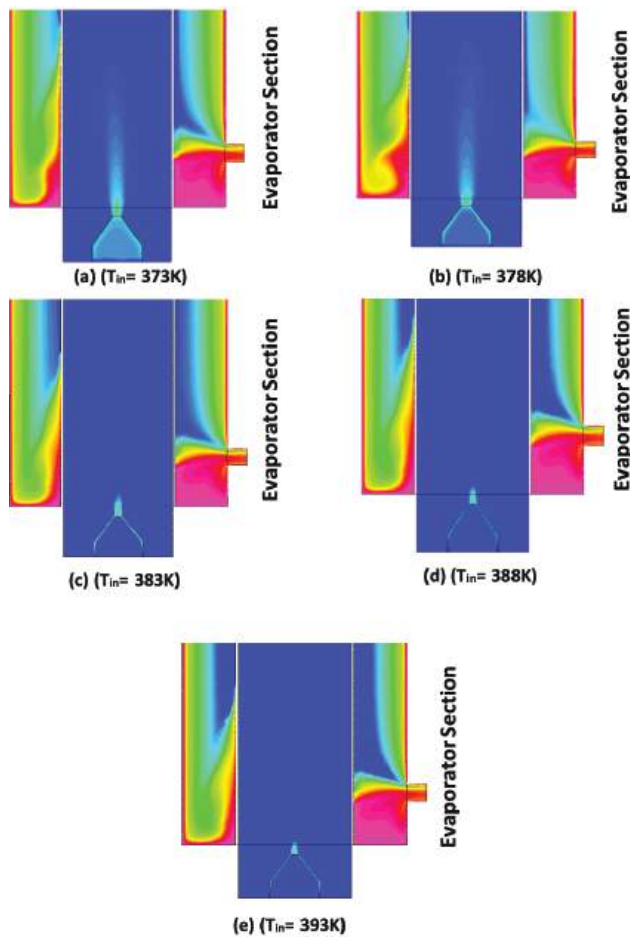


Figure 12. Numerical results of the liquid mass fraction of the evaporator section for operating condition of (inlet tempture. (a)373K, (b)378K, (c)383K, (d)388K, (e) 393K.

values. A strong correlation is observed between both datasets, showing that the outlet temperature increases with rising inlet water temperature. The maximum outlet temperature in the CFD results reaches 392.658 K. Minor discrepancies between the experimental and numerical results are noted, which can be attributed to practical factors such as heat losses to the surroundings, measurement uncertainties, and simplifications in the numerical model like assuming laminar flow and neglecting surface roughness or minor turbulence effects. Despite these, the overall agreement validates the accuracy and reliability of the numerical simulation.

Figure 14 presents a comparison between the experimental and numerical results for the saturation temperature under the same operating conditions as previously described. The figure demonstrates that the saturation temperature increases proportionally with the inlet temperature. This behavior is attributed to an enhanced boiling rate and a simultaneous increase in the wall temperature driven by the evaporation process.

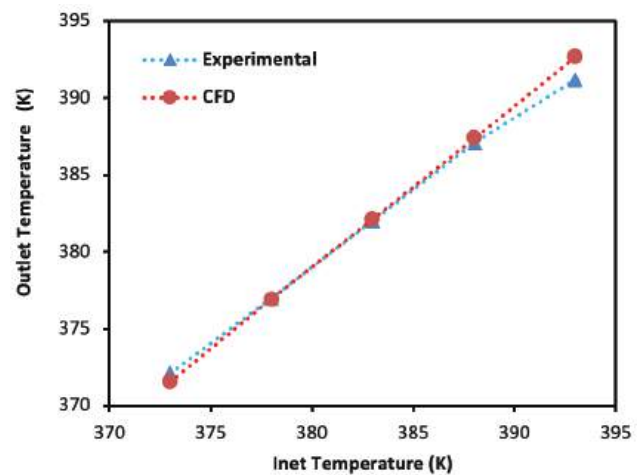


Figure 13. Comparison between experimental and numerical results of the outlet temperature with five value of inlet temperature at 0.00138 kg/s liquid flow rate.

Since the wall temperature typically exceeds the saturation temperature by a certain margin, the heat transfer is sustained, and the saturation temperature correspondingly follows the wall temperature. Any change in the temperature gradient between the wall and the saturated fluid directly influences the boiling rate, thereby restoring thermal equilibrium within the system. The curve further illustrates the influence of flash evaporation on the saturation temperature of water along the test section. The saturation temperature was indirectly determined using the saturation pressure obtained from steam tables. Overall, the figure indicates good agreement between the experimental data and the numerical predictions, affirming the validity of the simulation approach.

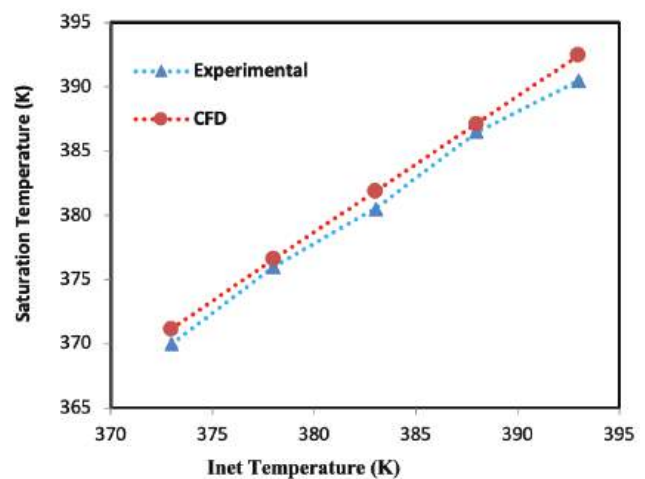


Figure 14. Experimental and numerical results of the measured saturation temperature with five inlet temperature.

Condensate flow rate

The condensate flow rate was determined experimentally by conducting experimental tests. The amount of condensate flow rate determines the efficiency of the system, as the higher amount of condensate flow rate means that the flash evaporation effect is good. Figure 15 displays the experimental and numerical change of the condensate flow rate with inlet temperature for all liquid flow rates. The figure shows the condensate flow rate increases with increased liquid flow rate and increased inlet temperature. For the same amount of electricity, the evaporator's saturation pressure rises as the inlet temperature rises because the vapor space gets smaller. When the produced vapor comes into contact with a small area, it tends to rise in density; consequently, its saturation pressure (a process known as choking). The maximum flow rate of the condensate is 0.00134 kg/sec and 388k.

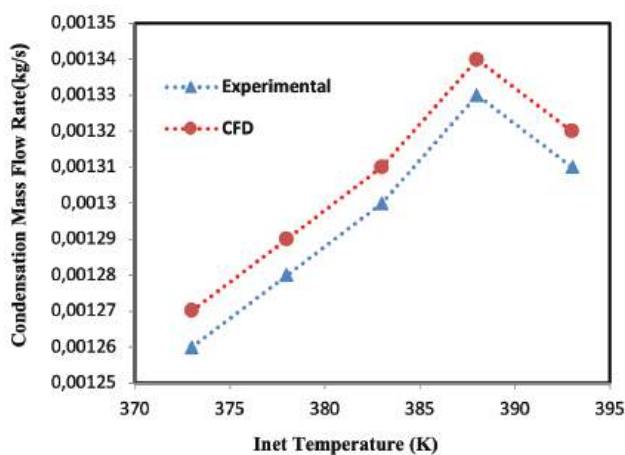


Figure 15. Experimental and numerical results of the measured condensation mass flow rate with the inlet temperature at 0.00138kg/s.

Thermal resistance of the condenser

Thermal resistances of the condenser are found experimentally and numerically by equation (13). And the investigational changes of the observed condenser resistances with the inlet temperature for the inlet mass flow rate are portrayed in Figure 16. These figures manifest that as condenser thermal resistances increased when the evaporator inlet temperature increased. The minimum value of thermal resistance of the condenser is 0.036 (K/W) at 373 K for CFD results. The figure demonstrates that the experimental and numerical results correlate well. The numerical results of the condenser heat transfer coefficient are not available because it is considered a boundary condition for the simulation of the system. Figure 17 validation

between experimental results of the condenser heat transfer coefficient with inlet temperature. The figure shows the condenser coefficient decreases when the evaporator inlet temperature increases, and at an input temperature of 373 K and a liquid flow rate of 0.00138 kg/s, the condenser heat transfer coefficient reaches its maximum of 5059 W/m².K.

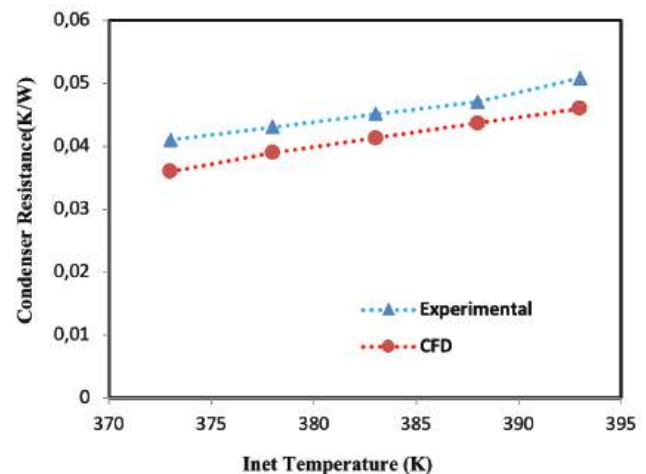


Figure 16. Experimental and numerical results of the measured condenser resistance with the five inlet temperature at 0.00138kg/s of liquid mass flow rate.

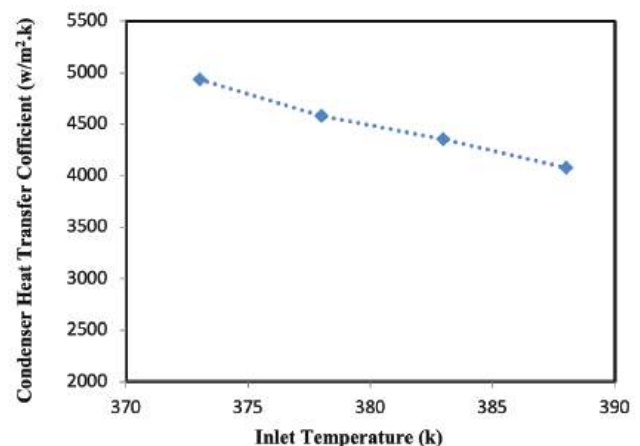


Figure 17. Experimental results of the measured condenser heat transfer coefficient with the inlet temperature at 0.00138kg/s liquid mass flow rate.

Flash efficiency

The flash efficiency of the system was evaluated both experimentally and numerically using Equation (14). Figure 18 illustrates the relationship between flash efficiency and inlet temperature at a fixed inlet liquid mass

flow rate. The figure demonstrates a strong agreement between the experimental and numerical results, indicating the reliability of the simulation model. Flash efficiency is primarily influenced by the amount of vapor generated at the nozzle and is directly linked to the volume of condensate collected. A higher condensation rate reflects a more effective phase change, thereby enhancing the overall flash efficiency. The results reveal that flash efficiency increases with rising inlet temperature, reaching its maximum value of approximately 80% at 393 K. This indicates that higher inlet temperatures improve thermal performance and vapor generation, thus boosting the desalination effectiveness of the system.

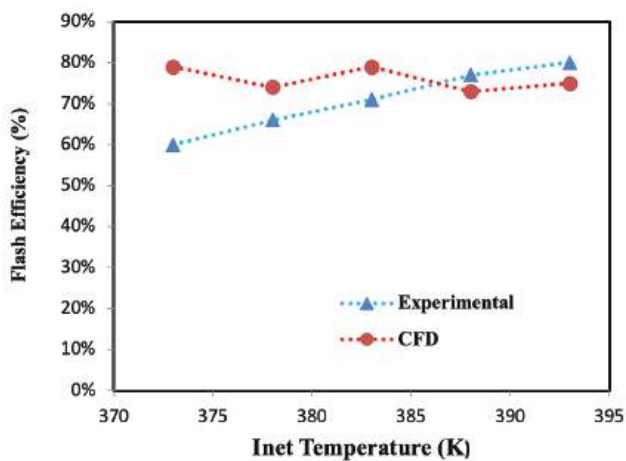


Figure 18. Experimental and numerical results of the measured flash efficiency with the five inlet temperature at 0.00138 kg/s of liquid mass flow rate.

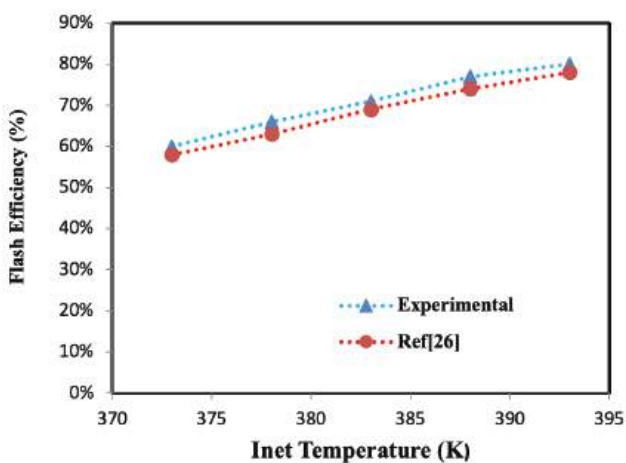


Figure 19. Comparison between the experimental and Ref [26] results of the measured flash efficiency with the five inlet temperature.

Comparison with other work

A comparison with El-Fiqi et al. [26], who used superheated water jets to experimentally study flash evaporation under low-pressure settings, is done in order to validate the current findings. According to both experiments, flashing efficiency increases with superheat and inlet temperature. The current study, however, makes two significant improvements: it employs a 0.4 mm micro-nozzle to more accurately control flashing conditions and a closed wickless heat pipe (thermosyphon) in place of an open jet system. Furthermore, the work combines a non-homogeneous multiphase technique with 3D CFD modeling, enabling a thorough examination of phase transition and heat transport. The reliability and uniqueness of the suggested method for thermal desalination were confirmed by the measured flash efficiency in this work, which reached 80% at 393 K and was in good agreement with El-Fiqi's findings (Fig. 19).

CONCLUSION

A comprehensive experimental and numerical investigation was conducted to evaluate the flash evaporation phenomenon in a wickless heat pipe system using water as the working fluid. The study particularly focused on flash evaporation induced by a superheated water jet injected through a 0.4 mm of diameter nozzle. The numerical simulation was carried out using the 3D ANSYS Fluent 22.2 software to model the evaporation and condensation processes.

Experiments were performed using inlet water temperatures ranging from 373 K to 393 K at a constant feed mass flow rate of 0.00138 kg/s. The analysis incorporated multiple performance indicators, including condensate flow rate, cooling water flow rate, flash efficiency, inlet temperature, and mass flow rate. The following key conclusions can be drawn:

1. Increasing the inlet water temperature leads to enhanced flash vapor generation, confirming the temperature's significant impact on the evaporation process.
2. The condensate flow rate increases with both inlet temperature and liquid flow rate. At higher inlet temperatures, vapor formation intensifies, and when the vapor encounters a smaller volume, its density and saturation pressure increase (a phenomenon known as choking). The maximum condensate flow rate was observed at 388 K.
3. Flash efficiency improves with higher condensate flow rates and elevated inlet temperatures, reaching an optimal value of 80% at 393 K based on CFD results.
4. A strong agreement was observed between the numerical (CFD) simulations and the experimental results for the temperature profiles, validating the model's accuracy.

It is advised to investigate various nozzle geometries and materials in light of the findings in order to improve heat transfer performance and flash efficiency even more. Condensate return and vapor generation may also be enhanced by using multi-nozzle configurations and experimenting with different tilt angles. Future research can also

concentrate on improving the CFD models for turbulent and transient situations and integrating the system with renewable heat sources.

NOMENCLATURE

C_{p_w}	Specific heat of liquid water (J/kg.°C)
C_{v_l}	Specific heat of water (J/kg.°C) Evaporator, m ²
E	Energy (kJ) J/(kg.°C)
h_c	Condenser heat transfer condenser (W/m ² .°C)
g	Gravitational acceleration (m/sec ²)
K	Thermal Conductivity (W/m.°C)
l	liquid
$\dot{m}_w$	Mass flow rate of cooling water (kg/s)
P	Pressure (kpa)
Q_{cond}	Rejected power of the condenser(W)
R_{cond}	Thermal resistance of the condenser (°C/W)
SE	Energy source term (kJ)
ρ	Density (Kg/m ³)
T_i	Inlet temperature of the evaporator (°C)
T_o	Outlet temperature of the evaporator (°C)
T_{w1}	Inlet temperature cooling water (°C)
T_{w2}	Outlet temperature of cooling water (°C)
T_{sat}	Saturation temperature (°C)
t	Time (Sec)
v	Vapor
α	Void Fraction
σ	Surface tension (N/m)
μ	Dynamic viscosity (N sec/m ²)
u	Specific volume (m ³ /kg)
V	Velocity (m/sec)
$\vec{V}$	Velocity vector
SHP	Straight Heat Pipe
η	Flash Efficiency (%)

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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.

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Research Article

Mathematical simulation of magnetohydrodynamic slip transportation phenomena towards non-linear stretching cylinder induced with free stream velocity utilizing buongiorno's model

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ABSTRACT

The manuscript's goal is to examine how magneto-hydrodynamic slip flow affects a non-linear stretching cylinder while producing heat and radiation in presence of free stream velocity. Non-linear behavior produces a flow along with Brownian motion and thermophoresis. The shooting technique model was employed to solve moulded equations numerically following the advent of the Runge Kutta Fehlberg approach in MATLAB programming. The attractive pattern combined thermophoresis with a study of the effects of Brownian motion are also discussed. The effects of important fluid characteristics, such as outer velocity, heat radiation, and velocity of slip, thermophoresis, and Brownian motion are studied and represented via graphs and tables. It is revealed that, heat transfer rate falls down by 71.31% with increment in free stream velocity whereas heat transfer rate rises up by 34.43% with rise in Brownian motion parameter. Moreover, skin friction coefficient intensifies with increment in free stream velocity parameter. Current research has major applications in the production of glass fibers, biotechnological field, encompassing power plants, manufacturing of tetra packs, refrigeration systems, medical science, micro-electro-mechanical Systems and in wide variety of industries.

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INTRODUCTION

In many industrial applications that depends on understanding the mechanisms of different types of energy transfer occurring within the system. The flow towards a stretching sheet was examined by Crane [1]. By taking into consideration a thin metal stretching sheet, the comparison between experimental and theoretical results has been

displayed by Yoon et al. [2]. Sarma and Rao [3] presented the viscoelastic behavior of fluid flow. Manjunatha et al. [4] also investigated the heat source & radiation effects in porous medium. These days, a lot of scientist are concentrating on MHD non-Newtonian nano-fluids because of their many uses in fields like engineering, biological materials, photodynamic therapy, and food processing. Nano-fluid is the general term for a fluid that contains nanoparticles

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(sizes between 1 and 100 nanometers), according to Choi [5]. Additionally, the prospective uses of nanofluids in drug delivery and biomedical imaging have been investigated. To completely comprehend the features and transport capabilities of nanofluids as well as their potential for practical uses, more study is necessary. Before nanofluids are extensively used, concerns regarding the safety and potential effects of nanoparticles on the environment must be thoroughly examined. In the fields of pharmaceuticals, electronics, refrigerators and other household appliances, and power and chemical engineering, nanofluids have unique qualities and characteristics. After that, Convection in nanofluid was described by Buongiorno [6]. Many scientists have studied the transport phenomenon of many Newtonian or non-Newtonian fluid models [7–11].

Magnetohydrodynamics, or MHD for short, is the study of how electrically conducting liquids behaves in presence of magnetism. It analyses how electrically charged particles and magnetic fields interact within a fluid by combining concepts from fluid dynamics and electromagnetism. Studies in the field of magnetohydrodynamics are frequently focused on plasmas, which are states of matter made up of charged particles (ions and electrons) that are capable of electrical conduction. This field includes a wide range of research questions, such as studying plasma dynamics in fusion reactors, studying solar flares and steroidal projectiles, and delving into the mechanisms underlying Earth's magnetosphere. Hayat et al. [12] provided an interpretation of the Powell-Eyring MHD nanofluid flow. The flow conveyance on a non-linear stretched sheet with suction/injection was researched by Siddheshwar and Mahabaleshwar [13]. Researchers have also looked at non-linear influence towards stretched surfaces in a variety of real-world scenarios [14–19]. Recently, impact of MHD nanofluid with existence of slip conditions is revealed by [20]. Akaje et al. [21] studied the transport phenomenon induced by inclined surface in presence of MHD Casson nanofluid. Bolarin et al. [22] discussed the nanofluid boundary layer fluid flow induced by inclined surface. Influence of two phase transport phenomenon in presence of magnetic field induced by resolving system analytically has been investigated by Gholinia et al. [23]. Rana et al. [24] discussed the influence of Ag–MgO (50: 50)/water based hybrid nanofluid utilizing neural network by following FEM simulation. Free convection MHD energy transformation has been studied by Rana et al. [25] in presence of hybrid nanofluids. In recent years, influence of MHD under different boundary conditions has been studied by researchers in [26–31].

Investigations on combined influence of MHD slip flow, outer velocity, heat radiation & generation on non-linear stretching cylinders have not yet shown any results. To fill this gap, impact of free stream velocity with slip conditions and thermal radiation over non-linear surface has been studied out in this research. Mainly the quantitative results gives a new hike to this research to make novel one and has major applications in the biotechnological field,

encompassing power plants, refrigeration systems, medical science, micro-electro-mechanical Systems and in wide variety of industries. The following list indicates this paper's main goal. To establish the mathematical equations that relate to the problem and the equivalent boundary conditions, such as continuity, momentum, energy, and concentration. Using the appropriate non-dimensional parameters and variables, dimensional O.D.E's are converted into dimensionless P.D.E's. Major objectives of the study are:

- To obtain the numerical solutions, MATLAB 2014a will be used in conjunction with the Runge Kutta Fehlberg approach.
- To investigate the effects of incidental features on skin friction, temperature, concentration, velocity, Nusselt and Sherwood number.
- To examine the combined effects of heat radiation and heat generation on temperature and velocity profiles, together with plots and tables.
- To illustrate the contrast between the impact of radiation and thermoelectric power on nanofluids.

MATERIALS AND METHODS

In current study, 2-D incompressible slip flow in MHD nanofluid towards non-linear stretched surface has been induced. Non-linear behavior produces a flow along with Brownian motion and thermophoresis with considered stretching velocity $U_w = cx^n/L$, where c is a constant, n represents non-linear stretching parameter and L represents characteristic length. Two component Buongiorno's [6] model has been followed in current study & Magnetic field is assumed to be applied in radial direction. The physical representation of the current issue is shown in Figure 1. Here, R is the radius of the cylinder. In terms of Cartesian coordinates, the fundamental governing equations for nanofluid are defined as follows [32, 33]:

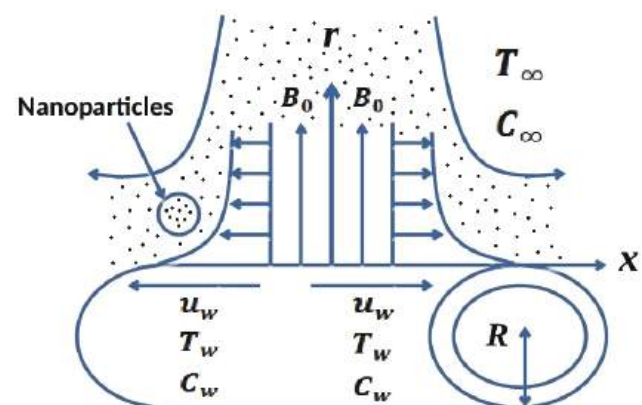


Figure 1. Physical model of the problem.

$$\frac{\partial}{\partial x}(ru) + \frac{\partial}{\partial r}(rv) = 0, \quad (1)$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial r} = v \left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} \right) - \frac{\sigma B_0^2}{\rho} (u - U) + U \frac{\partial U}{\partial x}, \quad (2)$$

$$\begin{aligned} u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial r} = & \tau \left[D_B \frac{\partial C}{\partial r} \frac{\partial T}{\partial r} + \frac{D_T}{T_\infty} \left(\frac{\partial T}{\partial r} \right)^2 \right] \\ & + \alpha \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right) \frac{1}{(\rho c)_f} \frac{1}{r} \frac{\partial}{\partial r} (rq_r) \\ & + \frac{Q_0}{(\rho c)_f} (T - T_\infty) \end{aligned} \quad (3)$$

$$u \frac{\partial C}{\partial x} + v \frac{\partial C}{\partial r} = D_B \left(\frac{\partial^2 C}{\partial r^2} + \frac{1}{r} \frac{\partial C}{\partial r} \right) + \frac{D_T}{T_\infty} \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right), \quad (4)$$

The flow model is described by boundary conditions as follows:

$$\begin{aligned} u = eu_w(x) + N\mu \left(\frac{\partial u}{\partial r} \right), v = 0, T = T_w, C = C_w \text{ at } r = R, \\ u \rightarrow U = \frac{bx}{L}, T \rightarrow T_\infty, C \rightarrow C_\infty \text{ as } r \rightarrow \infty \end{aligned} \quad (5)$$

Where $N\mu$ the velocity is slip factor and e is the stretching velocity parameter.

Using similarity variables

$$\begin{aligned} \zeta = \frac{r^2 - R^2}{2R} \left(\frac{c}{vL} \right)^{\frac{1}{2}}, \quad \psi = \left(\frac{vc}{L} \right)^{\frac{1}{2}} x R f_\delta(\zeta), \\ \theta_\delta(\zeta) = \frac{T - T_\infty}{T_f - T_\infty} \text{ and } \phi_\delta(\zeta) = \frac{C - C_\infty}{C_w - C_\infty} \end{aligned} \quad (6)$$

After substituting (6) into (2)–(4), we obtain

$$(1 + 2\zeta\gamma) f_\delta''' + 2\gamma f_\delta'' + \left(\frac{n+1}{2} \right) f_\delta f_\delta'' - n f_\delta'^2 - M(f_\delta' - \lambda) + n\lambda^2 = 0 \quad (7)$$

$$\begin{aligned} (1 + 2\zeta\gamma) \left(1 + \frac{4}{3} Rd \right) \theta_\delta'' + \left[2\gamma \left(1 + \frac{4}{3} Rd \right) + Pr \left(\frac{n+1}{2} \right) f_\delta \right] \theta_\delta' \\ + Pr \left[Nb(1 + 2\zeta\gamma) \theta_\delta' \phi_\delta' + Nt(1 + 2\zeta\gamma) \theta_\delta'^2 + Q\theta_\delta \right] = 0 \end{aligned} \quad (8)$$

$$(1 + 2\zeta\gamma) \phi_\delta'' + \left(\frac{n+1}{2} \right) Sc f_\delta \phi_\delta' + 2\gamma \phi_\delta' + (1 + 2\zeta\gamma) \frac{Nt}{Nb} \theta_\delta'' + 2\gamma \frac{Nt}{Nb} \theta_\delta' = 0 \quad (9)$$

Transformed (5) which become,

$$\begin{aligned} f_\delta(0) = 0, \quad f_\delta'(0) = e + df_\delta''(0), \quad \theta_\delta(0) = 1, \quad \phi_\delta(0) = 1, \\ f_\delta'(\zeta) \rightarrow \lambda, \quad \theta_\delta(\zeta) \rightarrow 0, \quad \phi_\delta(\zeta) \rightarrow 0 \text{ as } \zeta \rightarrow \infty \end{aligned} \quad (10)$$

Later, the velocity slip parameter d is defined as

$$d = N\mu \frac{r}{R} \left(\frac{c}{vL} \right)^{\frac{1}{2}} = N_0 \left(\frac{c}{vL} \right)^{\frac{1}{2}} \quad (11)$$

where $N = N_0 R / \mu r$, N_0 being the constant wall slip velocity.

Dimensionless parameters; free stream velocity, Prandtl number, Brownian motion, thermophoresis, heat generation, heat radiation and magnetic parameters are symbolically given by

$$\begin{aligned} \lambda = \frac{b}{c}, Pr = \frac{v}{\alpha}, Sc = \frac{v}{D_B}, Nb = \frac{(\rho c)_p D_B (C_w - C_\infty)}{(\rho c)_f v}, \\ Nt = \frac{(\rho c)_p D_T (T_f - T_\infty)}{(\rho c)_f T_\infty v}, Q = \frac{Q_0 L}{c(\rho c)_f}, Rd = \frac{4\sigma^* T_\infty^3}{kk^*}, \\ M = \frac{\sigma B_0^2 L}{\rho c} \end{aligned} \quad (12)$$

Sh_x, Nu_x and C_f are defined as:

$$Sh_x = \frac{xq_m}{D_B(C_w - C_\infty)}, Nu_x = \frac{xq_w}{k(T_f - T_\infty)}, C_f = \frac{\tau_w}{\rho U_\infty^2} \quad (13)$$

where q_m, q_w and τ_w are as follows,

$$\begin{aligned} q_m = -D_B \left(\frac{\partial T}{\partial r} \right)_{r=R}, \quad q_w = - \left(\alpha + \frac{16\sigma^* T_\infty^3}{3(\rho c)_f k^*} \right) \left(\frac{\partial T}{\partial r} \right)_{r=R}, \\ \tau_w = \mu \left(\frac{\partial u}{\partial r} \right)_{r=R} \end{aligned} \quad (14)$$

with k and m are the thermal conductivity and dynamic viscosity of the nanofluid serially. Using the similarity variables eq (6), eq (13) becomes

$$\begin{aligned} Sh_x Re_x^{-1/2} = -\phi_\delta'(0), \quad Nu_x Re_x^{-1/2} = - \left(1 + \frac{4}{3} Rd \right) \theta_\delta'(0), \\ C_f Re_x^{1/2} = f_\delta''(0) \end{aligned} \quad (15)$$

where $Re_x = U_\infty x / \nu$ is the local Reynolds number.

Numerical Solution

Due to their highly non-linear behavior, D.E's (7)–(9) cannot be solved analytically. Our method of choice for solving D.E's (7)–(9) along with B.C's (10) is the numerical procedure known as the RKF approach. Shooting technique has been adopted to solve the system of equation and the process has been displayed via figure 2 using flow chart. The following is a rearrangement of the controlling non-linear ordinary differential equations:

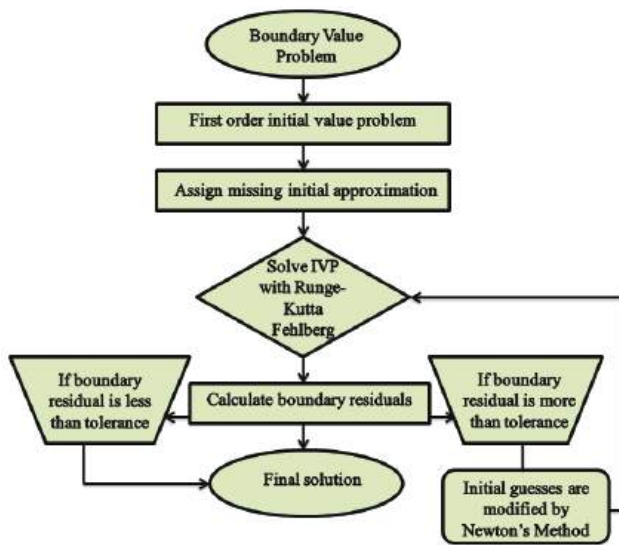


Figure 2. Flow chart of shooting technique.

$$f_{\delta}''' = \frac{-1}{(1+2\zeta\gamma)} \left[2\gamma f_{\delta}'' + \left(\frac{n+1}{2} \right) f_{\delta} f_{\delta}'' - n f_{\delta}'^2 - M(f_{\delta}' - \lambda) + n\lambda^2 \right] \quad (16)$$

$$\theta_{\delta}'' = \frac{-1}{(1+2\zeta\gamma)} \left[\frac{\left\{ 2\gamma \left(1 + \frac{4}{3} Rd \right) + Pr \left(\frac{n+1}{2} \right) f_{\delta} \right\} \theta_{\delta}'}{\left(1 + \frac{4}{3} Rd \right)} + Pr \{ Nb(1+2\zeta\gamma) \theta_{\delta}' \phi_{\delta}' + Nt(1+2\zeta\gamma) \theta_{\delta}'^2 + Q\theta_{\delta}' \} \right] \quad (17)$$

$$\phi_{\delta}'' = \frac{-1}{(1+2\zeta\gamma)} \left[\left(\frac{n+1}{2} \right) Sc f_{\delta} \phi_{\delta}' + 2\gamma \phi_{\delta}' + (1+2\zeta\gamma) \frac{Nt}{Nb} \theta_{\delta}'' + 2\gamma \frac{Nt}{Nb} \theta_{\delta}' \right] \quad (18)$$

Transformed 1st order D.E's are:

$$s_1' = s_2 \quad (19)$$

$$s_2' = s_3 \quad (20)$$

$$s_3' = \frac{-1}{(1+2\zeta\gamma)} \left[2\gamma s_3 + \left(\frac{n+1}{2} \right) s_1 s_3 - n s_2^2 - M(s_2 - \lambda) + n\lambda^2 \right] \quad (21)$$

$$s_4' = s_5 \quad (22)$$

$$s_5' = \frac{-1}{(1+2\zeta\gamma)} \left[\frac{\left\{ 2\gamma \left(1 + \frac{4}{3} Rd \right) + Pr \left(\frac{n+1}{2} \right) s_1 \right\} s_5}{\left(1 + \frac{4}{3} Rd \right)} + Pr \{ Nb(1+2\zeta\gamma) s_5 s_7 + Nt(1+2\zeta\gamma) s_5^2 + Qs_4 \} \right] \quad (23)$$

$$s_6' = s_7 \quad (24)$$

$$s_7' = \frac{-1}{(1+2\zeta\gamma)} \left[\left(\frac{n+1}{2} \right) Sc s_1 s_7 + 2\gamma s_7 + (1+2\zeta\gamma) \frac{Nt}{Nb} s_5' + 2\gamma \frac{Nt}{Nb} s_5 \right] \quad (25)$$

B.C's are reduced to

$$\begin{bmatrix} s_1 \\ s_2 \\ s_3 \\ s_4 \\ s_5 \\ s_6 \\ s_7 \end{bmatrix} = \begin{bmatrix} 0 \\ e + \sigma h_1 \\ h_1 \\ 1 \\ h_2 \\ 1 \\ h_3 \end{bmatrix} \quad (26)$$

Where,

$$\begin{aligned} f_{\delta} &= s_1, \quad f_{\delta}' = s_2, \quad f_{\delta}'' = s_3, \quad f_{\delta}''' = s_4, \\ \theta_{\delta} &= s_5, \quad \theta_{\delta}' = s_6, \quad \theta_{\delta}'' = s_7, \\ \phi_{\delta} &= s_8, \quad \phi_{\delta}' = s_9, \quad \phi_{\delta}'' = s_{10} \end{aligned} \quad (27)$$

Code Validation

For determining the final results in the current study, we employ the RKF approach, and the numerical procedure we are using the shooting technique. At the border, we assume a constant temperature and concentration of nanoparticles. The boundaries' restrictions are as follows:

$$u_w = ax^n; \quad T = T_w \quad \text{and} \quad C = C_w \quad (28)$$

In present problem, we compare our results for $-\theta_{\delta}'(0)$ for different Pr and fixed $n = 1$, $d = 0$, $Nt = Nb = 0$ and as shown in Table 1.

Table 1. Comparison of for Nusselt number $-\theta_{\delta}'(0)$.

Pr	Gorla and Sidawi [34]	Residual Error	Mankinde and Aziz [35]	Residual Error	Present Result
2	0.9114	-0.0001	0.9114	-0.0001	0.9113
7	1.8954	0.0000	1.8954	0.0000	1.8954
20	3.3539	0.0000	-	-	3.3539
70	6.4622	-0.0001	-	-	6.4621

NUMERICAL RESULTS AND DISCUSSION

The current investigation determines the numerical solution for D.E's (7)-(9) when B.C's (10) are obtained using the RKF method. Determining the influence of important fluid factors namely λ , Rd , d , Nt and Nb on $f'_\delta(0)$, $\theta_\delta(0)$ and $\phi_\delta(0)$ is the primary motivation for solving the current problem. Table 2-6 demonstrate the impact of fluid parameters $\lambda = 0.1, 0.3, 0.5, 0.7$, $Nb = 0.1, 0.3, 0.5, 0.7$, $Nt = 0.1, 0.3, 0.5, 0.7$, $Rd = 1, 3, 5, 7$, and $0, 0.2, 0.4, 0.6$ on $f''_\delta(0)$, $-\theta'_\delta(0)$ and $-\phi'_\delta(0)$ in current problem.

Table 2. $f''_\delta(0)$, $-\theta'_\delta(0)$ and $-\phi'_\delta(0)$ against λ .

λ	$f''_\delta(0)$	$-\theta'_\delta(0)$	$-\phi'_\delta(0)$
0.1	-1.01740	0.30784	1.11220
0.3	-0.87538	0.39182	1.12607
0.5	-0.67682	0.46528	1.16666
0.7	-0.43353	0.52737	1.21691

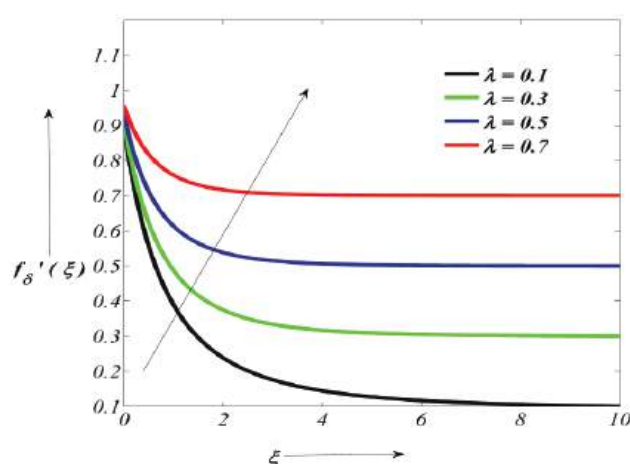


Figure 3. Velocity profile against λ .

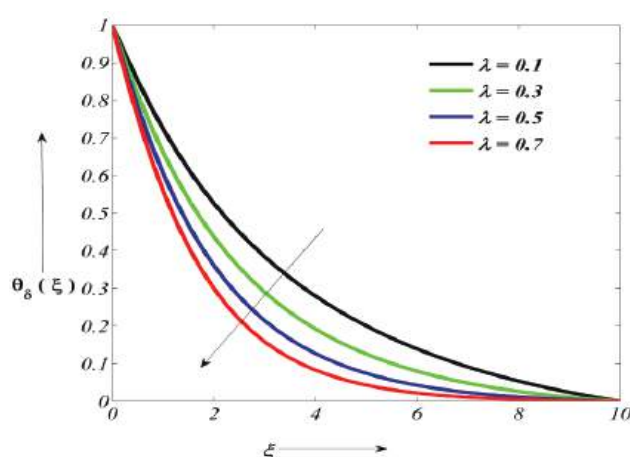


Figure 4. Temperature profile against λ .

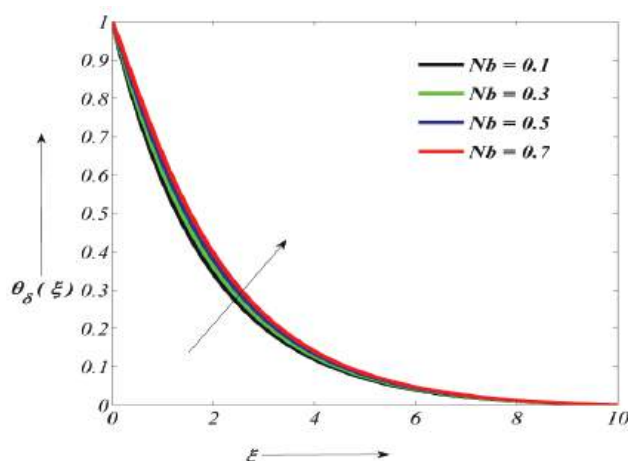


Figure 5. Temperature profile against Nb .

Table 3. $f''_\delta(0)$, $-\theta'_\delta(0)$ and $-\phi'_\delta(0)$ against Nb .

Nb	$f''_\delta(0)$	$-\theta'_\delta(0)$	$-\phi'_\delta(0)$
0.1	-0.67682	0.53271	0.99747
0.3	-0.67682	0.46528	1.16666
0.5	-0.67682	0.40426	1.19699
0.7	-0.67682	0.34930	1.20765

Table 4. $f''_\delta(0)$, $-\theta'_\delta(0)$ and $-\phi'_\delta(0)$ against Nt .

Nt	$f''_\delta(0)$	$-\theta'_\delta(0)$	$-\phi'_\delta(0)$
0.1	-0.67682	0.51061	1.15897
0.3	-0.67682	0.46528	1.16666
0.5	-0.67682	0.42344	1.21242
0.7	-0.67682	0.38480	1.29061

Figures 3, and 4 displayed the impact of λ ($0.1 \leq \lambda \leq 0.7$) on $f'_\delta(\xi)$ and $\theta_\delta(\xi)$ respectively. For large λ , it is described that the velocity profile rises and vanishes at the surface of cylinder as visualized in Figure 3. Because of this, very less declination in field temperature is noticed for various entries of that can be observed in Figure 4. Hence, local Nusselt number also rise as displayed in Table 2. Figures 5 and 6 demonstrate profile of temperature and concentration with change in value of Nb ($0.1 \leq Nb \leq 0.7$). A random striking of poised elements is the Brownian motion that shows its existence when liquid particles strike with each another will create an arbitrary motion and that performs an increment in temperature as well as in concentration as displayed in Figures 5 and 6. Brownian motion of suspended (Pendulus) particles is the main reason for rise in kinetic energy of nano particle and due to this it enhance the thermal boundary layer thickness. Consequently, temperature gradient falls down for higher Nb (See Table 3).

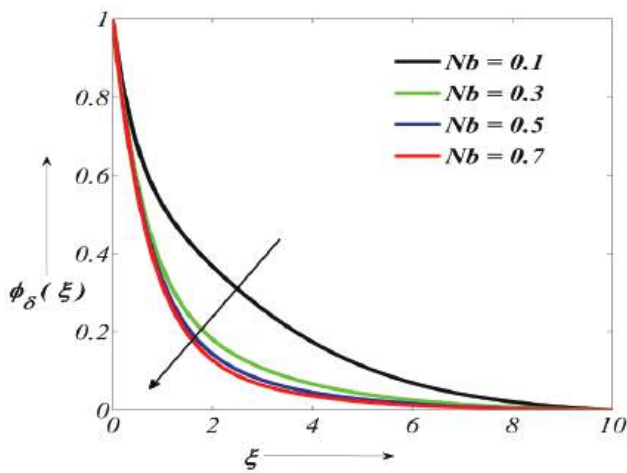


Figure 6. Concentration profile against Nb .

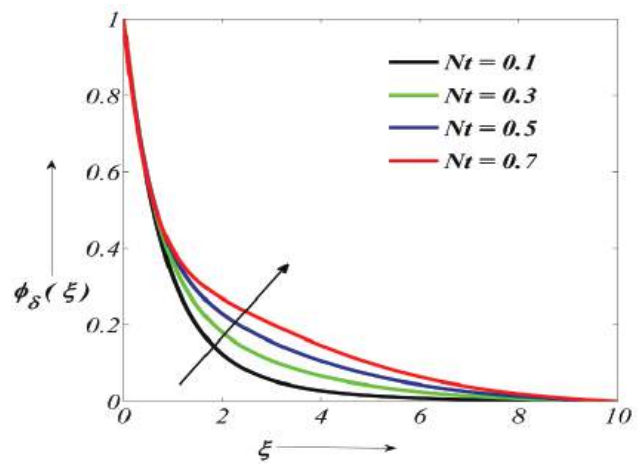


Figure 8. Concentration profile against Nt .

Table 5. $f''_{\delta}(0)$, $-\theta'_{\delta}(0)$ and $-\phi'_{\delta}(0)$ for distinct values of fluid parameter Rd .

Rd	$f''_{\delta}(0)$	$-\theta'_{\delta}(0)$	$-\phi'_{\delta}(0)$
1	-0.67682	0.46528	1.16666
3	-0.67682	0.43887	1.16466
5	-0.67682	0.42966	1.16669
7	-0.67682	0.42559	1.16790

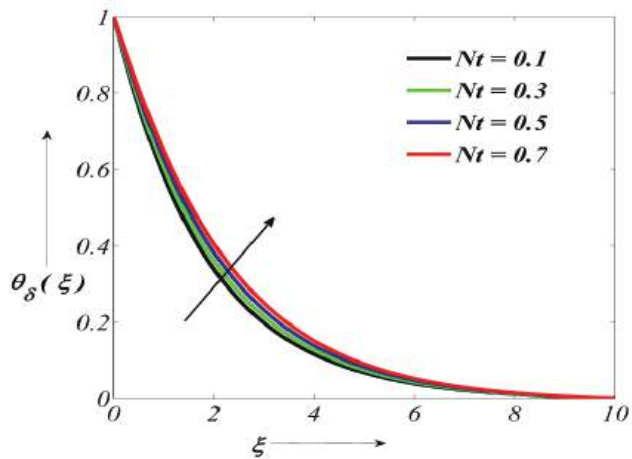


Figure 7. Temperature profile against Nt .

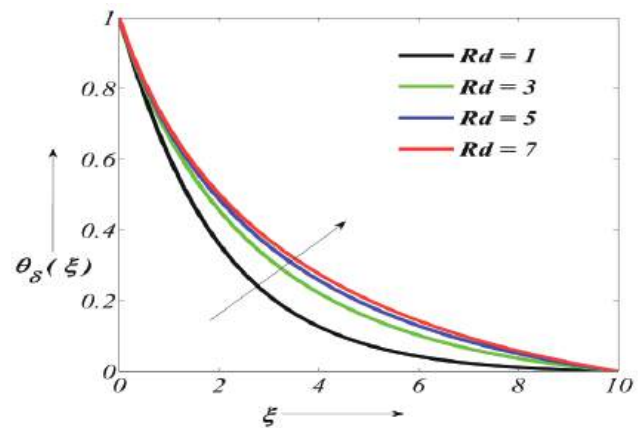


Figure 9. Temperature profile against Rd .

Little increment is seen in $\theta_{\delta}(\xi)$ for higher Nb . For higher value of Brownian motion parameter Nb , collision between fluid particle will by which nanoparticle concentration declines. Also, concentration boundary layer thickness decreases as observed in Figure 6. Figures 7 and 8 presents the $\theta_{\delta}(\xi)$ & $\phi_{\delta}(\xi)$ plots affected by the thermophoresis, respectively. With augmentation in thermophoresis Nt , there is a declination in the temperature gradient, due to

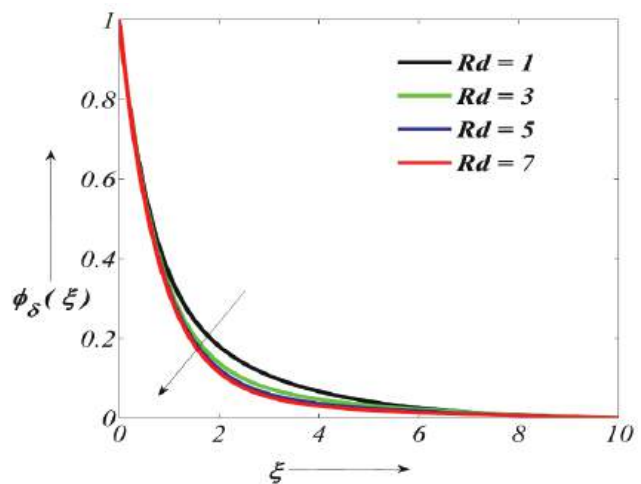


Figure 10. Concentration profile against Rd .

which nanoparticle conduction falls down. Consequently, ultrafine elements migrate from hotter to cooler zone, that simultaneously increasing the width of the boundary layer and raising the temperature. This increase in thermal boundary layer thickness causes a decrease in the Nusselt number, as observed in Table 4. Conversely, in Figures 8, $\phi_\delta(\xi)$ increases and the width of the concentration boundary layer also increases with higher values of the thermophoresis parameter. Additionally, plots are distinct for $0.0 \leq \xi \leq 8.2$ (approx) and converges with $\xi \rightarrow \infty$. In Figure 9, fluid temperature rises as the radiation parameter Rd increases, indicating a higher surface heat flux associated with higher Rd values. Additionally, the thermal boundary layer thickness expands with increasing Rd , suggesting a broader spread of heat transfer over the surface.

Table 6. $f''_\delta(0)$, $-\theta'_\delta(0)$ and $-\phi'_\delta(0)$ for distinct values of fluid parameter d .

d	$f''_\delta(0)$	$-\theta'_\delta(0)$	$-\phi'_\delta(0)$
0.0	-0.79492	0.47739	1.18293
0.2	-0.59077	0.45608	1.15436
0.4	-0.47280	0.44288	1.13680
0.6	-0.39512	0.43377	1.12477

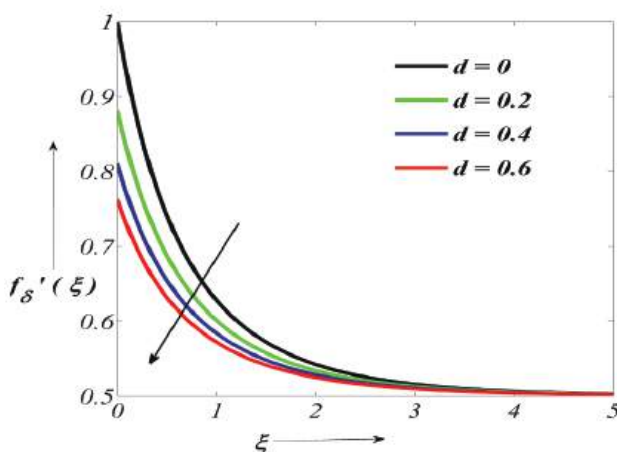


Figure 11. Velocity profile against d .

Figure 10 demonstrates that fluid concentration and concentration boundary layer thickness decline as values increase, implying more efficient mixing or dispersion. Table 5 reveals that the Nusselt number decreases while the Sherwood number increases as increases, indicating changes in convective heat transfer efficiency and mass transfer efficiency, respectively. Figure 10 illustrates the effect of d (0.0, 0.2, 0.4, 0.6) on the velocity distribution. It's noticeable that fluid velocity falls down as d rises up, as

depicted in Figure 11 and plots are distinct for $0.0 \leq \xi \leq 4.0$ (approx) and follows convergence criterion for $\xi \rightarrow \infty$. The primary reason for this velocity reduction is the occurrence of slip conditions. Consequently, the width of the boundary layer decreases with increasing values of velocity slip parameter d .

CONCLUSION

P.D.E's are converted into a set of O.D.E's by the use of similarity transformation on it. Final results include the velocity, temperature and concentration distribution of MHD Casson nanofluid flow and impact of outer velocity induced by stretching cylinder. Final outcomes of the study are:

- Augmentation in free stream velocity shows the increase in fluid velocity.
- The Skin friction coefficient intensifies with increment in free stream velocity parameter.
- Temperature of fluid reduced with increment of λ whereas noticed opposite pattern in case of Nb , Nt , d and Rd .
- Heat transfer rate falls down by 71.31% with increment in free stream velocity as $0.1 \leq \lambda \leq 0.7$.
- λ shows more deviation in temperature profile between fluid layer. On the other hand, Rd left more impact on temperature of fluid as comparative to the other parameter like Nb , Nt and d .
- Concentration of fluid reduced for higher λ , Nb and Rd and opposite trend has been seen in cases of Nt and d .
- Nt and Nb shows more deviation as comparative to other parameter in case of concentration of fluid.

Current study is limited to the incompressible laminar flow of nanofluids induced by stretching cylinder in presence of free stream velocity and thermal radiation. This analysis can be extended to three dimensional extended surface induced with Cattaneo-Christov heat flux, melting surface utilizing Lie symmetry analysis.

NOMENCLATURE

λ	Outer velocity parameter
Rd	Radiation parameter
d	Velocity slip parameter
Nt	Thermophoresis
n	Non-linear stretching parameter
L	Characteristic length
Rc	Constant
Nb	Brownian motion
U_w	Stretching velocity
r	Radius of cylinder
u, v	Horizontal & radial velocity
T	Temperature
C	Concentration
T_∞	Ambient temperature
D_B	Brownian diffusion coefficient

D_T	Thermophoretic diffusion coefficient
C_∞	Ambient concentration
σ	Electrical conductivity
B_0	Magnetic field strength
ρ	Density
e	Stretching velocity parameter
α	Thermal diffusivity
τ	Ratio of heat capacity
$N\mu$	Velocity slip factor
Pr	Prandtl number
M	Magnetic parameter
θ_δ	Temperature distribution
f_δ	Velocity distribution
Sc	Schmidt number
Rd	Heat radiation
ϕ_δ	Concentration distribution
Q	Heat generation
Sh_x	Sherwood number
Nu_x	Nusselt number
C_w	Wall concentration
T_f	Fluid temperature
C_f	Skin friction coefficient
k	Thermal conductivity
μ	Dynamic viscosity
Re_x	Local Reynolds number
γ	Curvature parameter
b	Constant
N_0	Constant wall slip velocity

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

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Research Article

Design and thermal analysis of light weight radiator for automobile applications

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ABSTRACT

This present study deals with the simulation of the radiator and its fin efficiency, where the coolant is used as air, and the working fluid is used as water and air to measure the thermal performance. Heat dissipation by conventional materials has enormous demerits. Hence, lightweight aluminium material has considerable benefits in better heat dissipation and energy transfer. The methods used in this analysis for design SolidWorks and analysis are done by cosmos software. The materials used here were copper for the fins and aluminium as the outer cover material. The properties such as thermal conductivities, heat capacity, and vapour absorption rates were calculated. From the experimental results, it was observed that the heat transfer rate was achieved in the range of 30 degrees Celsius with a change in temperature difference. An increase in fins from 10 to 20 number tends to increase the heat transfer rate by 10 to 15% as compared to other conventional materials. The results also concluded that there would be an increase in the overall efficiency of the radiator. The results also reveal that implementing copper tubes with the combination of aluminium fins has an extent decrease in heat transfer rate. Hence, a radiator for heat transfer applications can achieve a marginal temperature difference of 25 degrees Celsius to 30 degrees Celsius.

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INTRODUCTION

The dissipation of heat from one medium to another with the convective medium or conduction medium by applying a radiator is a critical factor for upcoming automobile design [1]. Even the engine in the automobile is the heart, but the radiator is the main device for the automobiles, and it dissipates the heat from the different hot temperatures [2]. The temperatures may vary between 1500 degrees Celsius to 2100 degrees Celsius. Hence, it must be designed with that bee designed using different materials [3]. The selection of materials depends upon the heat-conducting capacity of the material as well as the material life span characteristics [4]. The material also possesses high corrosive resistance and better thermal dissipation. The thermodynamic performance of the radiator should be very high compared to other cooling methods adopted for engine cooling [5]. One more major drawback of the analysis is the selection of optimal cooling fluids owing to the substantial temperature heat dissipation parameters [6]. When concerned with the light addition of additives, the right selection of coolants improves the engine's cooling performance [7]. Hence, selecting the best cooling additives that corrugate the material performance and the system performance of the radiator performance is desirable. The best materials can be chosen as lightweight materials because heat dissipation must rapidly occur with optimal cooling characteristics [8]. According to [9], the variation in fluctuations of fin efficiency depends upon the fin's stiffness characteristics and material properties, which offer resistance during the internal resistance to the flow of conduction of the coolant used in the radiator. The flat-shaped fin with a thickness of 3.8 mm gives an optimal performance with a fin effectiveness of 89%. Results also reveal that dynamic analysis with a combination of composite materials improves the heat conduction capacity of the system [10]. The fin's efficiency was gradually increased by changing optimal performance parameters such as the radiator core's stiffness and poison ratio characteristics [11]. It is also required to improve the thermodynamic properties of the fluid by utilizing the pumping fans with the aid of convective transfer mediums, which can ideally be operated in between minimum to maximum temperature conditions, which is extremely good under the operating temperature of 200 degrees celsius and atmospheric pressures [12]. From this investigation, it is understood that the system's efficiency depends upon the convective medium and type of working fluids used in the analysis [13]. The fouling factor determines the key performance factor to maximize the fin's efficiency. The impurities that occur at the inner side and surface of the material, which affect the system's performance, are predicted by this factor [14]. Many researchers [15,16] reported that radiators made up of stainless steel have more fouling factors because inadequate corrosion resistance offered by the material tends to cause poor system performance. The optimal

performance is 90% in the inner load case when the nano-fluid of Al_2O_3 particles is blended with graphene fluids as a coolant. Hence, this paper focuses on the analysis of radiators with Multiphysics software to find optimal cooling and minimal residual time, which can be implemented for automobile applications. The novelty of this work is to replace the steel materials with aluminium-based materials for better thermal dissipation and heat conduction capacity. The mini prototype model is designed, analyzed, and fabricated in a research centre that is available with well-equipped lab facilities. The source of working fluid utilized here is water and air. The objectives are to determine the heat dissipation capacity of the radiator with these two working fluids and to conclude which one is best for practical applications. The research gap and the novelty in work, followed by many failures, were faced by the conventional radiators made up of steel and composites in terms of less durability and high corrosion resistance. Hence, research related to aluminium material is less. This research gap and novelty of utilizing the aluminium radiator with light weight has enormous benefits in less corrosion resistance and high durability, causing better heat dissipation than other materials. The work is a prototype model constantly working further for real-world application implementations. The authors find difficulties during the fabrication of the model. They are willing to work further to add a few microns ranging from 300 microns to 400 microns of ceramic coatings to enhance the stiffness characteristics of the radiator. This model was developed to fit automotive applications and HVAC systems concerned with lightweight applications where the mass flow rates of water and air are quite simple compared to other applications such as marine, aerospace, and industrial sectors. The motivation behind the study is the utilization of lightweight radiator material made up of aluminium has enormous benefits in terms of high dissipation and low cost to avoid less corrosion behavior and less fouling factors, owing to the prevention of frequent replacements of radiators in the automobile industry.

MATERIALS AND METHODS

Materials

Table 1 shows the materials required and specifications for this analysis. Fins are made of aluminium alloy [17]. Aluminium is chosen as a fin material due to its lower density and ability to withstand higher conduction heat than other materials. Table 2 tabulates aluminum's properties.

Methods

The entire setup was drawn in SolidWorks, as shown in Figure 1. The radiator's cooling capacity estimates the performance [18]. The selection of materials is discussed in Table 1. In contrast, the meshing strategy involved here is an adaptive mesh strategy with an elemental size of 10 mm selected to minimize the computational time of the

Table 1. Material properties and specifications

S. NO	Material	Type	Quantity	Length (m)	Breadth (m)	Thickness (mm)	Diameter (mm)
1	Copper Tube	Metal	1	5	3	1	10
2	Aluminum sheet	Metal	50	2.7	2.7	1	—
3	Thermocouple	Sensor	3	1	—	—	2
4	Temperature controller	Digital	2	—	—	—	—
5	Coolant	Liquid	10 Lit	—	—	—	—

Table 2. Properties of aluminum

S. NO	Material	Aluminum
1	Point of Melting	661.3 °C
2	Point of boiling	2460 °C
3	Thermal Conductivity (K)	241.0 W/mK
4	Thermal Expansion Coefficient (α)	-
5	Density	2.8 g/cm ³

analysis. The entire analysis is taken for a minimal time of 20 minutes, from the preprocessing stage to the post-processing stage. The different types of materials are studied in the literature study. The study shows aluminium materials are less weighted and can conduct more heat dissipation than other materials [19]. One more merit behind using aluminium as a fin material is that it is lighter and gives higher heat dissipation than other materials. Hence, aluminium is the best insulating material for dissipating heat and transforming the heat at a convective heat medium.

The model is designed and analyzed in multiphysics with different boundary conditions.

Radiator Design

The radiator and associated parts were modelled using 3D modelling software such as SolidWorks. Each part is drawn individually and assembled using the assemble option available in the Solid Works software [20]. SolidWorks is a friendly tool that can minimize errors during the assembly of radiators. The assembled model is shown in Figure 1. Figure 2 represents the front view of the radiator with the assembly of fins. The fins are corrugated in nature, improving the system's efficiency [21]. The accurate enhancement during the assembly of fins with radiator core and fins surface using surface morphology is possible in solid works. Also, the principal merit behind the usage of this tool is the efficient technique, which enhances the optimization of the design with minimal time to complete the initial part design to assembly. Hence, the perfect assembly is done using SolidWorks software, which is more accurate than other software [22].

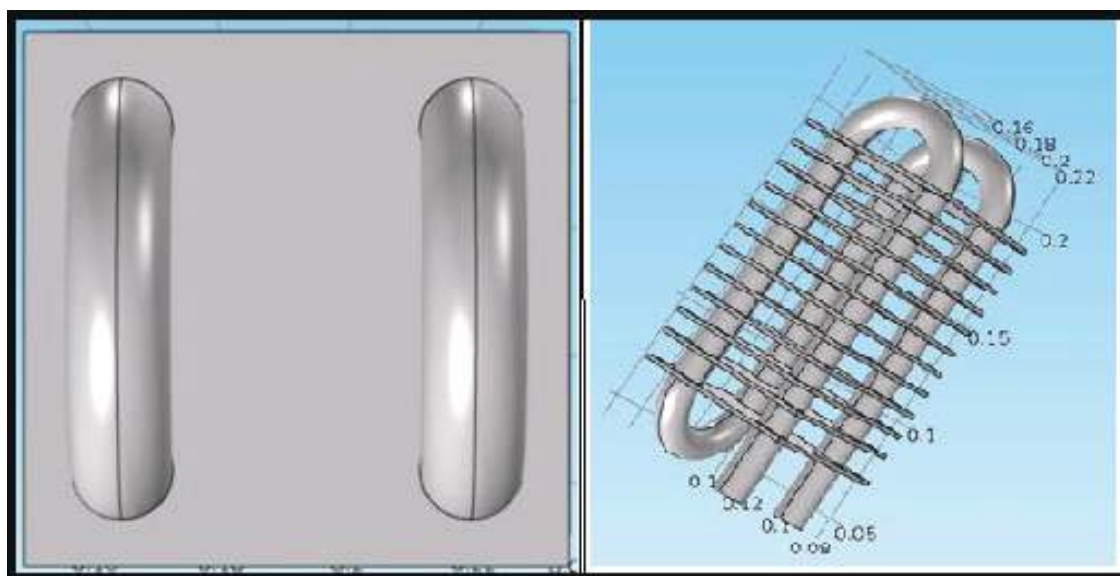


Figure 1. Top and side view of radiator developed in solidworks software.

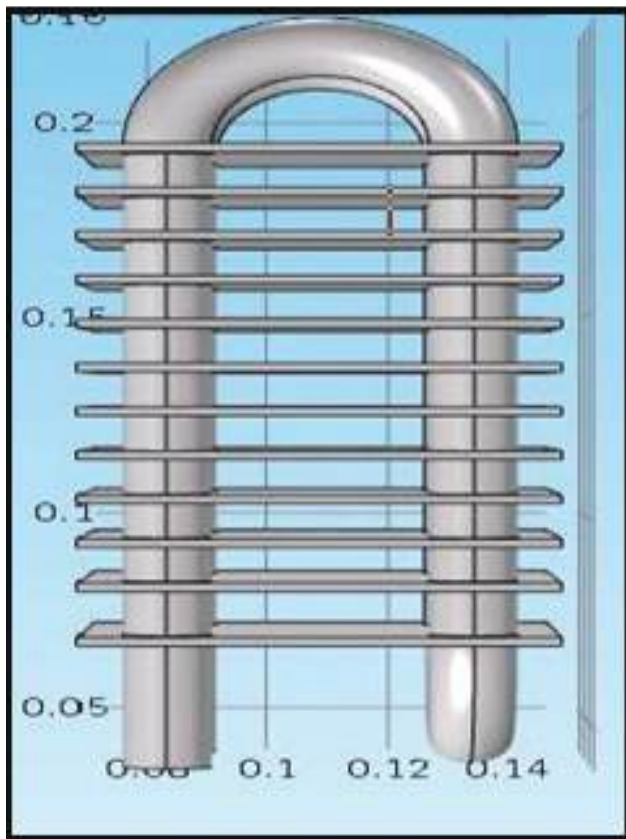


Figure 2. Front view of the radiator with fins.

ANALYSIS OF RADIATOR

Meshing Strategy

Figure 3 represents the meshing strategy of the entire radiator developed in cosmos multiphysics. The mesh elemental size of 10 mm was selected, followed by tetrahedron mesh for achieving accurate results, and the no of nodes and elements formed during the analysis is 40980 nodes

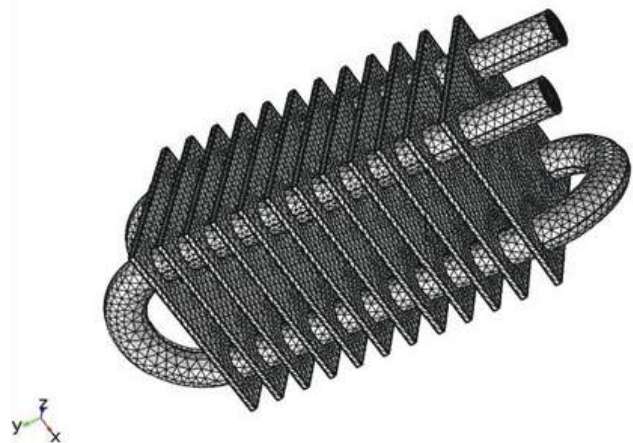


Figure 3. Meshing model of the radiator.

and 20200 elements for the radiator and 20100 nodes and 9015 elements for fins with fewer computational time of 15 minutes. The mesh convergences are obtained by 0.4%, which gives optimal deviation and less standard deviation with possible finer results. The other essential parameters, such as skewness, Jacobian, and aspect ratio, were checked, and these parameters were satisfied, which had more accurate results [23].

Boundary Conditions

The term boundary condition is defined as the term that acts as a single point of force contact, which is all the forces acting concerning the displacement due to rotation. Bending and transverse forces acting on it, technically called the degree of freedom, are zero and cause stringent resisting motion [24]. Figure 4 represents the degree of liberty applied at the top surface of the fin, the top surface of the radiator, and the bottom surface of the radiator and fin's bottom surface. Hence, all the forces acting over these surfaces become zero in terms of translational, rotational, sliding, and twisting motions [25]. The inlet condition of

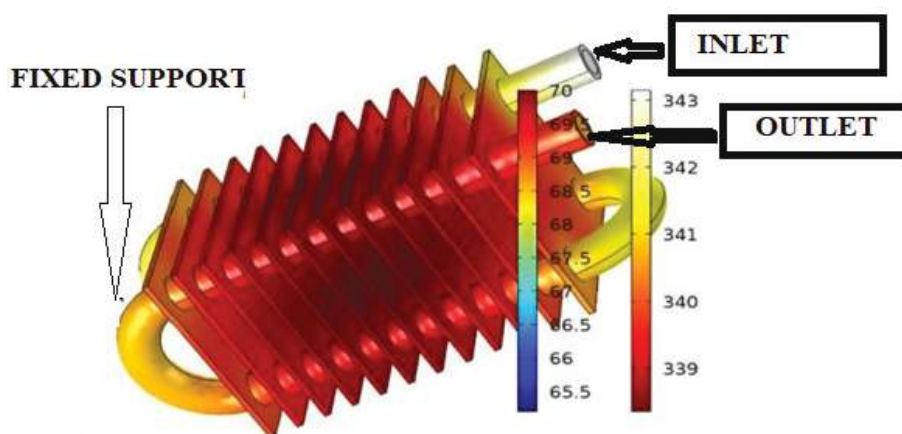


Figure 4. Boundary conditions.

the working fluid at different velocities is tested for three different mass flow rates working fluid as water and air. This stage is predominantly critical and concerned with the various velocities and other pressures. At the bottom portion of the radiator, the displacement is arrested; hence, the DOF is zero at these conditions. These values are governed by Navier stroke, which is ideally designated for real-world scenarios [26].

Governing Equations

The required equations for determining the mass flow rates of the working fluid and the base fluid depend upon the main equations, such as the continuity equation, which depends upon the mass and the Navier equations. Both equations describe the path of the mass flow rates and the tangential accelerations of the working fluids at different velocities.

The continuity equation defined in equation (1) referred to in [27] depends upon the temperature differences of the densities of the base fluids and the irrotational turbulent conditions concerned with the pressure, velocity, and time of the working fluids.

The Navier stroke equation as shown in the below equation (2) referred from [28], describes the rotational flow of the hot working fluids and its significance parameters concerned with velocities, pressures and working substance densities.

$$\rho H (rV + - + v z) = g\rho - + \mu [([v] + - +) - - \quad (2)$$

The required equation (3) referred to [29], which is used for determining the specific heat of the hot working fluids depends upon the time concern factor and the coefficients of friction that are released from the hot gas surfaces to the radiator pipes.

$$\rho H (\partial P + r v + + z v) = U [(p) + +] + \mu \text{-----} \quad (3)$$

Heat Dissipation Equations

The Amount of heat dissipation for the different mass flow rates is described by the following equation (4) referred from [30]

$$\text{-----} \quad (4)$$

Q = Heat flux or heat dissipation concerned with the amount of heat flow in w/m²

m = Amount of mass flow in kg/s

The mass flow rate that passes through the radiator can be measured with the help of a flow gas meter, and it is expressed by the equations (5) referred from [31]

= The volume of the hot gases passing through the radiator.

= The density of the working fluid in kg/m³

The theory of mixing is used here for calculating the fractions concerning the Volume of working fluids followed by the following equation (6) referred from [32]

Density of the base fluid in kg/m³

Density of the hot working fluid concerned to working fluid in kg/m³

The density of the working fluid is the main physical parameter that signifies the thermodynamic performance of the radiator. The higher the density causes, the higher the dissipation of the molecules. Hence, the dissipation efficiency is very low in natural convection mode, in the range of 5.5% less than in forced convection modes. The Reynolds number identifies the nature of the working fluid, and its significance concerns the radiator's performance. In addition to the Reynolds number, the Nusselt number has more impact on identifying the nature of dissipation concerned with convective heat transfer [33].

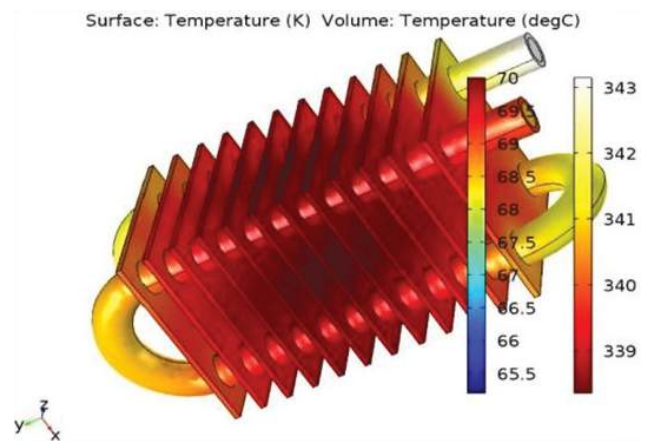


Figure 5. Post-processing results obtained.

Table 3. Testing Results

SNO	Observation	Water	Air
1	Inlet Temperature (°C)	65	27
2	Outlet Temperature (°C)	44	32.43
3	Mass flow rate (Kg/s)	0.0238	0.1459
4	Specific heat (kJ/kg°C)	4.187	1.005
5	Thermal conductivity (W/mK)	0.66	0.024
6	Density (Kg/m ³)	998	1.225

Table 4. Calculated values

S. NO	Observation	Value
1	Heat transfer coefficient (U)	350 W/mk
2	Log mean temperature difference (LMTD)	18.25 °C
3	Amount of heat transferred (Q)	2869.93 W
4	Capacity ratio (C)	0.679
5	NTU	1.68
6	Effectiveness (E)	0.65

Table 5. Cost benefit analysis

S. NO	Equipment/Component	Cost	Durability	Quantity	Total cost
1	Aluminum sheet	1800/-	5 years/till rust	1	1800
2	Copper tube	790/-	5000 cycles	1	790
3	CNC Sheering	6/-	-	50	300
4	Thermocouples	840/-	3years	3	2520
5	Temperature controller	4500/-	3 years	1	4500
6	Welding	60/-	-	6	360
7	Coolant	20/-	10 years	3	60
8	Transportation	300/-	-	3	900
9	Brine solution	500/-	10 years	1	500
10	Misce Other costs	1200	-	1	1200
11	Operating costs	20	Infinite	1	20
12	Total cost				12900/-

Table 6. Uncertainties of instruments used

S. NO	Devices	Deviations	Range	%Uncertainty
1	Load Indicator	± 2.7 kg	0-150 kg	0.55
2	Working fluid Temperatures	± 2.2 °C	0-840 °C	2.22
3	Brazing temperatures	± 7 °deg	200-580 deg	2.1
4	Stopwatch	± 0.8 s	-	0.65
5	Thermocouple	± 0.8	0- 700 deg	1.8
6	Pressure gauge	± 2 bar	0-800 bar	1.8
7	Coolant Mass flow	± 25 kg/s	0-305 kg/s	2.3

The representation of data that experimentally occurred during the simulation is termed post-processing results. It indicates the physical occurrences by simulation and the significant nature of occurrences in pictorial representation [34]. Figure 5 represents the post-processing results that occurred during the simulation. It was observed from Figure 5 that the maximum heat transfer occurred at the radiator fins up to 343 W/mk with the rise in temperature differences of 70 degrees Celsius. This is due to the phenomenon of heat conduction capacity of heat transfer by the conduction of fins made up of aluminium, which has a limited thermal conductive capacity of 345 W/mk [35]. This is due to the collision of oxygen molecules with others. This interrupted collision tends to conduct the heat to a limited extent [36]. Table 3 represents testing results and Table 4 represents calculated values.

Table 5 represents the cost-benefit analysis involved in the fabrication of this project. The materials were purchased from the local manufacturers available at chennai, and the aluminium sheets were purchased from the cosmos aluminium dealer in chennai; the thermocouples were purchased at chennai. A few products, such as coolant, are purchased from local coolant dealers in chennai.

Uncertainty Analysis

The main essential criteria satisfy the precision and accuracy of the radiator's fabrication, and possible errors depend upon the radiator's least calibration compared to the measuring instruments. Table 6 significantly represents the uncertainties that occurred during the radiator fabrication trials. Table 7 depicts the accuracies of parameters used in experimental analysis. The trials are carried out with three definite readings calibrated as a source of Gaussian distribution calculations to identify possible errors.

Table 7. Parameters and its accuracies

S. NO	Parameters	Percentage of uncertainty
1	Working fluids temperatures	± 1.9
2	Mass flow rates	± 1.8
3	Coil bending Pressures	± 2.1
4	Fins heat transfer	± 2.2
5	Flame temperatures of brazing	± 1.76

The required correlations from various kinds of literature have been collected and the range of accuracies obtained from different experiments that calibrated is defined by equation [7], referred from [37]

Equation (7) defines the deviations obtained during the experimental segment and the deviations depend upon the atmospheric variables such as pressure, Temperature, and Densities of the working fluid, causing the least possible errors

The maximum possible readings that caused the errors or range of calibrated readings is defined by Equation (8) referred from [38]

The range of uncertainties that developed during the experimental segment followed by Equation [9] referred from [39]

= Obtained deviations and calibrated readings.

Fabricated Model

Fabrication is required before testing the performance of the radiator. The project laboratory does all the fins and is equipped with the model setup. The above Figure 6 illustrates the fabricated model. The transformer with 440 volts is used to carry out plasma arc welding, and aluminium baffle plates were inserted between the tubes [40]. The copper tubes were tough to bend. Hence, the required inclinations were bent using the bending machines to avoid creeping and fatigue disturbances. The tubes were cleaned with high detergents to achieve the best results [41]. The cleaned tubes and aluminium fins are exposed to air to achieve dryness. Before experimenting, the tubes are cleaned with salt sheets to avoid the corrosion effects [42]. The testing phase is important in finding fouling factors. The authors

have used salt spray techniques on the aluminium plates to determine their corrosion and durability. The aluminium plates are kept in salt spray with 10% sodium hydroxide, and there is no physical change in the behaviour of the aluminium radiator [43].

RESULTS AND DISCUSSION

Effect of Temperature Distributions

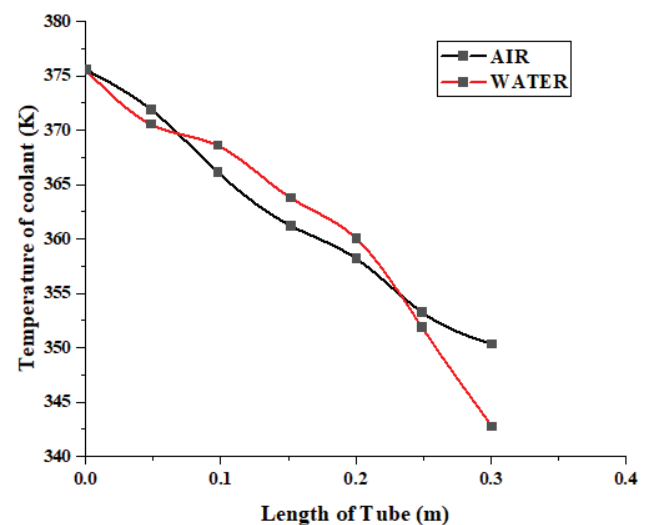


Figure 7. Temperature distribution of coolants.

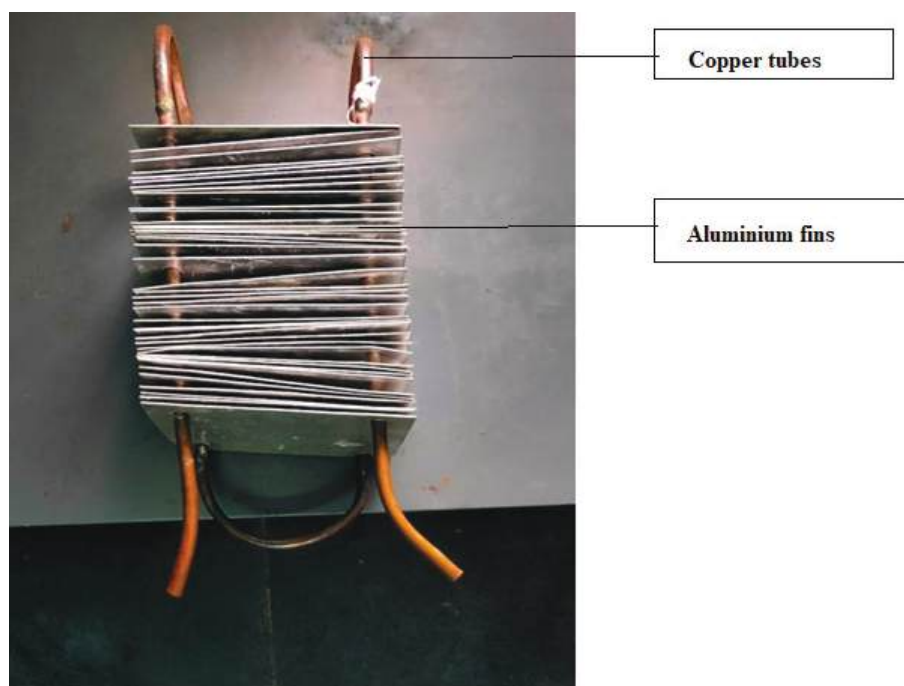


Figure 6. Fabrication model.

The temperature distribution is an essential factor in evaluating the thermal performance of the radiator; the higher the thermal dissipation, the higher the thermal efficiency of the radiator. The overall efficiency of the radiator depends upon the thermal differences and heat dissipation factors that decide the capacity of the heat transfer medium. Figure 7 explains the temperature differences obtained in the aluminium radiator experiment. It is observed from Figure 7 that the temperature distribution of water increases quite a lot with tube length. Because the specific heat of water is quite high as compared to water [44]. The specific heat of air is quite low as compared to water; hence, the medium of air as a coolant is quite tricky. As Temperature increases, the heat conduction for air decreases, but for water, as temperature increases, the heat conduction and heat dissipation rate also increase. It acts as a better conductor due to the higher specific heat and transmission capacity as compared to air [45]. The effectiveness achieved by water is good, and hence, water can dissipate heat at an enormous rate by conduction, a convection phenomenon compared to air [46].

Effect of Pressure Drop

The pressure drop in the radiator is essential in determining the pressure coefficients and their dynamic stagnation characteristics across the tubes. In other words, pressure drop factors can determine the coolant flow in the radiator with corrugated obstacles [47]. Figure 8 represents the pressure drop over coolants subjected to two different working fluids, water and air. It is keenly noted that the mass flow rate of the coolant is increased with definite intervals and tends to decrease in pressure gradually [48]. The length of the tubes corrugated with the fins increases and tends to decrease in pressure, whereas other materials possess high-pressure drops due to high density and mechanical characteristics [49]. The lower the weight of the aluminium, the less pressure drop, and the maximum

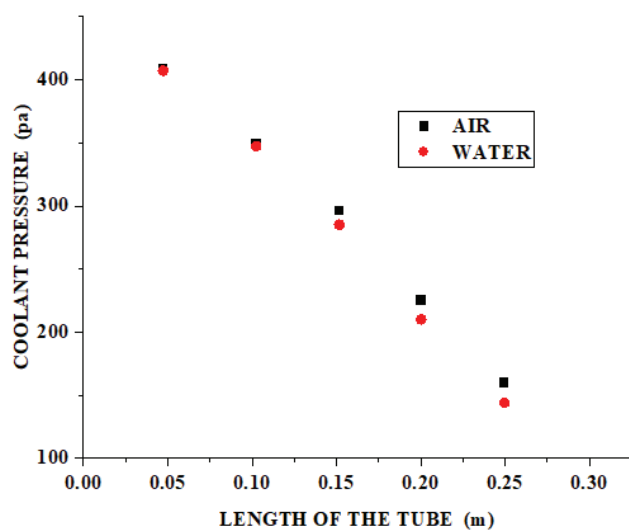


Figure 8. Pressure drop over coolants.

drop is achieved, up to 400 pa, subjected to maximum deflections. The pressure drop for water is low compared to air due to higher density differences occurring due to the forced and turbulence effects of water due to a convective medium [50].

Effect of Mass Flow Rate and Cooling Efficiency

The flow of air as a cooling medium to dissipate the heat is required to predict the cooling capacity of the radiator [51]. The thermal performance of the radiator depends upon the heat dissipation characteristics. Figure 9 represents the mass flow rate of the air as an external source, and the internal working fluids were utilized as water and air. It is predominantly noted that the working fluid of air has low dissipation compared to water [52]. The lower the mass flow rates from 100 kg/hr with the help of forced convection; the desired efficiency was noted as 50%; If the Density of the air increased from 100 kg/hr to 150 kg/hr; the cooling efficiency of the radiator for the working fluid air is increased from 50 to 57 % and for working fluid water the radiator efficiency is achieved from 53% to 64%. At higher optimal loading conditions, the optimal cooling efficiency of the radiator is increased up to 67% for the radiator operated by air as a working fluid, and for the radiator operated by water as a working fluid, the optimal efficiency is achieved up to 70%; the corresponding mass flow rates are achieved up to 275 kg/hr; It is desirable to note that the cooling efficiency for the working air-water medium is good compared to the air-air medium [53]. Water's higher thermal conduction behaviour gives optimal cooling effects compared to air. Hence, water is required as a coolant for optimized efficiency [54].

Mass Flow Rate and its Approach

It is experimentally needed to predict and analyze the approach based on temperature differences. This approach

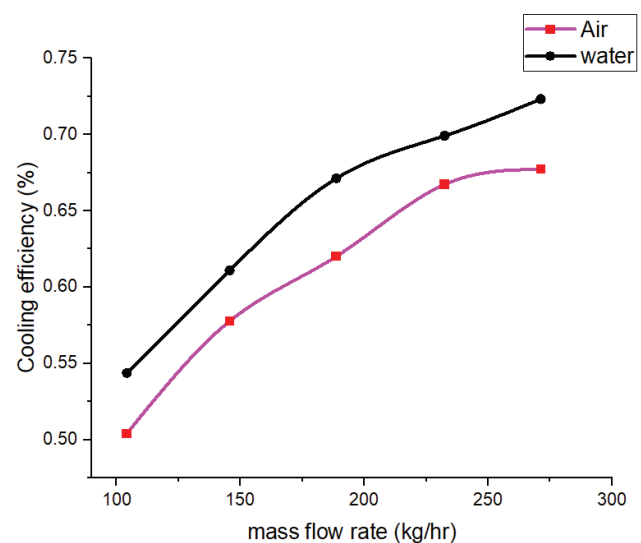


Figure 9. Mass flow rate and its cooling efficiency.

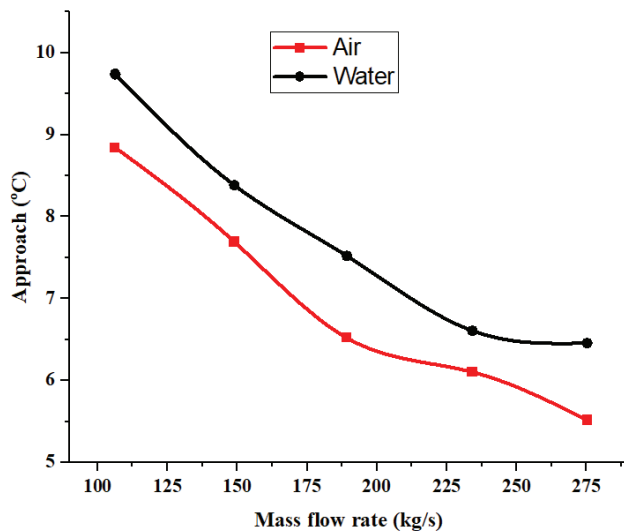


Figure 10. The approach of Temperature and its mass flow rate ($\Delta T = 30^\circ\text{C}$).

is highly used to determine the desired cooling effects during heat dissipation [55]. Figure 10 signifies the average temperature dissipation for the working fluids air and water. The desired effectiveness is achieved for the working fluids air with a minimal temperature difference of 10 Degrees Celcius. When the temperature reaches 20 Degrees Celsius, the corresponding mass flow rate is achieved at a maximum of up to 275 kg/hr. The mass flow rate for water as a working fluid; the corresponding temperature difference is achieved by 30 Degree Celsius because a higher dissipation rate is achieved through the water [56]. The main reason why it is the best medium with the difference in temperatures is that it acts as the best thermal conductor and the turbulent forces and higher thermal differences are created during the evolving period. The losses during the heat transfer phenomenon concerning inside the radiator are overcome by the radiator fins made up of aluminium. Hence, water as a cooling medium gives optimized results compared to air.

Cost-Benefit Analysis

The welding is done at the college campus. The CNC shearing, welding, and brazing processes are done with the help of well-trained non-teaching faculties with the least atmospheric errors. Owing to the cost study, the cost of fabricating this radiator is 12900. Which is very cheap, and the manufacturers can get enormous benefits compared to radiators made of other materials. Hence, implementing this radiator can give enormous benefits in terms of better heat dissipation characteristics and the best and cheapest cost compared to other materials. Therefore, every customer or radiator manufacturer interested in working on lightweight materials can use aluminium-based radiator fins for better heat dissipation at the lowest cost. Due to the

cost-benefit analysis, aluminium is very cheap compared to other materials like steel and is more durable than steel. This is because the aluminium material's lighter weight and less corrosion resistance offered by the aluminium material results in better performance than steel and other conventional materials.

CONCLUSION

The design of the radiator is done using Solid works software, and the entire setup is imported into multiphysics software for analysis. However, the material properties of the radiator and the elemental characteristics are mentioned in the preprocessing stage of the cosmos application software. Among the various literature and various researchers' studies, it is noted that the effectiveness of the radiator depends upon the heat-conducting medium of the radiator material and the weight of the radiator. In addition, the radiator's mechanical properties must be analyzed before the radiator must be designed. The following observations were concluded during the analysis followed by

- ❖ The maximum heat conduction capacity of the fins made up of aluminium can able to withstand 345 W / MK.
- ❖ The maximum heat transfer capability of the fins is 2900 watts.
- ❖ The capacity ratio of aluminium fins can be achieved up to 0.8, whereas it can be limited to 0.6 for conventional materials. Hence, aluminium possesses better heat dissipation and better capacity than traditional materials.
- ❖ Owing to thermal heat dissipation through rapid cooling, which is simulated in the Cosmos software, the results show that rapid cooling is possible within the rapid cooling rate of 22 Degree Celsius, in contrast to other materials.
- ❖ Since the conductivity medium or source here is utilized as air or water, both the medium and the aluminium materials possess better heat conducting capability and better heat dissipation in terms of conduction, convection, and radiation.
- ❖ The thermal diffusion of the aluminium alloy-based radiator is better as compared to a conventional radiator; hence it is suggested that the use of an aluminium-based radiator gives predominated efficiency in terms of better heat dissipation;
- ❖ The life span of the radiator is increased if the working fluid is air because aluminium possesses high resistance capacities compared to other fluids.
- ❖ Aluminum's thermal conductivity is excellent compared to other conventional materials because aluminium has a high resistance to heat.
- ❖ Aluminum fins have proven optimal performance in heat dissipation, lower thermal elongation, and higher stability. Hence, aluminium was considered a material for radiator design and fabrication analysis.

- ❖ The mass flow capacity of the radiator for air is low compared to water because the specific heat of water is very high compared to air. This will decrease the performance in terms of heat dissipation but improve the life span of the radiator because fewer corrosion formations will occur if the working fluid is air.
- ❖ Overall, the fabrication and implementation of the radiator give good results owing to thermodynamic performance and life span characteristics.
- ❖ Owing to the cost study, the cost of fabricating this radiator is 12900.
- ❖ Hence, every customer or radiator manufacturer who is interested in working with lightweight materials can choose aluminium-based radiator fins for better heat dissipation at the lowest cost.

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.

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Research Article

Ultrasound-assisted desulfurization of crude oil using sodium hydroxide

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ABSTRACT

To meet the required standards on air quality and emissions from engine exhaust, several countries have instituted policies to set and reduce the limit of crude oil and petroleum product sulfur concentration to very low levels. This research studies the effectiveness of combining the assisted ultrasound desulfurization, and the traditional sodium hydroxide desulfurization methodologies, to lower the amount of sulfur contained in fuels produced even further. The sample in question was 25mL of crude oil obtained from the East Baghdad Oil Field in Iraq, containing 4.31 wt.% sulfur. The main underlying parameters of the UADS process; sodium hydroxide concentration, (temperature, 20-80 C°), reaction time (10-30 min), and intensity (30-60%) of ultrasound. The study contrasted the ultrasound desulfurization of crude oil performing the sodium hydroxide process with 'standard' caustic desulfurization, using the same parameters. The synergy of NaOH and ultrasonic irradiation in desulfurization of crude oil has surpassed their individual capabilities. The experimental data also compares the ultrasonic assisted empirical and reaction measures of the crude oil deuterium capture. She concluded that the maximal sulfur concentration deduced from the system was close to 68% at optimal conditions of 3M NaOH concentration, reaction temperature 60 C°, 30 minutes, with 40% of collimated ultrasound.

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INTRODUCTION

Sulfur compounds in crude oil pose significant environmental challenges due to their detrimental effects, including catalyst deactivation and corrosion in petroleum production, processing, and pipeline systems [1]. These compounds can be classified into two main categories based on their chemical behavior: active and inactive sulfur. Active sulfur compounds—such as hydrogen sulfide, mercaptans, and elemental sulfur—can directly interact with

metallic surfaces, leading to equipment corrosion. On the other hand, inactive sulfur compounds, including sulfides, disulfides, and thiophenes, do not react directly with metals but can generate hydrogen sulfide when heated, owing to their poor thermal stability [2]. Elevated sulfur content in crude oil often results in similarly high sulfur levels in refined petroleum products. Regulation of sulfur content is required because the compound is highly toxic, corrosive in nature, and has an ecological impact. The main two acid rain causes are the sulfur release and the nitrogen oxides.

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Thus, for avoiding such unsafe ecological consequences, desulfurization processes have been used widely [3, 4]. Although it has lesser applications in crude oil, catalytic hydrodesulfurization (HDS) is the most widely practiced desulfurization process in refinery operations for petroleum fractions. HDS is effective for the removal of active sulfur species like mercaptans and hydrogen sulfide, but not for disulfides. Sulfur disulfides are resistant to HDS processing and are transported through in the treated crude due to their low reactivity and high thermal stability [5, 6]. High temperatures (300–350°C) and pressures (15–90 bar) are utilized for the HDS process to operate at its best. It also contains costly cobalt-molybdenum catalysts and a substantial amount of hydrogen [7–10]. All these disadvantages have encouraged scientists to search for cheaper and efficient alternative methods for the desulfurization of fuels like oxidative desulfurization, adsorptive desulfurization, extraction desulfurization, alkylation desulfurization, bio-desulfurization, radiation desulfurization, and caustic desulfurization using sodium hydroxide (NaOH) [11–14]. Sodium hydroxide is a strong base that still remains a good option for desulfurization due to its simplicity, low price, and effectiveness in removing a high amount of sulfur compounds from sour crude oil [15].

In 1993, Ahonen [16] employed an electrolysis cell to oxidize and remove sulfur from heavy oil fractions blended with a sodium hydroxide solution. This process resulted in the conversion of most sulfur compounds into sulfites or sulfates. K. Jeyajothi [17] evaluated the effectiveness of sulfide removal from crude oil using the caustic washing method with varying NaOH concentrations. The desulfurization efficiency exhibited a notable reduction in H₂S content, achieving 50% efficiency at a concentration of 0.2 N NaOH. Shakirullah et al. [10] investigated the application of sodium hydroxide for the desulfurization of various petroleum products, including diesel oil, heavy residue, kerosene, and commercial furnace oil, and observed that the desulfurization efficiency varied among products. Specifically, kerosene exhibited an efficiency of 60%, diesel oil 68%, heavy residue 70%, and commercial furnace oil 71%. Al-Khodir and Albayati [15] utilized sodium hydroxide to remove sulfur from high-sulfur crude oil containing 5.8% sulfur by weight. Following theoretical optimization and experimental evaluation, a desulfurization efficiency of 56.89% was achieved. Kumari and Sengupta [18] employed ultrasound oxidation to investigate the simultaneous denitrogenation and desulfurization of model fuel and found that ultrasound improved oxidation efficiency and enhanced the conversion of sulfur-based compounds up to 93.7%.

Ultrasound consists of pressure waves that pulsate at extremely higher frequencies than sound waves that the human ear can hear, which ranges between 20 and 20,000 Hz [19]. A gas, liquid, or solid medium can produce waves when it undergoes a shock or vibratory motion displacement of a particle or a number of particles which can be

either singular or repetitively triggered. A remarkably high, linear power density of sound waves leads to a phenomenon known as ultrasonic waves. This phenomenon involves the formation and “violent” collapse of gas bubbles which subsequently creates shock waves that speed up chemical reactions and increase their effectiveness. The fleeting collapse of gas bubbles produce high fluid waves and the pressure and temperature raised to 1000 atm and 5000 K, respectively, over the points of explosions. The very high, center local conditions in an explosive manner foster rapid chemical reactions of the system by forming active intermediates in the micro environment [20].

Different strategies for desulfurization have had some success, but have also struggled with efficiencies targeting complex sulfur species and high energy consumption. This study focuses on ultrasound technology to overcome these hurdles by integrating ultrasound with caustic desulfurization. Ultrasound-assisted desulfurization (UADS) is able to use acoustic ‘cavitation’ to promote and enhance sulfur compound breakdown, and improve mass transfer over time. The combination of ultrasound and caustic soda is expected to improve the rate of desulfurization, while also reducing temperature and pressure operational conditions. This results in lowering energy demands and improving process efficiencies. The resultant new method is also expected to process the material in less time, leading to higher throughputs and productivity. UADS in particular helps target complex sulfur compounds, something traditional methods find very difficult. Also, the use of ultrasound in UADS over the traditional methods helps ensure compliance with international agreements by reducing sulfur emission, and supporting cleaner fuel production with less overall desulfurization energy consumption.

This work aims to confirm the effectiveness of UADS in reducing the sulphur content of crude oil to below 50 ppm using NaOH, as well as to assess the effect of fundamental UAD processing parameters—NaOH concentration, temperature, time, and ultrasound power—on the process. By addressing these parameters, this research aims to close existing gaps in the desulfurization process and establish a scalable, energy-efficient solution for producing cleaner fuels.

MATERIALS AND METHODS

Materials

In all experiments, crude oil samples were obtained from the East Baghdad Oil Field in Iraq. The sodium hydroxide (NaOH) used had a purity of 95%. For washing the glassware, 99.5% pure toluene (C₆H₅CH₃) was employed. All chemicals used were commercial laboratory supplies purchased from local suppliers and were utilized without prior preparation or purification. The physical parameters of the sample are detailed in Table 1.

Table 1. Properties of East Baghdad Crude Oil

Property	Value	Unit
API gravity	21.8	-
Specific gravity SG	0.923	-(at 15.56 °C)
Kinematic viscosity	78.731	cSt (at 26.7 °C)
Sulfur content	4.31	wt. %
H ₂ S content	4.36	ppm
Sediment content	0.0126	vol. %
Micro carbon residue	9.947	wt. %
Asphalting	6.412	wt. %
Salt content	21	ppm
Water content	1.60	wt. %

Equipment

The ultrasonic experiments were conducted using a Digital Ultrasonic Cleaner, a flexible and powerful ultrasonic liquid handler from Guangzhou CinTuo Technology Co., Ltd., China. The program-controlled microprocessor-based equipment was utilized at 800 watts and 40 kHz with 10 to 30 minutes of time settings and an 80 °C maximum temperature. For routine non-ultrasonic tests, the work was done with the help of a 2000 rpm mechanical stirrer to obtain effective mixing. The experiments were carried out with a 150-mL Pyrex Griffin beaker. Upon ultrasonic and non-ultrasonic treatment, a 100-mL cone-type glass separating funnel was used in the solvent

extraction process. The total sulfur content of crude oil was determined by RX-630SA (Cannon Instrument Company, USA), which uses energy-dispersive X-ray fluorescence (EDXRF) according to ASTM D4294-03 and ISO 8754 standards. This method is renowned for being quick, not destructive, cost-effective, and accurate.

Experiments

To perform the desulfurization process, 25 mL of crude oil is first placed in a beaker, and 15 mL of NaOH solution with varying concentrations (ranging from 1M to 4M) is added. NaOH powder is dissolved in distilled water to prepare the NaOH solution. The contents of the beaker are sonicated with continuous stirring to facilitate good mixing. The beaker is then immersed in distilled water and loaded into an ultrasonic processor. Sonication is activated with an ultrasonic generator whose control is preset at the desired reaction temperature. After the pre-programmed sonication time has elapsed, the ultrasonic generator is turned off, and the reaction mixture is transferred into a separator funnel. The funnel is hand-shaken for 2 minutes to allow phase separation and left standing for 10 minutes to separate the organic phase from the aqueous phase distinctly. A small sample from the organic phase is collected to test the sulfur concentration. Detailed experimental setup for the traditional NaOH desulfurization process can be found in previous research [15]. the same procedure is followed in non-ultrasonic experiments, with the ultrasonic apparatus being replaced by the mechanical stirrer. The experimental process for UADS is illustrated in Figure 1, while Table 2 details the experimental conditions and procedures..

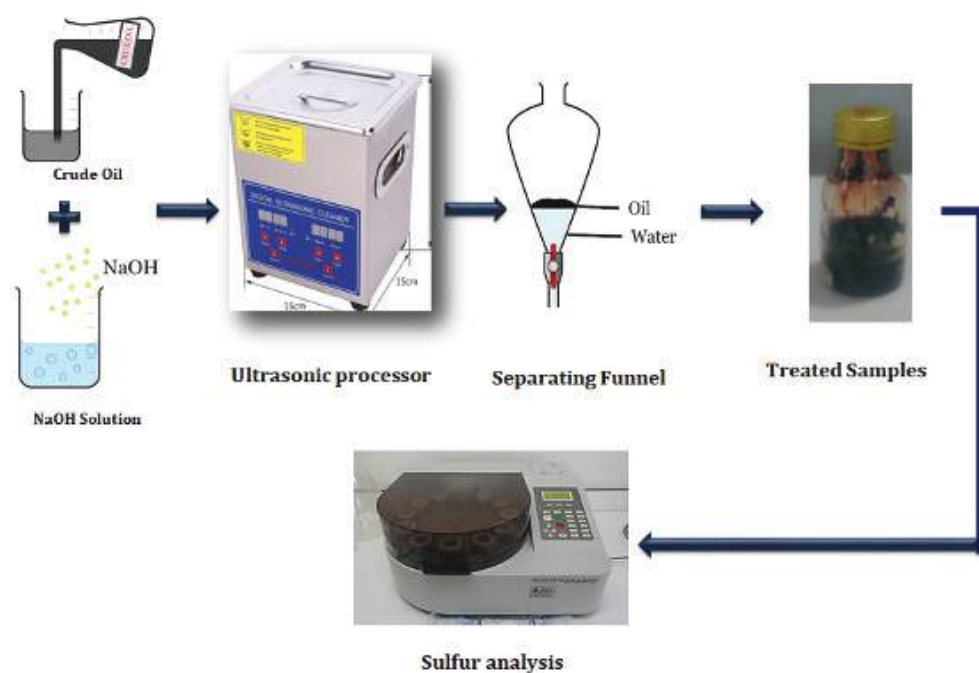
**Figure 1.** The UADS experimental setup.

Table 2. Specifications of experimental operations

Exp. Number	Parameter	Variable	Operating Conditions	Ultrasonic Tests	Non- Ultrasonic Tests
1		1			
2		1.5			
3		2			
4	Molarity of NaOH	2.5	time = 30 min; temp = 60 °C	frequency = 40KHz	Speed = 2000 rpm
5		3			
6		3.5			
7		4			
8		20			
9	Temperature in Celsius	40	time = 10 min; NaOH = 3M	frequency = 40KHz	Speed = 2000 rpm
10		60			
11		80			
12		20			
13	Temperature in Celsius	40	time = 20 min; NaOH = 3M	frequency = 40KHz	Speed = 2000 rpm
14		60			
15		80			
16		20			
17	Temperature in Celsius	40	time = 25 min; NaOH = 3M	frequency = 40KHz	Speed = 2000 rpm
18		60			
19		80			
20		20			
21	Temperature in Celsius	40	time = 30 min; NaOH = 3M	frequency = 40KHz	Speed = 2000 rpm
22		60			
23		80			
24		30			
25	Ultrasound amplitude %	40	time= 30 min; temp = 60 °C;		
26		50	NaOH = 3M		
27		60			

Sulfur Removal Efficiency (η %)

Desulfurization efficiency is calculated using the following equation: it is the ratio of the sulfur removed to the sulfur initially present in the crude oil. The desulfurization efficiency (η) is determined using the equation:

$$\eta (\%) = \left(\frac{C^o - C}{C^o} \right) \times 100 \quad (1)$$

Where C^o is the original sulfur concentration (ppm), and C is the sulfur concentration in the treated sample (ppm).

RESULTS AND DISCUSSION**Comparison between Ultrasonic and Conventional Desulfurization using NaOH**

The study investigates the comparative effect of ultrasonic-assisted desulfurization and traditional NaOH-based

desulfurization procedures, respectively, through performance comparisons at different NaOH concentrations, temperatures, and reaction times. Figures 2, 3, and 4 indicate that ultrasonic desulfurization tends to outdo plain NaOH treatment alone in rates of sulfur elimination. Internal benefit of ultrasonic treatment is the physical phenomenon of cavitation, where the high-frequency sound waves produce tiny bubbles that collapse at a rapid rate. The collapse creates infinitesimal areas of high pressure and temperature, and this increases the rate of desulfurization tremendously. These intensification effects of ultrasound, including enhanced mass transfer and reaction kinetics, are consistent with findings reported in other chemical processes [21]. It does so by speeding up the reaction between sulfur in the crude oil and sodium hydroxide (NaOH) and enhancing mass transfer. Ultrasonic waves provide the necessary activation energy sulfur removal and is at a much lower energy barrier, hence making the process much faster and more efficient than with normal techniques.

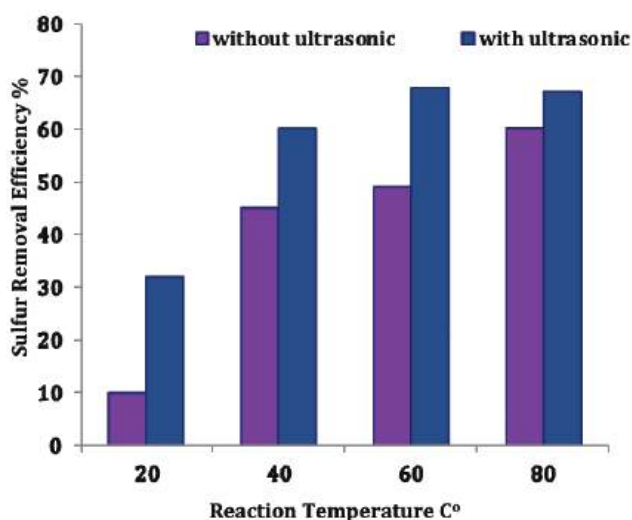


Figure 2. Ultrasonic desulfurization versus traditional NaOH desulfurization at different temperatures, 30 minutes, and 3M NaOH concentration.

Additionally, the mechanical action of cavitation that also decomposes sulfur compounds increases the active surface area of the NaOH-sulfur reaction and enables more efficient interaction of NaOH with sulfur [19]. These synergistic effects—fast reaction kinetics, mass transfer optimum, and accessibility of sulfur sites—lead to the greater result of desulfurization by ultrasound.

Figure 2 shows the effect of temperature on desulfurization efficiency with and without ultrasonic waves. The most striking enhancements are found at low to mild

temperatures (20°C to 60°C). With 20°C, ultrasonic support significantly enhances efficiency with a considerable performance compared to traditional methods. With 60°C, the rate of sulfur elimination is optimal at around 70%, with the best condition for ultrasonic desulfurization. However, at a temperature increase to 80°C, the advantages of ultrasonic assistance diminish, suggesting that at some sort of thermal ceiling, there is little to be gained by further energy from the ultrasonic source. The pattern is such that synergism between ultrasound and temperature is best at lower temperatures and that this is likely due to more extensive cavitation at these conditions, which causes maximum mass transfer and reaction rate.

Figure 3 illustrates the effect of reaction time on desulfurization efficiency. Without the use of ultrasonic assistance, sulfur removal increases incrementally from 10% efficiency after 10 minutes and eventually 50% after 30 minutes. With the use of ultrasonic assistance, the desulfurization process is significantly improved. For the first 10 minutes, the efficiency of sulfur removal is already at about 30%, while it is about 70% at 30 minutes. This rapid jump in efficiency upon ultrasonic treatment indicates the enhanced reaction kinetics with cavitation. Cavitation effect will improve the mixing and mass transfer so that NaOH will be more reactive with sulfur compounds within a shorter time frame and thus achieve maximum desulfurization within a shorter time period.

Figure 4 shows the effect of NaOH molarity on sulfur removal efficiency with or without the assistance of ultrasonic. As seen from the curve, as there is a transition in the quantity of NaOH from 1M to 3M, there is therefore an increase in sulfur removal efficiency. It is due to the fact that greater NaOH will be present to react with sulfur

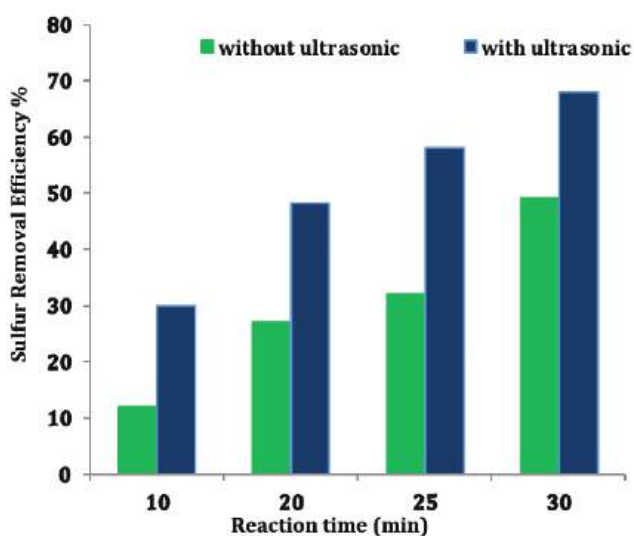


Figure 3. Comparison of ultrasonic desulfurization versus traditional NaOH desulfurization at different time intervals, operating at a temperature of 60°C with a NaOH concentration of 3M.

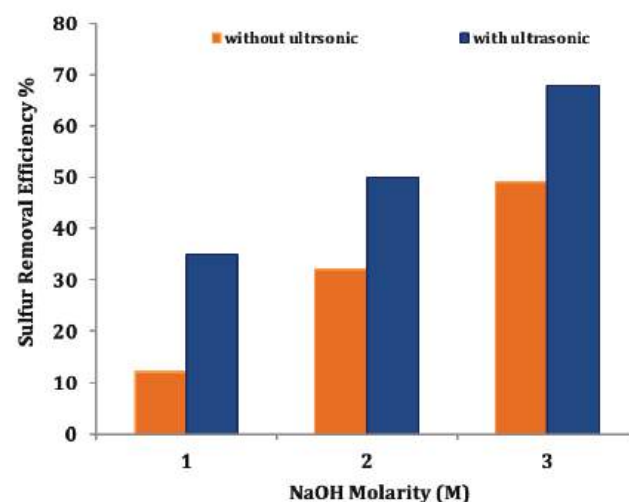


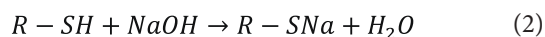
Figure 4. Comparison of ultrasonic desulfurization versus traditional NaOH desulfurization at various NaOH molar concentrations, operating at a temperature of 60°C and a processing time of 30 minutes.

compounds. But by the addition of ultrasonic assistance, the impact is enhanced in all molarities, with optimal improvement at the maximum quantity of 3M, where efficiency is achieved up to 70%.

The results show not only that ultrasonic waves enhance NaOH reactivity but also optimize the utilization of higher concentrations of NaOH. This performance is likely a result of the ultrasonic waves inducing greater mixing and greater mass transfer rates, overcoming the limitations typically imposed by diffusion on typical desulfurization processes. In this regard, ultrasonic assistance becomes more active at higher NaOH concentrations and thus maximizes the opportunity for desulfurization.

Influence of NaOH Concentrations on Ultrasonic Desulfurization

Different levels of NaOH at 60°C were applied, and there was sonication under 40 kHz for 30 minutes. Figure 5 demonstrates that there was a decrease of sulfur content in the crude oil when there are higher concentrations of NaOH, indicating that the highest removal of sulfur takes place at high levels of NaOH. This is likely an outcome of the acidic components of some of the sulfur compounds, such as mercaptans, which transfer with ease from the oily phase into the alkaline sodium hydroxide layer where they are oxidized to sulfides. Since the aqueous solution of sodium hydroxide reacts with the crude oil [22], it joins forces with inorganic sulfur compounds to create salts that are eventually separated out using a separating funnel. This results in desulfurized crude oil. Equation (2) explains the reaction:



Here R-SH is the thiol (mercaptan) group, and R-SNa is the sodium salt that gets produced in the course of the reaction. This conversion allows sulfur to be removed from the oil phase because the sodium salt (R-SNa) is more soluble

in the aqueous NaOH layer and can be separated from the crude oil.

This observation is in line with previous research, where increasing the concentration of sodium hydroxide solution improves the efficiency of the desulfurization reaction [23]. Further increased concentration of NaOH to over 3 M, for instance, to 4 M, results in no better than a slight increase in efficiency of desulfurization. As the process employs a difficult-to-regenerate solvent, wasteful consumption of NaOH would not be economical. Thus, the best concentration was revealed to be a 3 M NaOH solution, which extracted sulfur with an efficiency of approximately 67.8%.

Influence of Ultrasound-Assisted Desulfurization Temperature and Time to be effect

The literature review further identifies that temperature is crucial in influencing the properties of wave propagation in sonication, and as temperatures rise there are higher sulfur conversion [24, 25]. Elevated temperatures enhance the rate of mass transfer diffusion and accelerate achieving equilibrium [26]. However, since the NaOH-sulfur compounds reaction process is exothermic, external heat is not required. Though high temperatures can accelerate the rate of reaction, extremely high temperatures may lead to negative consequences, such as desulfurization or thermal cracking of crude oil. The Figure 6 indicates the efficiency of removing sulfur (%) with temperature (°C) at different reaction times (10, 20, 25, and 30 minutes).

The first one is general trend since with the increase in temperature from 10°C to 80°C, the efficiency of sulfur removal always improves at all the reaction times, which shows that there is a positive correlation between the desulfurization efficiency and temperature. The second one is effect of reaction time for shorter reaction times (10 minutes), the efficiency of sulfur removal is very low, starting from around 10% at 20°C and only reaching around 30% at

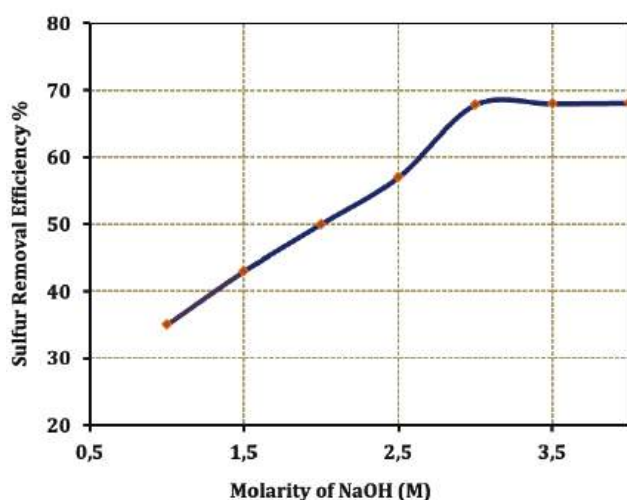


Figure 5. Influence of sodium hydroxide concentrations.

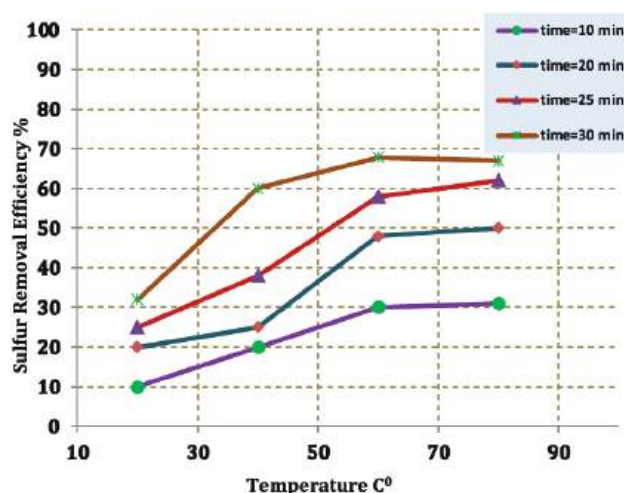


Figure 6. Influence of Ultrasound-Assisted Desulfurization Temperature and Time.

80°C. Because the reaction time is increased to 20, 25, and 30 minutes, the sulfur removal efficiency is significantly improved, especially within the range of temperatures 30°C to 60°C, with the 30-minute reaction being the most efficient at all temperatures. The optimal temperature range the sulfur removal efficiency seems to reach its peak at about 60°C for all the reaction times, especially for the 25-minute and 30-minute ranges, where it is approximately 70% and 80%, respectively. After 60°C, the improvement in efficiency plateaus or drops slightly, indicating that temperatures higher than this are not beneficial to a greater degree.

Influence of Ultra-Sonation Intensity

Since the amplitude of ultrasonic waves directly correlates with the intensity of the desulfurization process, the effect of various amplitudes (30%, 40%, 50%, and 60%) was thoroughly investigated at a fixed duration of 30 minutes and a steady temperature of 60°C. As depicted in Figure (7), the sulfur removal efficiency showed a noticeable improvement, climbing from 58% to an impressive 68% when the amplitude was increased from 30% to 40%. This striking enhancement aligns with the findings of Duarte et al., who emphasized that boosting ultrasonic intensity significantly enhances cavitation. This process generates more free radicals, activates oxygen atoms to a greater degree, and dramatically improves the overall sulfur removal efficiency [27]. Nevertheless, when the amplitude exceeded 40%, there was a slight but detectable decrease in desulfurization efficiency. This slight decrease can be explained by high-intensity ultrasound waves causing excessive bubble agitation, a phenomenon also reported in other ultrasound-assisted desulfurization studies [28]. These high waves cause excessive vibration of the bubbles during the negative acoustic phase, and thus, the bubbles fail to have enough time to collapse effectively in the positive acoustic phase [29]. Therefore, while initially increasing capacity boosts performance, exceeding 40% leads to diminishing

returns. We can consider a 40% amplitude to strike a perfect balance. This value provides just the right intensity to achieve maximum efficiency without compromising the microbubble dynamics.

CONCLUSION

In this work, the effect of combining ultrasound with sodium hydroxide treatment for crude oil desulfurization was examined. The experiments showed that ultrasound made a clear difference. The highest sulfur removal, about 68%, was achieved when a 3 M NaOH solution was used at 60 °C for 30 minutes with 40% ultrasound power. This improvement compared with the 49% obtained without ultrasound indicates that sound waves helped the caustic solution reach and react with more sulfur compounds. Even though the results were encouraging, the final sulfur content was still well above the 50 ppm level required for ultra-low-sulfur fuels. This means that the process, in its current form, is not yet sufficient for industrial use. Reaching that target will probably require extra steps, such as integrating catalytic oxidative desulfurization methods, for instance using efficient catalysts like supported ceria [30], or further tuning of the operating conditions. Still, the method shows good potential. It works under relatively mild conditions, needs less energy and fewer chemicals, and could help lower sulfur-related emissions. Future studies should test different combinations and perhaps link ultrasound with other techniques to push the sulfur level even lower and make the process more practical for cleaner-fuel production.

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.

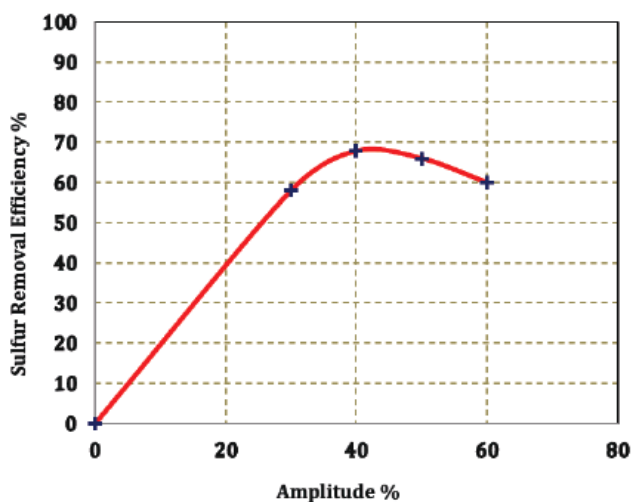


Figure 7. Influence of ultra-sonication intensity.

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Research Article

Study the influence of thermophoretic effect buoyancy force in prandtl fluid flow on a stretching sheet in presence of joule heating and viscous dissipation

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ABSTRACT

There are many natural phenomena and engineering usage susceptible to magneto fluid dynamics. The boundary layer theory is precisely such as asymptotic theory for the ease of very large Reynolds number. A numerical investigation was made to analyze the effects of Brownian motion, thermophoresis, and buoyancy force of non-Newtonian nanofluid relying on the Prandtl fluid model over a stretching sheet. The energy equation is empowered by the influence of Joule heating, viscous dissipation, and thermal radiation, whereas the concentration equation is determined by a chemical reaction. Similarity transformations are used to convert the governing partial differential equations (PDEs) into a system of nonlinear ordinary differential equations (ODEs). The resulting systems are effectively solved using the numerical scheme known as the Runge-Kutta-Fehlberg's method with shooting technique and bvp4c solver in MATLAB code. The graphical results show the effect of pertinent parameters under buoyancy-assisting/opposing forces on velocity, temperature, and concentration profiles. Momentum transport gets accelerated for higher material/characteristic parameter of the non-Newtonian fluid model irrespective of presence or absence of porous matrix but thermal energy gets depleted for higher material parameter across the flow domain contributing to thermal stability. The elastic property of the Prandtl fluid and the effect of electromagnetic force increase slightly the solutal concentration but significantly to temperature. Thus, it is concluded that the generated Lorentz force (the additional body force) increases the thermal transport and solutal concentration of non-Newtonian nanofluid model.

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INTRODUCTION

Generally, metallic oxides, metals, or carbon nanotubes nanoparticles etc are mixed with a base fluid, like ethylene glycol, water, or oil, to produce nanofluid. They exhibit an improved thermal, optical, and electrical properties. Choi [1] initially proposed the idea of a nanofluid, demonstrating how adding nanoparticles to a fluid can dramatically increase its thermal conductivity. The boundary layer flow, past a stretched plate in a viscous fluid, is studied by Siddappa and Abel [2]. The heat transmission across a stretched sheet with varying wall temperature and internal heat generation of an electrically conducting fluid are examined by Vajravelu et al. [3]. Wang et al. [4] focused on the fluid flow and heat transmission properties of nanofluids in both forced and free convection flows. The inadequacy of macroscopic analysis on heat transfer in nanofluids is examined by Keblinski et al. [5]. They also provide explanations with experiments for the fluids with higher thermal conductivity. Furthermore, a number of researchers [6–10] investigated the different aspects of viscous nanofluid flows across a stretching surface due to their distinct and durable qualities. The nanofluids applications in a variety of industries, including manufacturing, microelectronics, thermal power, and biomedicines.

Non-Newtonian fluids are used in many technological processes as well as house hold usage including toothpaste, paints, emulsions, oil, and polymer sheets. The class of non-Newtonian fluids is divided into many branches. The shear thinning fluid is one of the most significant branch. In these fluids, the apparent viscosity decreases with applied shear force. Many fluid models are proposed to explain the characteristics of shear thinning fluids. The Prandtl fluid model is one which can adequately explain the shear thinning phenomenon. Kuik [11] worked on Prandtl's solution on examining the flow at infinity. The flow around an exponential spiral with a finite length, is first studied in order to introduce a length scale. Prandtl's equations are recovered in the limit for infinite spiral length. Khan et al. [12] discovered the effects of homogenous/ heterogeneous reactions on 2-D flow of Prandtl fluid over a stretching sheet. Ganesh Kumar et al. [13] considered Joule heating, to study the melting effects, on flow of Prandtl fluid over an expanded sheet and found that velocity profiles are directly proportional to the Prandtl parameter. Eid et al. [14] studied the 3-D flow and heat transfer of Prandtl nanofluid over a convectively heated porous sheet in the presence of non-linear radiation and higher order chemical reaction. Asad [15] examined entropy generation on the mixed convective MHD flow of Prandtl fluid past a non-linear stretched sheet. Zafar et al. [16] investigated the irreversibility flow of Prandtl nanofluid over a stretched surface in Darcy-Forchheimer medium.

The buoyant force produced by temperature changes within a flow domain is referred to as thermal buoyancy. The interplay of heat and mass transfer along with

buoyancy effects is crucial in a wide range of practical applications to enhance the efficiency and effectiveness of various thermal and chemical processes like solar collectors, nuclear reactor coolants, chemical process industries, environmental engineering, etc. Raju and Sandeep [17] considered the assisting/opposing flow of non-Newtonian fluid past a cone with Lorentz force. Habib et al. [18] studied the importance of Stephen blowing and Lorentz force on dynamics of Prandtl nanofluid using Keller box method. Further, Yaseen et al. [19] considered hybrid nanofluid flow past a permeable moving surface. Ramanjini et al. [20] examined the effects of first-order slip, and chemical reaction on unsteady Prandtl nanofluid flow through stretching sheet due to mixed convection. Recently, many researchers [21–33] explained the importance of flow of nanofluid and hybrid nanofluid using different flow geometry.

The following salient features are embodied in the present analysis:

- Impacts of Joule heating, thermal radiation, chemical reaction, and viscous dissipation on the flow.
- The contributions of thermophoresis and Brownian motion on the heat transfer.
- Buoyancy assisting/opposing flow of nanofluid.
- The solution of mathematical model by Runge-Kutta-Fehlberg's scheme with bvp4c solver in MATLAB code.

Research Questions

The following scientific research issues are addressed in this study:

- The effects of Lorentz force, fluid parameter, elastic parameter and porosity parameter on the velocity profile.
- The transport of heat in presence of Joule heating, viscous dissipation and Brownian motion.
- The effect of chemical reaction parameter, Schmidt number, thermophoresis, and Brownian motion on nanoparticle concentration.
- Variations of skin friction coefficient, Nusselt number, and Sherwood number, the surface criteria, due to change in fluid parameter, elastic parameter, Eckert number and Biot number.

Mathematical Formulation

Consider a two-dimensional, steady and incompressible flow of nanofluid on a stretching sheet as illustrated in Figure 1. Impacts of thermophoresis and Brownian motion on the flow are taken into account. Further, Joule heating, thermal radiation and viscous dissipation empower the energy equation. The x -axis is taken along the stretching sheet and the y -axis is vertical to the flow direction. The sheet is stretched with the velocity $u_w(x) = ax$ with $a > 0$, a constant. The temperature and concentration of the stretching surface are taken as T_w and C_w while the ambient temperature and concentration are at T_∞ and C_∞ .

The following assumptions are made:

- The nanofluid is considered to be non-Newtonian and incompressible.
- Both the nanoparticles and fluid phase are in thermal equilibrium.
- The induced magnetic field is so small (low magnetic Reynolds number) and hence it is ignored.
- The constant uniform transverse magnetic field of strength B_0 is applied to the sheet.

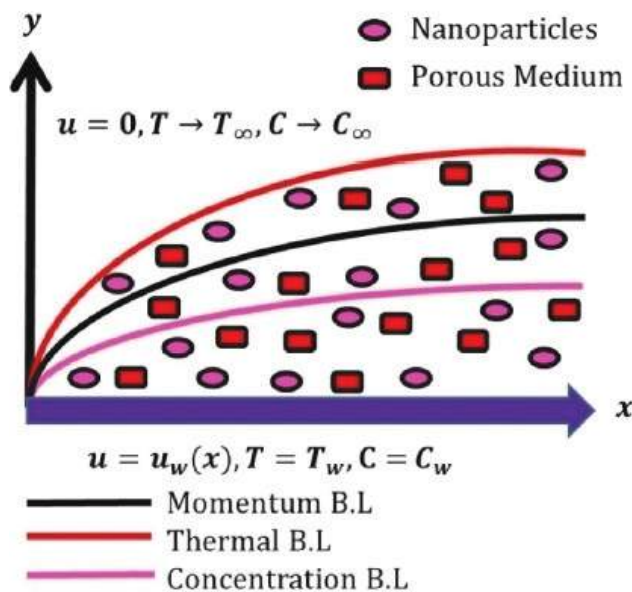


Figure 1. Physical model of the problem.

The Cauchy stress tensor for the Prandtl fluid is given by

$$S = \frac{A \sin^{-1} \left\{ \frac{1}{c_1} \left[\left(\frac{\partial u_i}{\partial x_j} \right)^2 + \left(\frac{\partial u_j}{\partial x_i} \right)^2 \right]^{\frac{1}{2}} \right\}}{\left[\left(\frac{\partial u_i}{\partial x_j} \right)^2 + \left(\frac{\partial u_j}{\partial x_i} \right)^2 \right]^{\frac{1}{2}}} A_1 \quad (1)$$

where A and c_1 are the material parameters and A_1 is the first Rivlin-Erickson tensor.

Under the above assumptions, the governing equations and corresponding boundary conditions are given by [15, 20]

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0, \quad (2)$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = \nu \frac{A}{c_1} \left\{ 1 + \frac{1}{2c_1^2} \left(\frac{\partial u}{\partial y} \right)^2 \right\} \frac{\partial^2 u}{\partial y^2} - \frac{\sigma B_0^2}{\rho} u - \frac{\nu}{k^*} u + g \beta_1 (T - T_\infty), \quad (3)$$

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \left(\xi + \frac{16\sigma^* T_\infty^3}{3\rho c_p k^*} \right) \frac{\partial^2 T}{\partial y^2} + \frac{\sigma B_0}{\rho c_p} u^2 + \frac{(\rho c)_p}{(\rho c)_f} \left\{ D_B \frac{\partial C}{\partial y} \frac{\partial T}{\partial y} + \frac{D_T}{T_\infty} \left(\frac{\partial T}{\partial y} \right)^2 \right\} + \frac{\mu}{\rho c_p} \left(\frac{\partial u}{\partial y} \right)^2, \quad (4)$$

$$u \frac{\partial C}{\partial x} + v \frac{\partial C}{\partial y} = D_B \frac{\partial^2 C}{\partial y^2} + \frac{D_T}{T_\infty} \frac{\partial^2 T}{\partial y^2} - k_c (C - C_\infty). \quad (5)$$

Boundary conditions

$$\begin{aligned} \text{at } y = 0: & u = ax, v = 0, -k \frac{\partial T}{\partial y} = h(T_w - T), C = C_w, \\ \text{as } y \rightarrow \infty: & u \rightarrow 0, T \rightarrow T_\infty, C \rightarrow C_\infty. \end{aligned} \quad (6)$$

Similarity transformations

The boundary layer equations (2) – (5) admit of a similarity solutions

$$u = axf'(\eta), v = -\sqrt{av}f, \theta(\eta) = \frac{T - T_\infty}{T_w - T_\infty}, \quad (7)$$

$$\phi(\eta) = \frac{C - C_\infty}{C_w - C_\infty}.$$

where $\eta = \sqrt{\frac{a}{\nu}}y$ is the non-dimensional vertical distance measured from the surface of the stretching sheet.

Now the equations (2) – (6) become

$$\alpha(1 + \beta f''^2) f'''' + ff'''' - f'^2 - Mf' - k_p f' + \lambda \theta = 0 \quad (8)$$

where $f'(\eta)$ and $f(\eta)$ represent horizontal and vertical velocity components respectively.

$$\begin{aligned} \frac{1}{Pr} \left(1 + \frac{4}{3} R \right) \theta'' + f\theta' + Nb\theta'\phi' + Nt\theta'^2 \\ - MEcf'^2 + Ecf''^2 = 0 \end{aligned} \quad (9)$$

$$\phi'' + \frac{Nt}{Nb} \theta'' + Scf\phi' - k_r Sc\phi = 0 \quad (10)$$

$$\begin{aligned} f(0) = 0, f'(0) = 1, \theta'(0) = -\gamma[1 - \theta(0)], \phi(0) = 1, \\ f'(\infty) = 0, \theta(\infty) = 0, \phi(\infty) = 0. \end{aligned} \quad (11)$$

Table 1. Comparison of skin friction coefficient $f''(0)$ for different values of M

M	Hibab et al. [18]	Ramanjini et al. [20]	Present study
0	-1.8428	-1.8478	-1.8465
1	-1.0100	-1.0012	-1.0126
5	-0.5213	-0.5205	-0.5218
10	-0.4150	-0.4156	-0.4162

Physical parameters of engineering interest

The physical parameters such as the skin friction coefficient, local Nusselt number, and local Sherwood number are respectively defined as

$$C_f = \frac{2\tau_w}{\rho u_w^2}, Nu_x = \frac{xq_w}{k(T_w - T_\infty)}, \text{ and } Sh_x = \frac{xq_m}{D_B(C_w - C_\infty)} \quad (12)$$

$$\text{Here, } \tau_w = \mu \left(\frac{\partial u}{\partial y} \right)_{y=0} \quad (\text{shear stress}),$$

$$q_w = -k \left(\frac{\partial T}{\partial y} \right)_{y=0} + (q_r)_{y=0} \quad (\text{heat flux}), \text{ and}$$

$$q_m = -D_B \left(\frac{\partial C}{\partial y} \right)_{y=0} \quad (\text{mass flux}).$$

$$\text{Finally, we get } \sqrt{\text{Re}_x} C_f = f''(0), \quad (13)$$

$$\sqrt{\text{Re}_x} Nu_x = - \left(1 + \frac{4}{3} R \right) \theta'(0), \sqrt{\text{Re}_x} Sh_x = -\phi'(0)$$

Work Validation

The system of non-linear coupled ODEs with prescribed boundary conditions (8-11) are solved numerically by Runge-Kutta Fehlberg's method combined with Shooting technique using bvp4c solver in MATLAB code. The convergence, stability and accuracy of the method has been carried out with step size 10^{-2} and error tolerance 10^{-5} . The computation of the skin friction coefficient has been made when $(\alpha = \beta = 0.2, Pr = 1, Ec = k_p = Sc = k_r = Nt = 0, Nb \rightarrow 0)$ for various values of M which are shown in Table 1. These outcomes are found in good agreement with the results reported by Hibab et al. [18] and Ramanjini et al. [20].

In this method, the governing equations are reduced to a set of following first order differential equations.

$$f_1' = f_2$$

$$f_2' = f_3$$

$$f_3' = - \left(\frac{1}{\alpha(1 + \beta f_3^2)} \right) [f_1 f_3 - f_2^2 - (M + k_p) f_2 + \lambda \theta]$$

$$\theta_1' = \theta_2$$

$$\theta_2' = - \frac{Pr}{\left(1 + \frac{4}{3} R \right)} (f_1 \theta_2 + Nb \theta_2 \phi_2 + Nt \theta_2^2 - MEc f_2 + Ec f_3^2)$$

$$\phi_1' = \phi_2$$

$$\phi_2' = - \left[\frac{Nt}{Nb} \theta_2' + Sc \{ f_1 \phi_2 - k_r \phi_1 \} \right]$$

subject to the initial conditions

$$f_1(0) = 0, f_2(0) = 1, \theta_1(0) = -\gamma [1 - \theta_1(0)], \phi_1(0) = 1$$

$$f_3(0) = r_1, \theta_2(0) = r_2, \phi_2(0) = r_3$$

where r_1, r_2 and r_3 are are guess values to be prescribed during computation. A self-corrective procedure has been applied to improve the accuracy.

RESULTS AND DISCUSSION

Figure 2 depicts the effect of material parameter (α) and porosity parameter (k_p) on the horizontal component of velocity distribution in the boundary layer, the domain of interest. The velocity decreases asymptotically to attain its ambient state throughout the flow field. Moreover, it increases with an increase in the value of Prandtl fluid parameter (α). It is interesting to note that when $\alpha = 0$, equation (8), the momentum transport equation, reduces to an electrically conducting Newtonian fluid with a transverse magnetic field and the order of differential equation is reduced by 1 i.e. from third order to second order which is the usual characteristics of non-Newtonian fluid. Further, it is seen that fluid velocity decreases in the presence of porous medium ($k_p = 0.5$), due to constricted passage and resistivity of porosity term which appears with a negative sign.

Figure 3 shows a fall in temperature distribution asymptotically for higher values of non-Newtonian/material parameter. Thus, it is concluded that the present non-Newtonian fluid model reduces the thermal power in the boundary layer for higher values of characteristic parameter irrespective of porosity of the medium. This is an important observation pertaining to the present fluid model. This may

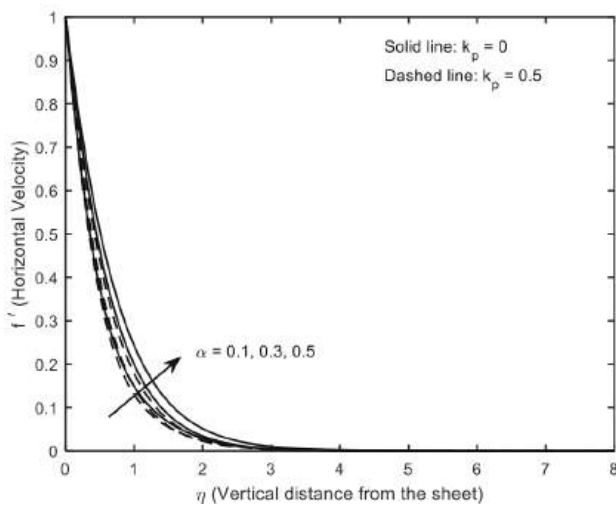


Figure 2. Impact of α and k_p on $f'(\eta)$.

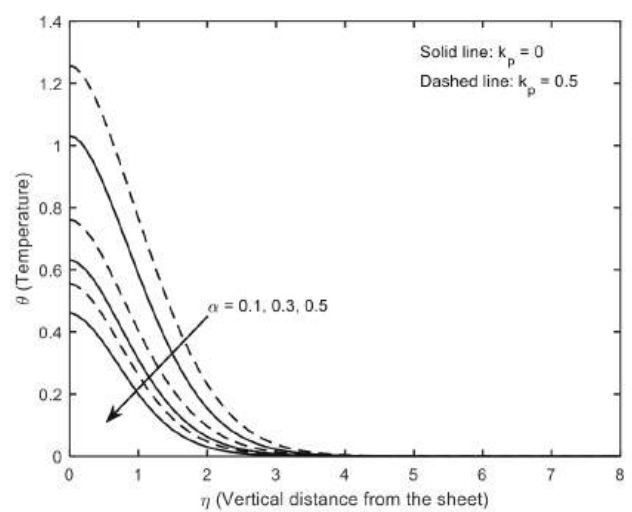


Figure 3. Impact of α and k_p on $\theta(\eta)$.

be of industrial use when asymptotic fall in thermal energy level is required for higher value of material parameter that is mostly required for thermal stability of the flow. From Figure 4 it is seen that the nanoparticle solutal concentration distribution decreases layer wise as well as vertically same as that of temperature distribution.

Figures. 5 and 6 show the fall of concentration and temperature across the flow field for higher elasticity fluid property. The effect of higher magnetic field intensity results in strengthen of both temperature and concentration level but the higher magnetic field intensity rises slightly concentration (Fig. 5) and significantly temperature distribution (Fig. 6). It is concluded that electromagnetic force affects significantly thermal property of the fluid but not to nanoparticle concentration.

Figure 7 depicts that an increase in elastic property of the fluid as well as electromagnetic body force. Higher magnetic field strength increases the velocity but opposite effect is observed in case of elastic parameter. This outcome finds application in an appropriate field of interest such as chemical processing unit where reducing the velocity distribution is essential by regulating voltage vis-à-vis magnetic field strength in an electric circuit and thermal power system.

Figure 8 depicts the effect of two important parameters namely thermal buoyancy and Biot number on solutal concentration. It is observed that an increase in thermal buoyancy decreases the solutal concentration but the reverse effect is observed in case of Biot number. As the thermal buoyancy assisting parameter $\lambda > 0$ increases that leads

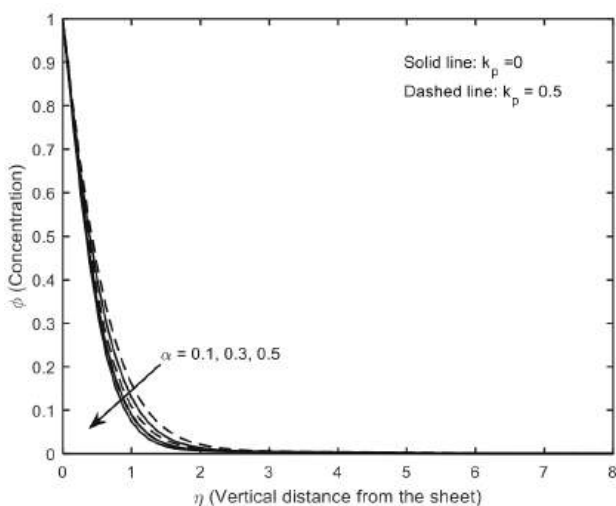


Figure 4. Impact of α and k_p on $\phi(\eta)$.

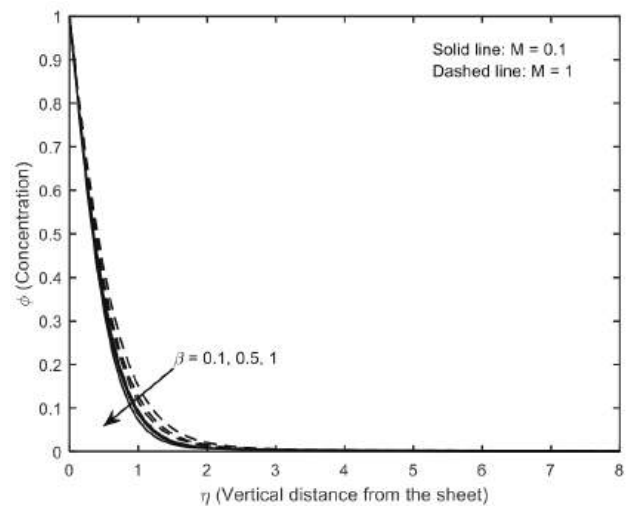


Figure 5. Impact of β and M on $\phi(\eta)$.

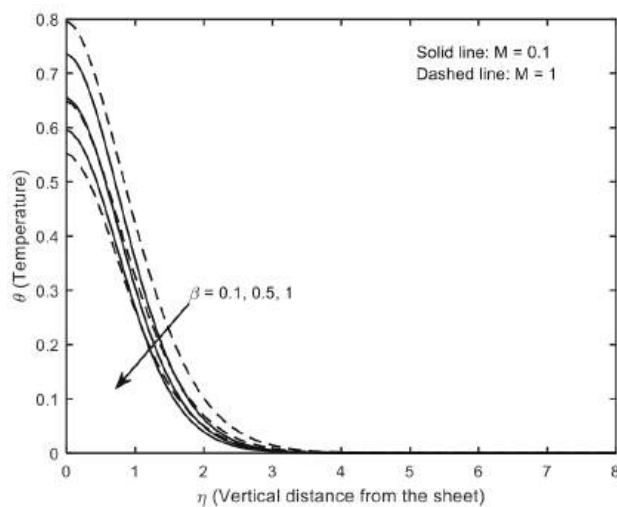


Figure 6. Impact of β and M on $\theta(\eta)$.

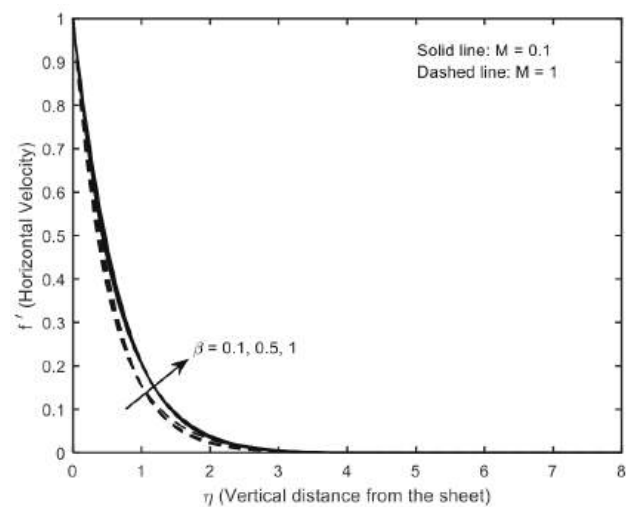


Figure 7. Impact of β and M on $f'(\eta)$.

to decrease the solutal concentration but in case of Biot number the solutal concentration increases. The results admit of a physical interpretation as: since $\gamma > 0$ relates to Newtonian cooling and Biot number being an index of the ratio of the heat transfer resistances inside the flow domain to that of bounding surface, the concentration increases in the flow domain.

Figure 9 shows that the temperature distribution admits of the same effect as that of concentration for both the parameters. The explanation is provided in case of Figure 8. Thus, higher Biot number enhances the fluid temperature resulting cooling of the bounding surface. It serves as a cooling device of the stretching surface resulting a desired quality product specifically in polymer processing. On careful observation of Figures. 8 and 9 it is remarked that in case of buoyancy assisting flow, the

volume fraction of nanoparticle i.e. concentration behave in same way as that of temperature variation in case of enhanced thermal buoyancy parameter (λ) i.e. down grading the concentration and temperature due to enhanced thermal buoyancy.

Figure 10 depicts a higher velocity distribution with $\lambda > 0$ i.e. in case of buoyancy assisting flow the velocity increases. The result coincides with physics of the buoyant force. The Biot number (γ) also enhances the fluid velocity but slightly. Figures. 11 and 12 show the effect of thermophoresis (Nt) and Brownian diffusion (Nb) on concentration and temperature distributions respectively. The higher thermophoresis enhances the solutal concentration but higher Brownian diffusion downgrades the level of concentration in the flow domain. The physical

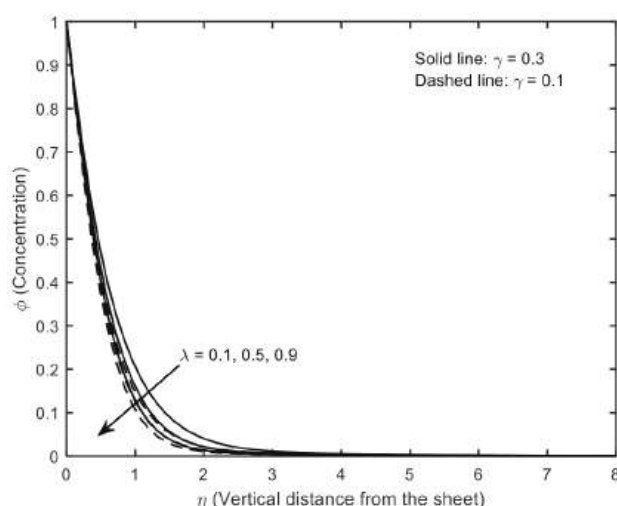


Figure 8. Impact of λ and γ on $\phi(\eta)$.

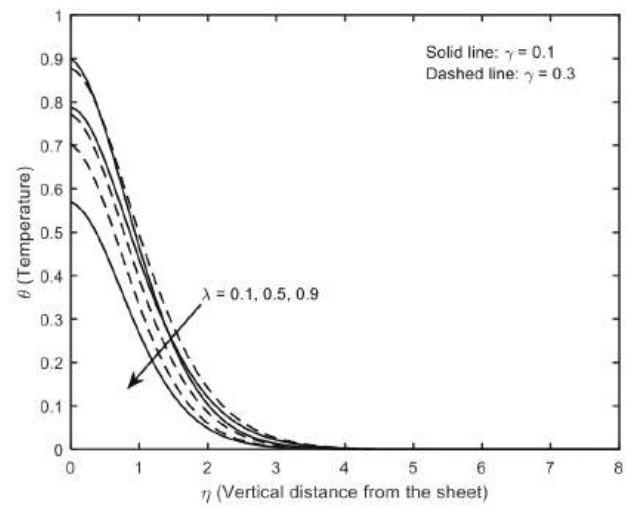


Figure 9. Impact of λ and γ on $\theta(\eta)$.

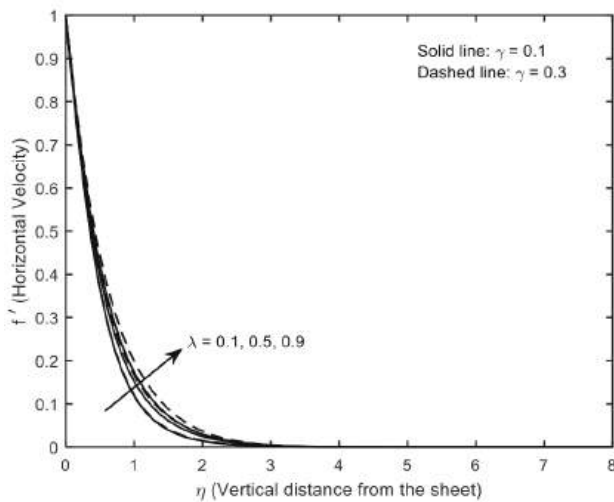


Figure 10. Impact of λ and γ on $f'(\eta)$.

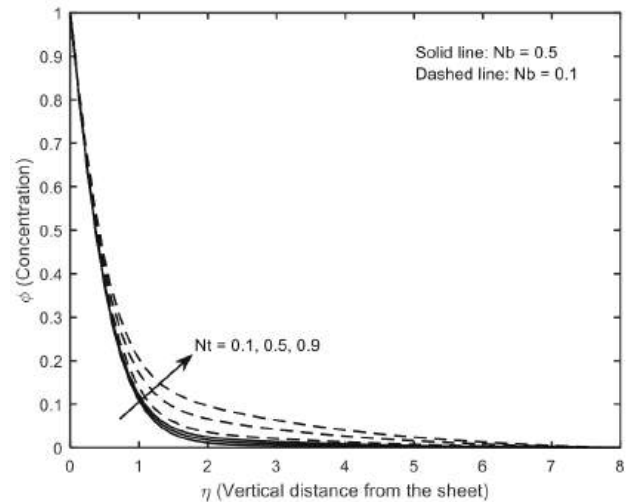


Figure 11. Impact of Nt and Nb on $\phi(\eta)$.

reasoning attributed as follows: thermo diffusion gives rise to the additional mass transfer because of the temperature gradient contributing to higher concentration level but Brownian motion reduces the level of concentration. Figure 12 shows that both thermophoresis and Brownian motion enhance temperature distribution in the flow domain. Figure 13 presents the effects of two important parameters such as Ec and R on temperature distribution. It is interesting to note that both the parameters enhance the temperature distribution. The Eckert number is a measure of the dissipation effects in the flow. Since this grows in proportion to the square of velocity, it can be neglected for small velocity. The observed finding is that temperature increases with higher values of Ec as well as radiation parameter (R). The effect of Ec is more distinct. Figure 14 depicts that enhancement of destructive chemical reaction

parameter ($k_r > 0$) and Schmidt number reduce the concentration level throughout the flow domain. On careful observation, it reveals that the heavier species Sc ($Sc = 5$) attends ambient state of concentration faster than that of lighter one. This is due to inherent property of the heavier species which gradually downgrades the fall of concentration level to reach ambient state.

The values of skin friction coefficient, Nusselt number and Sherwood number is computed and shown in Table 2. It is seen that fluid parameter, elastic parameter and thermal buoyancy parameter increases the magnitude of $-\theta'(0)$, $f''(0)$ and $-\phi'(0)$ whereas Brownian motion have same effect on $-\theta'(0)$ but opposite effect is observed in case of $-\phi'(0)$. Further, destructive and constructive chemical reaction have opposite effects on Sherwood number.

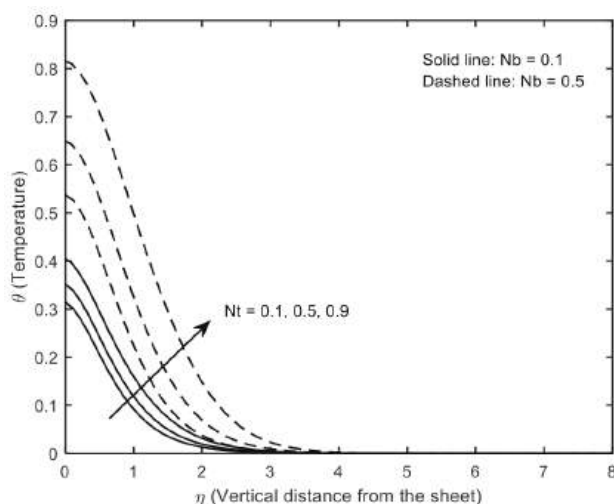


Figure 12. Impact of Nt and Nb on $\theta(\eta)$.

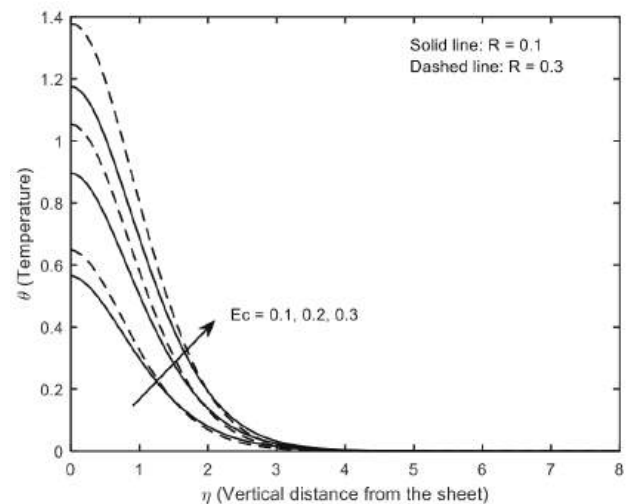


Figure 13. Impact of Ec and R on $\theta(\eta)$.

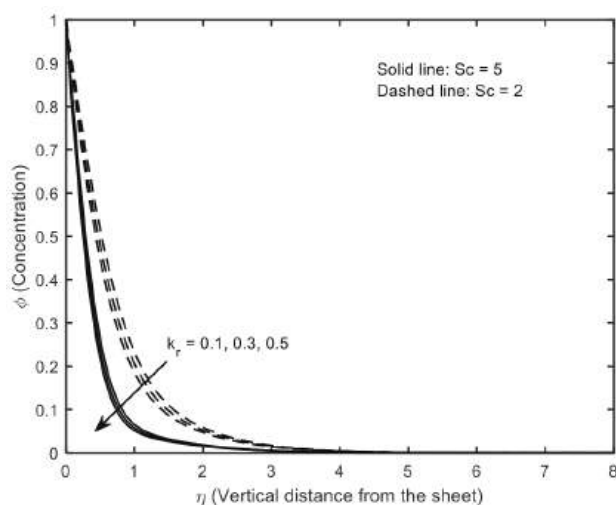


Figure 14. Impact of k_r and Sc on $\phi(\eta)$.

CONCLUSION

From the present study, the following conclusions can be drawn:

- Momentum transport gets accelerated for higher material/characteristic parameter of the non-Newtonian fluid model irrespective of presence or absence of porous matrix (Fig. 2) but thermal energy gets depleted for higher material parameter across the flow domain contributing to thermal stability (Fig. 3).
- The nanoparticle solutal concentration distribution decreases layer wise as well as vertically same as that of temperature distribution (Fig. 4). The elastic property of the Prandtl fluid and the effect of electromagnetic force increase slightly the solutal concentration but significantly to temperature. Thus, it is concluded that the generated Lorentz force (the additional body force)

Table 2. Computation of $f''(0)$, $-\theta'(0)$ and $-\phi'(0)$ when $M = k_p = 1.0$, $Pr = 0.71$, $Sc = 0.6$.

α	β	λ	Nb	Nt	Ec	λ	k_r	$f''(0)$	$-\theta'(0)$	$-\phi'(0)$
0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	-0.809671	0.093315	0.966063
0.3								-0.568545	0.099859	1.023789
0.5								-0.451017	0.102874	1.051753
	0.3							-0.449584	0.102895	1.052040
	0.5							-0.448189	0.102917	1.052321
	1.0							-0.444854	0.102968	1.052998
		0.3						-0.432493	0.103034	1.054064
		0.5						-0.420224	0.103098	1.055120
		1.0						-0.389938	0.103250	1.057718
			0.2					-0.382747	0.102342	1.085527
			0.3					-0.374647	0.101338	1.095509
			0.5					-0.355330	0.099006	1.105087
				0.2				-0.354459	0.098913	1.097858
				0.5				-0.351742	0.098624	1.076792
				0.9				-0.347859	0.098217	1.050225
					0.2			-0.366950	0.101039	1.029284
					0.3			-0.384687	0.103674	1.009820
					0.5			-0.413456	0.104713	0.980635
						0.2		-0.327032	0.190967	0.932763
						0.3		-0.246689	0.244508	0.911678
						0.5		-0.118833	0.339039	0.880934
							0.3	-0.414595	0.108160	1.145392
							0.5	-0.412800	0.107910	1.297682
							-0.1	-0.419115	0.108788	0.778311
							-0.3	-0.422152	0.109213	0.540814
							-0.5	-0.426185	0.109763	0.227303

increases the thermal transport and solutal concentration of non-Newtonian nanofluid model.

- Biot number being the index of the heat transfer resistance that enhances the concentration level of the nanoparticle.
- The higher thermophoresis enhances the concentration level that is an asset for the quality product.
- The Eckert number being the measure of dissipative effects that enhance the temperature distribution co-opted with radiation effect.
- The heavier species contribute to lower level of concentration which may be of industrial use with a care to destructive reaction rate as it is detrimental to level of concentration.

 σ^* μ a k^* g σ T T_∞ C C_∞ f θ ϕ

Stefan-Boltzmann constant

Coefficient of viscosity

Constant

Coefficient of absorption

Gravitational force due to gravity

Electrical conductivity

Fluid temperature

Ambient temperature

Nanoparticle concentration

Ambient concentration

Dimensionless velocity

Dimensionless temperature

Dimensionless concentration

ABBREVIATIONS

$$\alpha = \frac{A}{c_1}$$

Material parameter

$$\beta = \frac{a^2 \text{Re}_x}{2c_1^2}$$

Elastic parameter

$$\gamma = \frac{h}{k} \sqrt{\frac{v}{a}}$$

Biot number

$$M = \frac{\sigma B_0^2}{a\rho}$$

Magnetic parameter

$$k_p = \frac{ak^*}{v}$$

Porosity parameter

$$\lambda = \frac{g\beta}{a^2 x} (T_w - T_\infty)$$

Thermal buoyancy parameter

$$R = \frac{4\sigma^* T_\infty^3}{kk^*}$$

Radiation parameter

$$\text{Pr} = \frac{\nu}{\xi}$$

Prandtl number

$$Nb = \frac{(\rho c)_p D_B}{(\rho c)_f \nu} (C_w - C_\infty)$$

Brownian diffusion parameter

$$Nt = \frac{(\rho c)_p D_T}{(\rho c)_f T_\infty \nu} (T_w - T_\infty)$$

Thermophoresis parameter

$$Ec = \frac{u_w^2}{c_p (T_w - T_\infty)}$$

Eckert number

$$Sc = \frac{\nu}{D_B}$$

Schmidt number

$$k_r = \frac{k_c}{a}$$

Chemical reaction parameter

$$\text{Re}_x = \frac{u_w x}{\nu}$$

Local Reynolds number

 ζ

Thermal diffusivity

 $(\rho c)_p$

Effective heat capacity

 $(\rho c)_f$

Heat capacity of fluid

 ρ

Density of the fluid

 ν

Kinematic viscosity

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

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Research Article

Performance enhancement of box solar cooker with photovoltaic panel

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ABSTRACT

In developing countries, the energy needed for cooking accounts for approximately 90 % of the total energy consumption in households. A solar cooker is a tool that utilizes the sun's power to heat food and prepare it for consumption. Cooking with the aid of the sun, also known as solar cooking, is a method of preparing food that is typically practiced in settings that prioritize using as little fuel as possible and maintaining good health. The focus of the present study is the inability to perform during the low solar intensity. It is significant for the performance of the solar cooker as it can hamper the motive of using the renewable energy and to conserve the energy used by fossil fuels through solar cookers. It is highly significant for the acceptability of the solar cooker in the society as it is one of the major causes of non-popularization of solar cookers. In the current work, an experimental approach is used in which the construction of a hybrid solar cooker is accomplished by merging the photovoltaic panel and the heater following the design adjustments done in the constructed solar cooker with optimized sidewall inclination to increase the heat intake to the solar cooker. The developed cooker has inclined sidewalls giving the heat to the food vessels from the sides also, which increases the heat given to the food vessel and it is incorporated with the 50 W photovoltaic panels at cover lid and a DC heater placed beneath the absorber plate for giving the auxiliary heat. The use of sidewall inclination with the photovoltaic panel of only 50 watts is rarely observed in the literature. Hence this work of incorporating photovoltaic panel and heater with the inclined side-walled solar cooker is novel for that aspect. Extensive experimental research is carried out on the newly created hybrid cooker, and its performance is evaluated about that of the conventional solar cooker. The maximum temperature achieved in the hybrid cooker during no-load tests is found 25% higher than that of the conventional cooker. The figures of Merit F_1 and F_2 with values 0.11 and 0.63 respectively were also found higher in hybrid solar cooker as compared to conventional solar cooker having values of 0.09 and 0.375 respectively and with few researches found in literature. Even during the month of December, when the solar radiation is as low as 500 W/m², cooking was possible within 120 minutes. The experimental results project new developed hybrid solar cooker to be more efficient than the conventional solar cooker. Performance parameters are also found better as compared to few researches found in literature. Cooking is also possible during low solar radiation which can be useful to make the solar cooker more popular.

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INTRODUCTION

Energy has been an integral component of human life and cannot be separated from it. The availability of energy exerts a beneficial influence on the overall quality of life. The sustained survival and welfare of both individuals and communities are significantly contingent upon the accessibility of energy resources. The strategies of energy conservation and the implementation of renewable energy sources constitute two principal methodologies for energy management and ought to be concurrently embraced. In this regard, it is imperative to initially identify and assess energy consumption patterns across various sectors. According to the report published by the International Energy Agency concerning global energy statistics (2022 Key World Energy Statistics [1]), in the year 2022, residential buildings accounted for 29% of the total consumption of natural gas worldwide. Furthermore, residential buildings utilized 28% of the global electrical energy consumption. In light of this data, Figure 1 illustrates the proportion of the major global sectors consuming the energy for both natural gas and electricity in the year 2022.

As illustrated in Figure 1, residential structures globally constitute the second most energy-intensive sector. This observation has prompted both researchers and corporations to explore various strategies aimed at mitigating the energy requirements of edifices. Nevertheless, the substantial energy utilization of buildings is not the sole issue of concern; energy consumption is also intrinsically linked to the emission of greenhouse gases (GHG). Evidently, diminishing energy utilization in residential structures will exert a considerable impact on worldwide energy consumption and greenhouse gas emissions. Globally, developing nations are typified by low per capita energy consumption, primarily for lighting and cooking; high rates of unemployment in rural areas; a significant reliance on biomass energy sources in rural areas; and a significant disparity in living standards between urban and rural populations [2]. Technologies for renewable energy have recently been used as a sustainable

development strategy [3]. It is possible to turn any renewable energy source into electricity, which may then be utilized for heating and cooking, resulting in the reduction of conventional energy utilization. Such renewable sources include Biomass energy, wind energy, geothermal energy, small hydro power energy and Solar energy. Renewable sources of power, such as those derived from the sun and the wind, have become increasingly popular over the last few decades. Solar energy is the most practical and advantageous kind of renewable energy source since it can be turned into a broad variety of different forms of energy both directly and indirectly. This makes solar energy the most effective and useful form of renewable energy source that has a negligible impact on the surrounding ecosystem. The radiant energy that the sun produces is known as solar energy. Kumar and Ramakrishnan [4] presented several significant outcomes regarding the role and advancements of solar thermal systems (STS) in domestic applications. Solar cookers are gadgets that harness the radiant energy of the sun to cook food. One of the important uses of solar energy is solar cooking, which can take the place of traditional cooking methods. Solar cooking is a well-established technology that is used to cook food in situations where maintaining a healthy diet is of utmost importance. It contributes to the reduction of deforestation as well as the usage of fuel, both of which are growing more expensive as time goes on. Cooking by solar cooker also helps in 17 SDG (Sustainable Development Goals) as per the policy of the United Nations [5]. Dewangan [6] outlined the current status and future possibilities for solar box cookers, focusing on their practical utility, performance parameters, design improvements, social acceptance, and the application of computational techniques to enhance their development. The efficient outcome for setting up a solar energy plant in a Turkish hamlet was determined by Taner and Dalkilic's [7] feasibility study of solar energy-technoeconomic analysis.

Various classifications for solar cookers may exist based on the research methodology. They fall into two primary

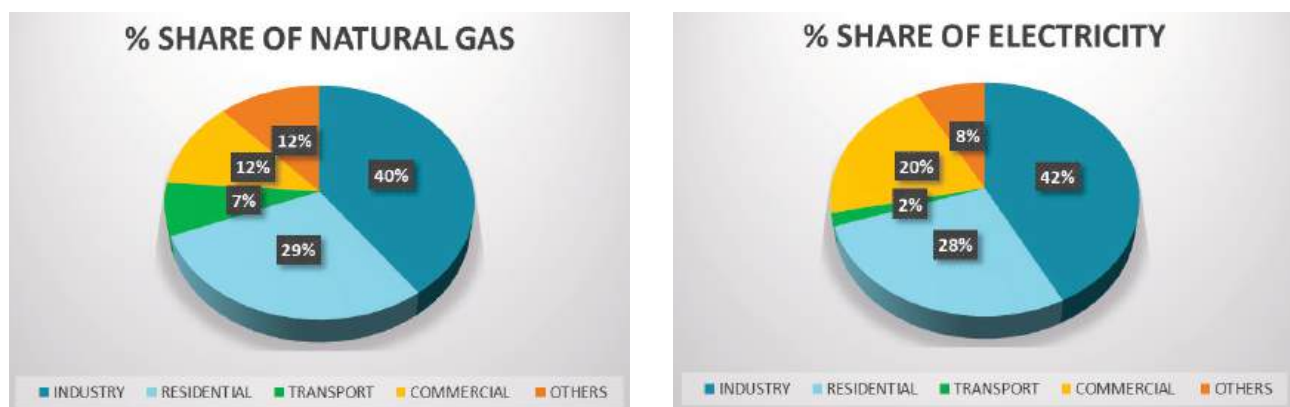


Figure 1. Share of major sectors consuming energy in the total global energy consumption from energy sources like natural gas and electricity in the year 2022 [1].

Table 1. Advantages and disadvantages of Different solar cookers

Type of Solar Cooker	Advantages	Limitations
Box Cooker	• Can cook multiple dishes at once. • Easy to build and use. • Good for slow cooking.	• Requires direct sunlight for effective cooking. • Bulky and may be difficult to transport.
Parabolic Cooker	• Cooks' food quickly (high temperatures). • Can be used for frying, grilling, and baking.	• Requires precise alignment with the sun. • Can be dangerous due to high temperatures.
Panel Cooker	• Lightweight and portable. • Relatively inexpensive to make. • Good for small quantities.	• Slow cooking times. • Less efficient in cloudy weather.
Concentrating Cooker	• Can achieve high temperatures for grilling and baking. • More stable than parabolic cookers.	• Requires consistent sunlight. • Can only cook small quantities at once.

categories: direct and indirect, depending on how solar thermal energy is transmitted to the cooking vessel [8]. Alternatively, there are four primary types of cookers when it comes to the device's configuration: box cooker, concentrating cooker, and panel cooker [9,10]. The advantages and limitations of different type of solar cookers are shown in the table 1.

Looking to the benefits available from the solar cookers, Yettou [11] provided critical insights into the thermal performance and efficiency of parabolic solar cookers in Algeria, emphasizing the importance of regional and seasonal factors in their adoption and use for solar cooking applications.

From the literature, it is found that researchers have worked on various aspects related to the solar cookers namely Mathematical modeling and computational analysis [12–19], Design modifications in solar cooker by [20–28] and by using phase change materials [29–37].

From the review of literature on mathematical modeling and design modifications of solar cookers, it is observed that only one of the models have considered the effect of the thermal contact resistance (TCR) between the absorber plate and the food vessel used to cook the food in the solar cooker which is given by Vaidya and Channiwal[19]. Also, from the literature of different design modifications, it is found that only few design modifications increase the solar insolation available for cooking apart from direct solar incidence. Though different designs and modifications are done in the solar cookers, the major drawback for the solar cooker is its inability to perform in the cloudy day when the solar insolation is less.

Various researchers have carried out experiments on solar cooker for improvement of its performance and analysis by electric back up. Nandwani [38] created an electric cum solar oven that can be used for cooking and baking practically any kind of dish at any time of day or night, regardless of whether solar or electric energy is being utilized. It was comprised of a black-painted electric plate that had a surface area of 0.16 square meters and a wattage of

1500. It was protected by a box made of wood and featured two glass coverings that were spaced apart by 2 cm. The thermostat had a range of 121 °C to 204 °C, which was used to adjust the temperature. Hussain et al. [39] developed a box-type cooker equipped with auxiliary heating. They investigated the performance of two distinct solar cookers through a series of trials. One has an outside box made of teak wood, and the other with an outer box made of Tor to supply 900 watts, six electric iron heating elements, each of which is 150 watts, were utilized. In a different configuration, a holder for a bulb was utilized to supply the heat. The diameter of the bulb holder measured 100 cm, while its length measured 20 cm. The bulb holder was outfitted with a 100-watt bulb that was inserted inside. The bulb holder was pierced with holes so that hot air could flow through it. The results of the first cooker's F_1 and F_2 measurements were found to be 0.16 and 0.27 respectively, whereas the results of the second cooker's F_1 and F_2 measurements were 0.17 and 0.32 respectively. On November 14, 1994, the highest temperature that could be achieved was 90 °C in cooker 1, and 45 °C in cooker 2. Chaudhari [40] estimated the amount of electrical power required by the heater for the solar box type cooker. In his estimation, a power backup of 160 W was plenty for any solar cooker. The natural convective heat transport within a trapezoidal enclosure was investigated by Kumar [41]. The plate was heated by five electrical heaters of the plate type, each measuring 0.38 meters in length and having a total capacity of 250 watts. These were fastened to the bottom of the aluminum tray in order to get consistent heating using an electric power source stabilized by a servo motor. Shehzad [42] designed a sun oven that is circular, electric, and cum. The bottom of the oven consisted of an electric heating plate that was regulated by a timer. The temperature within the oven was maintained by an electric thermostat. The diameter of the plate was 24 inches, and it was protected by an external frame made of wood that had a glass cover at the very top of it. The copper sheet used to make the plate had a 24 gauge.

Mahavar et al. [43] developed a solar cooker that had a backup electric heating system. The outside of the cooker has dimensions of $59 \times 59 \times 23 \text{ cm}^3$. To supply the solar cooker with additional heat, they utilized four Mica sandwich strip heaters, each of which had a power output of 90 W. Both the F_1 and F_2 figures of merit for solar cookers have been determined, and the results show that their values are 0.12 and 0.462 respectively. During the interior testing, they found that the use of electricity for the cooking of a single meal was 0.38 units, but during the outdoor test, it was just 0.18 units, which is less than 50% of the indoor value. Jani and Joshi [44] created a photovoltaic and thermal hybrid solar cooker and gave it the name Improved small-scale solar box type hybrid solar cooker (ISSBH). It weighed 6.5 kilograms and was a solar box-type hybrid cooker. As a backup system, they employed five solar panels, each producing 15 watts, and three rod-type DC heaters, each producing 25 watts. They also provided a battery for use during the night when cooking. In January, the value of F_1 was determined to be 0.087 based on the results of their performance tests on their cooker. The solar panel has dimensions of 0.3 meters by 0.3 meters by 0.005 meters in volume. In the beginning, they supplied a steady operating wattage of 30 watts, however, this was not enough for cooking, so they eventually increased the supply to 50 W.

Thus, with auxiliary heating system, food can be prepared at any time of the year by utilizing either only solar energy (when there is adequate or above-average sunshine), only electric energy (when there is insufficient solar radiation), or a combination of the two types of energy (when there is partial solar radiation). Thus, dependency on the conventional source is reduced. It is expected that with the use of auxiliary energy, when necessary, a solar cooker may be used throughout the year in areas where electricity is available.

From the literature, it is observed that the inability of solar cookers to perform during the low radiation is the main drawback of the solar cookers. It has been addressed in different ways by different researchers by incorporating biomass technology and incorporating electric supply with batteries. This increases the cost of batteries and is difficult to perform in the areas where there is no electric supply. Hence, some better design of solar cooker to increase the solar radiation intake and some source for auxiliary heat supply should be found. Jani and Joshi's [44] solar cooker has an increased weight and price as a result of their use of five photovoltaic panels with a total power supply of seventy-five watts. The use of photovoltaic thermal system is done by many researchers in varied applications [45–47].

Thus, the objective is framed to develop an energy efficient solar cooker with better design, considering the effect of thermal contact resistance between absorber and food vessel and to improve the design of box type solar cooker by incorporating solar panel so as to get benefit of extra heat supply and to reduce operating cost as compared to electric

supply back up. Hence, in the present work, as a novel approach, a hybrid box type solar box-type is designed following Goswami et al. [48] and Vaidya et al. [49], modelled incorporating thermal contact resistance (TCR) as per Vaidya and Channiwala [19] and fabricated with the side walls inclined as per the design modification suggested by Vaidya et al. [26] and further, it incorporates both a solar panel and a heater to reap the benefits of an additional heat supply and achieve the desired level of performance even when there is a lower amount of solar insolation e.g. in December when the global radiation is about 500 W/m^2 . This can benefit the society to popularize the solar cooker and to use the renewable energies for the sustainable development. The performance of the newly constructed hybrid solar cooker with the solar panel is analyzed through the lens of experimentation, and it is compared to that of the conventional cooker.

MATERIALS AND METHODS

This section is divided into two subsections namely (a) Development of Hybrid Solar Cooker and (b) Experimental studies on the developed hybrid solar cooker.

Development of Hybrid Solar Cooker

The methodology chosen is based on a mathematical model given by Vaidya and Channiwala [19]. Calculation of the size of a hybrid solar cooker is carried out considering the energy required to cook the food for a small family as suggested by Goswami et al. [48]. The aluminum plate having the dimensions of the base as $340 \times 340 \text{ mm}^2$ in size and has a depth of 70 mm is used as an absorber plate. The measurement of the plate's top aperture area is $560 \times 580 \text{ mm}^2$.

To attain these dimensions, an aluminum absorber plate is first cut and then bent suitably. The design calls for the side walls to be bent in such a way that they take advantage of the optimal angles as proposed by Vaidya et al. [26]. The surface is prepared by removing dust and contaminants from the surface in a sequential manner. Polishing and buffing are two types of mechanical cleaning processes. Abrasives are utilized during the polishing process in order to remove minute amounts of metal from the surface. The purpose of this step, which comes before buffing, is to create a surface that is free of the larger defects that grinding leaves behind. Following closely behind mechanical cleaning is the use of chemicals to remove pollutants. The pollutants include a wide variety of oils and greases, as well as waxes and organic compounds.

Glass and plastic have traditionally been the go-to options when it comes to solar cooker glazes. The transmittance of the glass is relatively high at around 90 %. The infrared radiation that is released by the hot absorber plate cannot pass through it very easily since it is almost entirely opaque to such radiations. As a result of its availability in very thin sheets, the use of plastic rather than

glass significantly reduces the expense associated with the construction of solar systems. However, exposure to sunshine causes the plastic to depolymerize, which results in it becoming brittle and brown in color. Therefore, the plastic's transmissivity will gradually decrease to a significant level as time goes on. The solar cooker with double glass can reach temperatures between 150 and 250 °C, which is a higher temperature than the solar cooker with a single glass, which means that the amount of time needed for cooking can be reduced. Therefore, there must be two panes of glass in the glazing. There were two pieces of float glass utilized, each measuring 4 mm in thickness, and there was a gap of 15 mm between two pieces of glass. After creating a groove gap in the frame, the inner and outer glass are inserted using epoxy adhesive, and then the assembly is left alone for five to six hours so that the glue may fully cure. Glass frame and housing body are connected together utilizing hinges to complete the connection. The insulation is placed behind the absorber plate as well as all around the absorber plate's perimeter. It prevents heat from escaping via the back and sides of the collector. The material Cerawool, which has a density of 64 kg/m³ and a thermal conductivity of 0.48 W/mK., is utilized for insulation purposes.

The solar cooker's various parts are safely housed in an enclosure that is impervious to the elements. This enclosure prevents moisture, dirt, dust, and air infiltration from entering the collection, which would have a negative impact on the collector's overall performance. Aluminum composite panel is utilized for the exterior body frame because of its low weight and promising lifespan in terms of payback period. Aluminum composite panel is a combination of aluminum and aluminum alloy. The top cover lid of the solar cooker has been outfitted with a foldable solar panel with dimensions of 540 x 540 mm². In conditions with no load, the solar panel has a voltage output of 35 V. Below the absorber plate is a heater that is 340 x 340 mm² in size and operates on 24 V of direct current.

The instrumentation includes photovoltaic panel, thermocouples and D.C. heater. The photovoltaic panel is where the supply comes from for the heater. Thermocouples of the T type are used to measure the mean plate temperature, the temperature of the glass cover, the temperature of the air inside, and the temperature of the food and water that is being cooked. The great sensitivity of 40-50 mv/ °C offered by T type thermocouples makes them the preferred option. The thermocouple undergoes calibration with the ice point serving as its reference junction. In order to determine the average temperature of the outer glass cover in both the loaded and unloaded states, thermocouples have been attached to the inner surface of the outer glass cover with the help of epoxy adhesive. In addition, thermocouples have been attached to the top and bottom of the absorber tray, the side walls of the housing, and the interior of cooking pots.

The standard solar panel has an input rate of around 1000 W/m², however the solar panels now available in the

market achieve an efficiency gain of no more than 15 to 20% at best. Consequently, a 1 m² solar panel will generate between 150 and 200 W when exposed to adequate sunshine. The area of the cover of the designed solar cooker is 600 × 600 m² and the monocrystalline photovoltaic panel of size 580 mm × 535 mm × 3 mm readily available in the market can be fixed on it.

Estimated average global radiation at Surat, Gujarat, India throughout the year for 9 am to 10 am considering clear days is 774.82 W/m². [50,51]

- Considering 17% efficiency, the power available will be 40.87 W ($=0.3103 \text{ m}^2 \times 774.82 \times 0.17$).
- Estimated average global radiation at Surat throughout the year for 12 pm to 1 pm considering clear days is 950 W/m². Power available will be 50.12 W.

Thus, flexible solar panel is applied at the top reflector cover of the solar cooker and the power generated by the photovoltaic panel is given to electrical heater. A flexible solar panel of size 580 × 535 mm² has been placed at the top cover lid of the solar cooker. The solar panel is generating 35.2 V at no load condition. The Specifications of the solar panel are given in Table 2.

Figure 2 shows the Line Diagram of hybrid solar cooker and Figure 3 shows the actual photograph of hybrid solar cooker.

Table 2. Specifications of Solar Panel

Type of Solar Panel	Monocrystalline
Maximum Power ($P_{\max}$)	50 W
Maximum Voltage ($V_{\max}$)	35.2 V
Maximum Current ($I_{\max}$)	1.42 A
Dimensions in mm	580 x 535 x 3
Weight in kg	0.6

STC: Irradiance: 1000W/m², Temperature: 25 °C, AM: 1.5

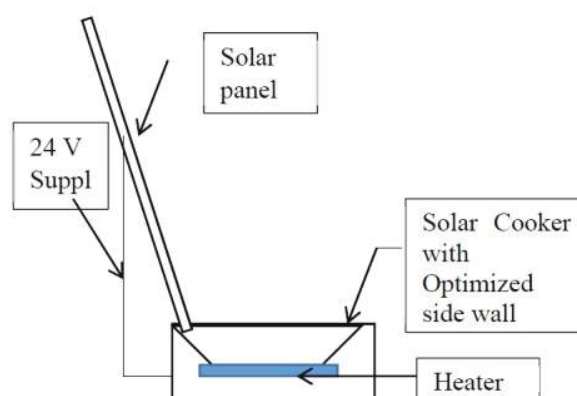


Figure 2. Line Diagram of hybrid solar cooker.

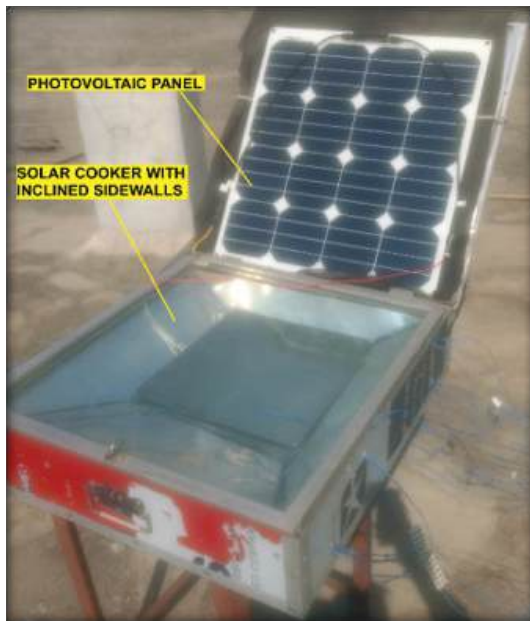


Figure 3. Actual Photograph of Hybrid Solar Cooker.

Experimental Studies on Hybrid Solar Cooker

The details of experiments performed on hybrid solar cooker to assess its performance are given in the Table 3.

Table 3. Details of Experimentation

Sr. No	Date of experiment	Details
1	21-11-18	No load tests with and without heater
2	04-12-18	No load tests. Cooker facing due south
3	17-12-18	No load tests. Cooker facing due south
4	05-12-18	Load tests. Cooker facing due south.
5	10-12-18	Load tests. Cooker facing due south.
6	31-12-18	Cooking tests

Finding out the figure of merit F_1 for no load situations and the figure of merit F_2 for load conditions was one of the ways that thermal performance tests for the newly built solar cooker were carried out [52]. A stagnation test with no load is used to determine the value of the figure of merit F_1 , which represents optical efficiency. This figure of merit is the ratio of optical efficiency to heat loss factor. It is calculated by the equation (1). [52,53]

$$F_1 = \frac{\eta_o}{U_{LS}} = \frac{T_{PS} - T_a}{I_{GS}} \quad (1)$$

Where,

η_o = the optical efficiency,

U_{LS} = the heat loss factor at stagnation

T_{ps} = the stagnation plate temperature ($^{\circ}\text{C}$)

T_a = the plate temperature ($^{\circ}\text{C}$)

I_{GS} = insolation on the horizontal surface at the time the stagnation temperature is reached (W/m^2)

In order to calculate the second figure of merit, F_2 , the solar cooker is left out in the sun with a full load of water and utensils while the reflector is removed from the device. The initial temperature of the water is maintained at temperature of 60°C which is an average between room temperature and the point at which it will boil. Both the water temperature and the sun radiation are recorded until the water temperature reaches 95°C . The figure of merit, F_2 , is calculated using the equation (2). [52,53]

The heat that is delivered to the plate by the heater is also considered in the calculation of F_1 and F_2 for the hybrid cooker, and the total insolation is determined by adding the heat that is received from the sun and the heat that is received from the heater. In the month of December, a cooking test is carried out in order to evaluate the hybrid cooker's capacity for cooking, and the results are compared with those obtained using the conventional cooker.

$$F_2 = \frac{F_1(MC)_w}{A(t_2 - t_1)} X \ln \left\{ \frac{1 - \frac{1}{F_1} \left[\frac{T_{w2} - T_a}{I_G} \right]}{1 - \frac{1}{F_1} \left[\frac{T_{w1} - T_a}{I_G} \right]} \right\} \quad (2)$$

Where,

F_1 is the first figure of merit from the stagnation test;

$(MC)_w$ is the product of mass of water (kg) and specific heat ($\text{J/}^{\circ}\text{C}$)

A is the aperture area of the cooker of the cover plate (m^2);

$t_2 - t_1$ is the time taken for heating from initial water temperature T_{w1} to T_{w2} (sec).

T_a = the average air temperature during the time period $t_2 - t_1$ ($^{\circ}\text{C}$)

I_G = insolation on the horizontal surface during the time period $t_2 - t_1$ (W/m^2).

Figure of merit 2, F_2 , depicts the ability of solar cooker to transform the available heat to the food through cooking vessels.

RESULTS AND DISCUSSION

The newly created hybrid solar cooker was put through a no-load test at first so that researchers could determine what impact the addition of a heater would have on the cooker. Cooking was done on the cooker with the heater connected and without the heater connected on the 21-11-18. The outcomes are presented as below.

Figure 4 demonstrates the fluctuation in temperature of the plate of inclined side-walled solar cooker as a function of time both with and without the heater on 21-11-18. Both curves exhibit a similar trend in the temperature profile, but the maximum temperature that can be achieved with the heater is far higher. It's possible that the additional

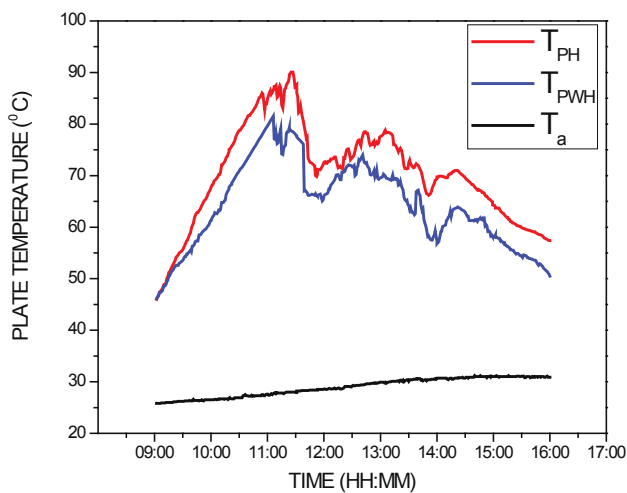


Figure 4. Temperature variation in inclined side-walled solar cooker with heater and without heater effect on 21-11-18.

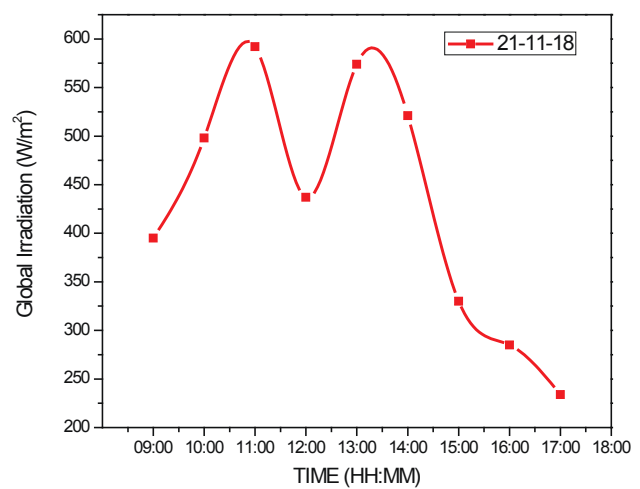


Figure 5. Global Horizontal Irradiation on 21-11-18.

heat made available by the photovoltaic panel is the reason for this. If the heater is present, it can reach a maximum of around 90 °C, but it can only get to about 82 °C on its own. In addition to this, the temperature decrease in the cooker appears to be less severe when the cooker is equipped with the heater than when it is without the heater. In addition, it has been seen that the starting temperature of the plate is maintained at the same level in both cases, when the solar radiation starts to fall on both the cookers. After 9:30 in the morning, a slow but steady rise in temperature may be noted. This is due to the fact that the solar panel does not have sufficient power for the heater to function at its maximum capacity during the aforementioned hours. As soon as a significant quantity of electricity is created by the photovoltaic panel, the heater is turned on at its maximum

capacity, also known as its highest wattage. This is reflected in the data presented in figure 4 as well. After 1:00 pm, the heater gets heated at full capacity, and as a result, the maximum temperature difference of around 6-7 °C between the two solar cookers is observed until 4:00 pm. Thus, by the auxiliary heating obtained by the photovoltaic panel and the heater, an increase of about 9% is observed in the maximum plate temperature.

Figure 5 shows the Global Horizontal Irradiation (GHI) on 21-11-18. From the figure, it is observed that there is a dip in the values after 11.00 a.m. The variations in the plate temperatures of solar cookers with and without heater may be attributed to these variations in the GHI.

Figure 6 demonstrates the difference in temperature between the plates in the conventional cooker, the hybrid cooker, and the ambient temperature as a function of the local time during no-load test on 4-12-18 and 17-12-18.

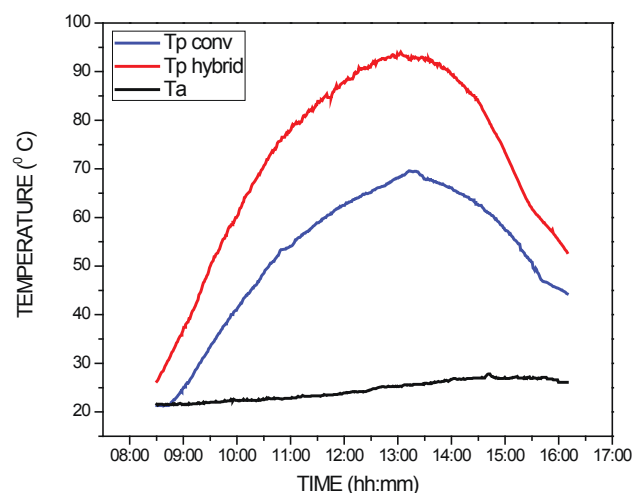
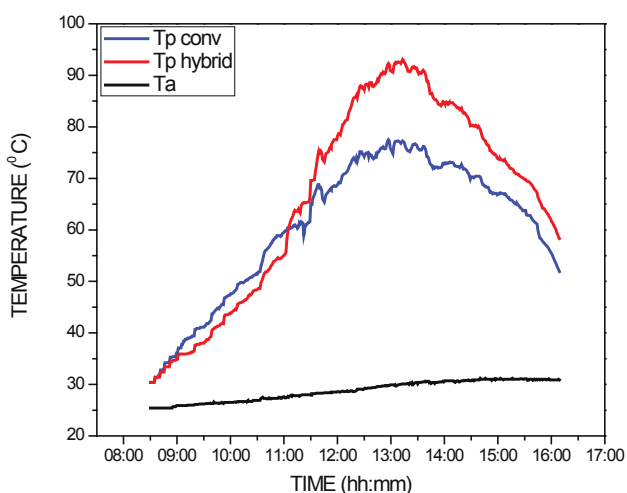


Figure 6. Variation of Plate Temperatures of conventional solar cooker, Hybrid solar cooker and ambient temperature with local time during no-load tests on 4-12-18 and 17-12-18.

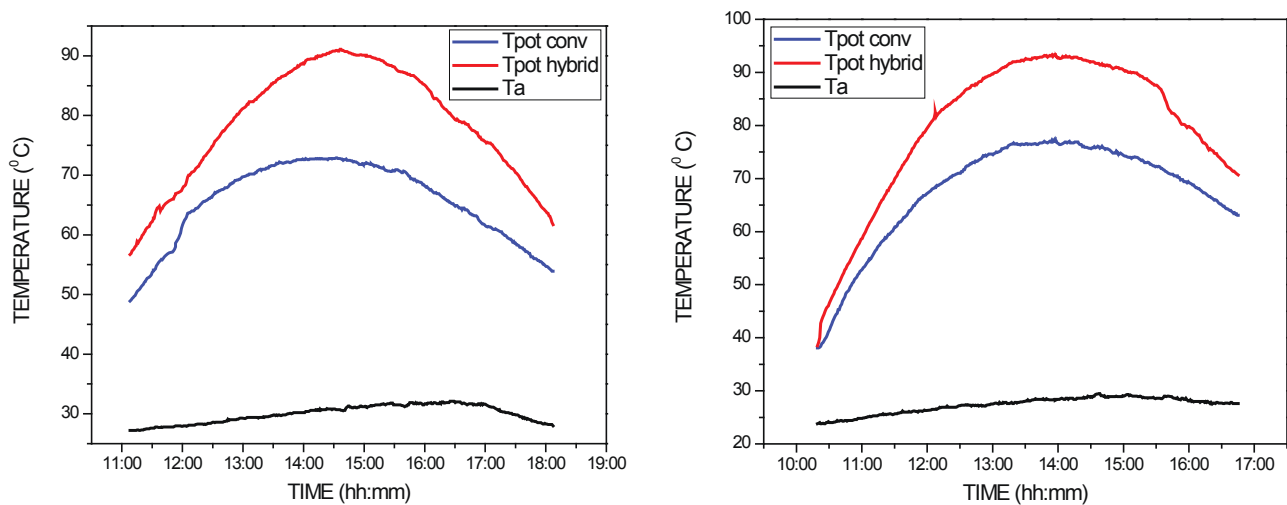


Figure 7. Variation of Pot Temperatures of conventional solar cooker, Hybrid solar cooker and ambient temperature with local time during load tests on 5-12-18 and 10-12-18.

There is a correlation between the passage of time and an increase in temperature, and this correlation is seen in both conventional and hybrid cookers. The hottest part of the day occurs around in the middle of the day and obviously, the peak solar insolation occurs at midday. Additionally, in comparison to a conventional cooker, the temperature that is maintained throughout the day in a hybrid cooker is higher.

The maximum temperature achieved in the hybrid cooker is found 25% higher than that of conventional cooker due to sidewall inclination of the solar cooker and the addition of an auxiliary heating by the DC heater empowered by the photovoltaic panel. Plate temperatures of both cookers reach to their maximum value at nearly 1:00 p.m., but then the rate of reduction in the temperature of hybrid solar cooker is found more which emphasizes the need to retain the heat for more time.

The range of figure of merit, F_1 , that may be estimated from equation (1), for a conventional cooker is 0.07 to 0.10, but the range for the solar cooker having a modified design with photovoltaic panel is 0.096 to 0.12 respectively. The maximum value of 0.10 for a conventional cooker is lower than the maximum value of 0.12 for a newly developed hybrid cooker which justifies the idea of using the photovoltaic panel. The average figure of Merit F_1 is found to be 0.11 for the hybrid cooker and 0.085 for the conventional solar cooker for no load tests.

Figure 7 demonstrates how the temperature of the water-filled pot changed throughout the load test using both a conventional and a hybrid solar cooker on 5-12-18 and 10-12-18. The tests were conducted for six days and the similar trends were observed on all days. In the beginning, the temperatures of pots in both the solar cookers are the same. However, after a short period of time, it is discovered that the temperature of the pot in the hybrid cooker is increasing more than that of the conventional cooker.

Finally, it reaches the maximum value around 14:00 hours in the afternoon. The pot temperature of hybrid cooker is found about 25 % higher during the period of 12:30 pm to 15:00 pm. Also, the maximum value of pot temperature is found to be around 18 °C higher than that of the conventional solar cooker. As a result, the capacity for accumulating heat that a hybrid cooker possesses can be deemed to be greater than that of a conventional cooker.

As calculated from equation (2), the figure of merit, F_2 values that can be achieved with a conventional solar cooker range from 0.3 to 0.45, whereas those that can be achieved with a hybrid cooker range from 0.36 to 0.91. In comparison, the typical value of F_2 for a conventional cooker is 0.375, whereas the value for a hybrid cooker is 0.635.

The results of figure of merit, F_1 and F_2 are summarized and given in Table 2 and Table 3.

Table 2. Summarization of Results for no-load tests

Date of Experimentation	F_1 for Conventional Solar Cooker	F_1 for Hybrid Solar Cooker
04-12-18	0.07	0.096
17-12-18	0.10	0.12
Average	0.09	0.11

Table 3. Summarization of Results for load tests

Date of Experimentation	F_2 for for CSC	F_2 for HSC
05-12-18	0.30	0.36
10-12-18	0.45	0.96
Average	0.375	0.635

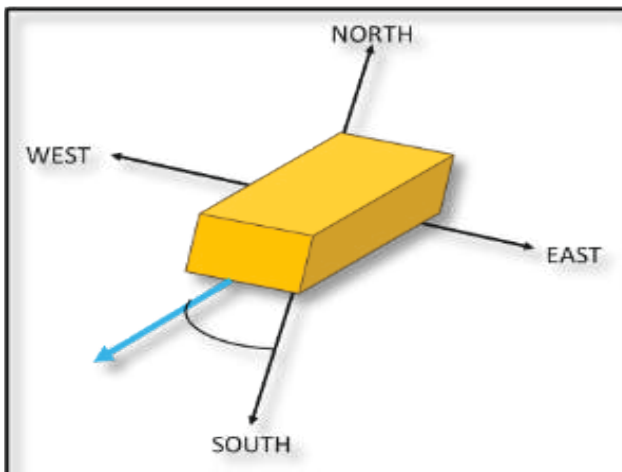


Figure 8. Schematic Diagram showing the direction of Hybrid solar cooker with respect to south direction.



Figure 9. Photographic view of Hybrid solar cooker showing the inclination of photovoltaic panel with the horizontal plane.

Cooking test was carried out on 31-12-2018 on hybrid solar cooker and conventional solar cooker. The direction of both the cookers were kept facing the sun. The surface azimuth angle was calculated to be 34.91° and inclination of the cover with solar panel to be 53.29° with the horizontal surface at 10 am and 18.97° and 49.9° at 1 p.m. [54]. Hence the cooker was placed at 26.94° from south as shown in Figure 8 and 50° inclination was given to the photovoltaic panel as shown in Figure 9.

The cooking test was performed using the hybrid cooker by placing two pots of Moong daal and two pots of Toor daal into the hybrid cooker. Two pots had 200 gm of Moong daal and 350 gm of water in each while other two pots had 200 gm of Toor daal and 350 gm of water in each as calculated from BIS standards[53]. The hybrid cooker was placed with its front facing due south. After the cooking period of



Figure 10. Toor daal and Moong daal placed in the hybrid solar cooker for cooking during experimentation.



(a)



(b)

Figure 11. Toor daal (a) and Moong daal (b) found fully cooked after the cooking period.



Figure 12. Cooked Toor Daal checked and found fully cooked.

panel to enhance the performance of the solar cooker is one of such approaches. The amalgamation of photovoltaic (PV) panels with solar cookers signifies a considerable progression in the realm of solar energy applications, thereby augmenting both culinary efficiency and energy sustainability. This hybrid methodology synergizes the advantages of solar thermal cooking with the electrical output generated by PV systems, thereby facilitating versatile energy utilization within domestic environments. Thus, with an objective to develop an energy efficient solar cooker with better design and performance during the low solar intensity conditions, a hybrid box type solar cooker is designed, developed and integrated with a monocrystalline photovoltaic panel of 50 watt placed at the top cover lid. A DC heater is placed beneath the absorber plate which gets the power supply through the photovoltaic panel. Refereeing the literature available, use of sidewall inclination with the

Table 4. Comparison of similar work from literature

Parameter	Joshi [55]	Mahavar[43]	Present study
Time of study	January	April and July	December
Source of Auxiliary heat supply	Photovoltaic panels	Electric back up supply	Photovoltaic panels
Power supply	75 watts	Power backup of 180 watts	50 watts
Figure of merit F_1	0.087	0.093	0.11
Figure of merit F_2	-	0.462	0.63
Cooking Time	Not mentioned	80 minutes with electric backup of 130-170 watts	120 min
Efficiency η	30% (SSBH model)	-	31.5%

120 minutes, both Toor daal and Moong daal were found to be fully cooked.

Figure 10 shows the pots of Toor daal and Moong daal placed in the hybrid solar cooker for cooking during the experimentation. Figure 11 shows the Toor daal and Moong daal found fully cooked after the cooking period. Figure 12 shows the condition of Toor daal after conforming that it is fully cooked.

Table 4 shows the comparison of results with the similar research work of Mahavar and Joshi with the present study.

CONCLUSION

With the anticipated high demand for energy to be utilized in cooking, it is currently highly relevant to make improvements to cooking devices based on the application of solar energy by which reduction in CO₂ emissions, deforestation and air pollution can be achieved. The focus of the present study is the inability of the solar cookers to perform during low solar intensity. Newer approaches for making solar cookers usable in the low intensity of sunlight are necessary to advance the technology. Use of photovoltaic

photovoltaic panel of only 50 watts is not observed. Hence this work of incorporating photovoltaic panel and heater with the inclined side-walled solar cooker is novel for that aspect. No-load tests, load tests and cooking test were carried out for the performance of the newly developed hybrid solar cooker.

Main findings of the present study can be summarized as below

- There is an increase of 9% in the maximum plate temperature of inclined side-walled solar cooker with the addition of auxiliary heating operated by solar panel.
- The temperatures of the plates that are achieved in the newly developed hybrid solar cooker are found to be greater by 25% than those that are achieved in the conventional cooker.
- The figure of merit F_1 for the hybrid solar cooker, which was acquired during the stagnation test, is discovered to be higher throughout the day than that of the conventional cooker. The average F_1 for conventional solar cookers is 0.09, whereas the average F_1 for hybrid cookers is 0.11. The value of F_1 is also found higher than that of Joshi & Jani and Mahavar et.al.

- During the load testing, the average figure of merit F_2 for the hybrid cooker was found to be 0.635, whereas the average figure of merit F_2 for the conventional cooker was found to be 0.375. It is also found higher than that of Mahavar et. al.
- Cooking tests performed on the hybrid and conventional cookers depict that food is found fully cooked in the newly developed hybrid solar cooker after the cooking time of 120 minutes. This suggests that the newly developed hybrid cooker can also be used even during December when the average solar intensity is around 500 W/m^2 or even less, with the help of the heat input by the heater which is available due to the solar panel.

Thus, the experimental results project new developed hybrid solar cooker to be more efficient than the conventional solar cooker.

Future research work may be focused to integrate solar cookers with various renewable resources together such as thermal, photovoltaic panels, biomass etc., to get an increase in the heat input. The use of various nanomaterials in combination of hybrid solar cookers can be done for better heat accumulation. For heat retention, the hybrid cookers may be integrated with different energy storage materials. Improvements in the design, materials used to construct solar cookers, insulating materials, reflective materials etc. can be focused. The design and performance of solar box cookers can be optimized through the use of a variety of computational techniques, such as computational fluid dynamics (CFD) and Artificial Intelligence Things (IoT). These methods provide novel approaches to evaluating and enhancing solar box cooker efficiency, opening the door for further developments in solar cooking technology.

NOMENCLATURE

A_p	Aperture area of the cooker of the cover plate (m^2)
F_1	The figure of Merit 1 from the stagnation test
F_2	The figure of Merit 2 from the load test
I_{GS}	Insolation on the horizontal surface when the stagnation temperature is reached (W/m^2)
U_{LS}	Heat loss factor at stagnation
T_{PS}	Stagnation plate temperature ($^{\circ}\text{C}$)
$t_2 - t_1$	Time taken for heating from T_{w1} to T_{w2} (s).
T_a	Ambient temperature ($^{\circ}\text{C}$)
MC_w	Heat Capacity of water ($\text{J/}^{\circ}\text{C}$)
T_{pH}	Plate temperature with heater ($^{\circ}\text{C}$)
T_{pWH}	Plate temperature without heater ($^{\circ}\text{C}$)
$T_{p \text{ Conv}}$	Plate temperature for conventional solar cooker. ($^{\circ}\text{C}$)
$T_{p \text{ hybrid}}$	Plate temperature for hybrid solar cooker. ($^{\circ}\text{C}$)
$T_{\text{pot Conv}}$	Temperature of Pot in the conventional solar cooker ($^{\circ}\text{C}$)
$T_{\text{pot hybrid}}$	Temperature of Pot in hybrid solar cooker ($^{\circ}\text{C}$)

Greek Letters

ε	Emissivity of the material
η_o	Optical efficiency

Abbreviations

TCR	Thermal Contact Resistance
GHI	Global Horizontal Irradiation (W/m^2)

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

Authors confirm that the data that supports the findings of this study are available within the article. The raw data that supports the findings of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The authors declare 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.

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Research Article

Potential performance of diesel engine powered by mango seed blends + pyrolysis oil doped with zirconium catalyst

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ABSTRACT

The release of exhaust gases from compression ignition engines has gradually increased daily due to increased demand for automobiles. This causes a heavy depletion of diesel. The novelty in this work is testing the single-cylinder diesel engine followed by 50D25MO25PO+ 20 ppm zirconium oxide, 70D15MO15PO + 40 ppm zirconium oxide, 80D10MO10PO + 60 ppm zirconium oxide, 50MO50PO + 80 ppm zirconium oxide, and diesel. The zirconium oxide is a novel catalyst for boosting engine performance, improving combustion rates, and reducing emission formations. The method used for proportions is the ultra-sonification process. The results were good in terms of superior brake thermal efficiency for the blend 50D25MO-25PO+ 20 ppm zirconium oxide is 34.2%, which is 11.67% greater than diesel, Brake specific fuel consumption is 0.20 kg/kWh for the blend 50D25MO25PO+ 20 ppm Zirconium oxide 31.43 % less than diesel, Maximum heat release rate is reached for the blend 80D10MO10PO + 60 ppm zirconium oxide followed by 89 KJ/CA with highest peak pressures of 69 bar for blend 70D15MO15PO + 40 ppm zirconium oxide. 70D15MO15PO + 40 ppm zirconium oxide blend offers very few Hydro carbon emissions, followed by 77.7% than diesel; the least Carbon monoxide formations are found for the same blend, followed by 40% less than diesel. 70D15MO15PO + 40 ppm zirconium oxide blend possesses very low carbon dioxide with a marginal difference of 5.6% than diesel, and lower nitrogen emissions are 50MO50PO + 80 ppm zirconium oxide found to be 700 ppm.

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INTRODUCTION

The generation of emissions in diesel-based vehicles is gradually increasing in the automotive sector. The thrust beyond the usage of automobiles is also growing, and the usage of automobile research in engine parts and emissions parts is also increasing [1]. Although several researchers focus on reducing emissions strategies by incorporating the latest emission techniques, there is a slight modification in engines, increasing preventive maintenance costs every year. This results in an increase in automobile costs by 5 to 10% per year [2]. The latest engine modifications require the highest specific fuel consumption and provide high sophistication and driving comfort. The people from the middle background did not get the latest modified vehicles. Hence, the researchers are working on reducing fuel costs and introducing alternative fuels with reduced emission techniques [3]. Alternative fuels such as blends extracted from cotton, soya beans, palm seeds, sunflower seeds, jatropha seeds, and nonedible seed oil are investigated in the automobile engine industry [4]. The main merit of these nonedible blends is that improvised brake thermal efficiency is found when the engine operates at optimal workload conditions [5]. It is also noted that few pure alternative fuels operated on diesel engines have enormous benefits regarding smooth running and the lowest possible emissions [6]. The major drawback that persists owing to the combustion of pure biodiesel blends is a higher heat release rate, which results in deteriorating the piston and cylinder-associated parts [7]. The recent research on Al_2O_3 coatings by modified piston crown with plasma arc process improves the brake thermal efficiency (BTE) by 0.8% and Brake-specific fuel consumption by 0.5 %, and emissions are also reduced by 2 % compared to CE engines [4]. The thermal barrier coating is applied on palm oil biodiesel and diesel, and they investigated different blend ratio combinations from B5 to B25. From the results, it is observed that there is a decrease in emission rate by 1.5 % for coated engines as compared to CE engines [8]. Increasing compression ratios from 16:1 to 18:1 for coated and uncoated engines maximizes the improvement by 2% higher for coated engines with less compression ratio of 16:1, and BSFC was achieved by 4% as compared to uncoated engines [9]. The combination of silicon with aluminium signifies improvement for BSFC and BTE, and a reduced emissions rate of 6% compared to diesel [10]. Adding Al_2O_3 and YSZ powder improves the BTE by 2% and reduces the BSFC by 1.4% compared to the CE engine [11]. Utilizing thermal barrier coatings leads to an improvement in the efficiency of 2.3%, reduced BSFC by 1.9%, and H.S.U by 1.9% in maximum load conditions operated by diesel engines [12]. The addition of coatings on the piston liner improves the engine efficiency by 1.06% compared with the CE engine, and the addition of liner coating in the range of 100 μm operates the engine at higher temperatures [13]. The introduction of magnesium alloys as an insulating material gives optimal performance in the range of 30 % as a BTE and less in BSFC as

compared to a CE engine. MgZrO_3 and YSZ coating is done at self-governed engine components such as engine combustion chamber piston, heads, and liner, and from the results, it is observed that MgZrO_3 possesses the best ceramic coating to attain complete combustion results in 10.5% more than compression work has been developed [14]. The seed within the mangoes (*Mangifera indica*) has been removed using the Soxhlet treatment solvent procedure and subsequently turned into biofuels using transesterification. It entailed the chemical reaction of the oil obtained with alcohol at a perfect temperature of sixty degrees Celsius, utilizing a catalyst dosage of one percentage point w/v (KOH) to achieve maximum biofuel output [15]. Mango (the *Mangifera indica*) fruit bio-oil is trans-esterified with methanol using hydroxide as an accelerator in this study, particularly investigating the effect of how much catalyst is used on the process. At the ideal ambient temperature of fifty degrees Celsius, a weight content of eighty-five percent was achieved. Similarly, when considering the effect of reagent amounts, an 85%the weight solution containing ten millilitres of zinc oxide nano catalyst was found to be optimum [16]. The following mixtures are made using magnetic stirring: at twenty percent, diesel, and all measured on a volume basis. Each of these mixes is called 5% mango biodiesel, 10% mango biodiesel and 15% mango biodiesel, 20% mango biodiesel [17]. Compared to MSB20B0, experimental results show that MSB20B5 reduces the BSFC percentage by 27.22 per cent while raising the amount of BTE by 8.94 per cent when loaded with full capacity and getting close to the diesel fuel level. Furthermore, while comparing the 5% mango biodiesel blend to the 20% mango biodiesel blend at optimum launching, the former cuts output contaminants by 9.21 % and the latter by 18.22 %, the former by 9.92 percent, and the third by thirteen percent [18]. All things considered, diesel engines benefit from the better efficiency and less pollution provided by the MSB20B5 multimodal mixture [19]. According to the findings, the introduction of five percent decanol to small and medium 20 per cent diesel substantially boosted braking thermal performance by four percent and significantly reduced emission levels of HC, CO, and smoke. At maximum capacity, nonetheless, NO_x emissions rise slightly [20]. Hence, in this research, the mango seed blends are utilized to reduce the viscosities and improve the desired qualities of the blends offered by the different proportions. The effects of mango seeds blended with pyrolysis oil, and the addition of zirconium catalysts have potential benefits in terms of improved engine performance followed by reduced emissions and better combustion characteristics. The research gap found from past literature is mixing mango biodiesel with pyrolysis oil owing to different proportions of zirconium oxide catalysts. The novelty in this work is many researchers tried the replacement of diesel with other bio-fuels, and some combustion studies and results were compared, Here in this research, the proportions of mixing for mango seed blends and pyrolysis blends have potential benefits, and also the addition of zirconium catalysts boosts the performance, boosts the combustion rates and reduces the

emissions levels of the engine. Hence this motivation study helps several researchers work with different blends with different proportions and different types of catalysts. The choice of the catalyst depends upon its morphological structures and the costs. This work is very interesting to refer to the researchers who work on biofuels with wide applications in the automobile industry. The motivation of this study is to analyze the combination of pyrolysis oil with mango seeds oil. Since pyrolysis oil has greater demerits in terms of pungent colour and high density that results from heavy vibrations, This property of reducing the viscosities for reducing the vibration of the engine has been happening with the blending of different blends. This work aims to test the different types of blends with the presence of zirconium coatings powered with mango seeds and pyrolysis oil combinations for improving engine performance and combustion levels. The other objective of this work is to reduce the emission of particulates from diesel engines powered with zirconium coatings powered with mango seeds and pyrolysis oil combinations, which improves the maintenance of peak loads.

MATERIALS AND METHODS

Materials

The materials used in this analysis are Mango seed biodiesel, purchased for RS 500 per litre at a local biodiesel company in Andhra Pradesh, Hyderabad. Pure diesel is purchased at RS 100 per litre. The Zr_2O_3 nanocatalyst is purchased at the Chennai dealer's price of RS 250 per kilogram. The pyrolysis oil is purchased from Andhra Pradesh for RS 500 per litre.

Methods

The proportions such as 50D25MO25PO + 20 ppm Zr_2O_3 , 70D15MO15PO + 40 ppm Zr_2O_3 , 80D10MO10PO + 60 ppm Zr_2O_3 , 50MO50PO + 80 ppm Zr_2O_3 , and 100D. The proportions of Zirconium catalyst with the pyrolysis and mango biodiesel blends are done with the help of ultrasonication method process with time interval of 10 minutes

for heating and solidification process. The mechanical stirrer rotating at 120 rpm is constantly worked on to mix this Nano catalyst with biodiesel blends properly. After this process is completed, it should be cooled for one hour and transesterified with the help of KOH and 5% ethanol to remove the impurities in the form of glycerine and dust. The final step is removing the dust particles with the help of a cotton filter and keeping it free for 8 hours before the experiment is started. Table 1 represents the properties of blends.

Experiment Setup

The Schematic experimental setup, as shown in Figure 1, defines the various components like the air supply system, exhaust system, fuel supply system, and loading arrangement setup. The experiment was conducted on a conventional engine, and the results of the performance, combustion, and emissions were recorded [21]. Again, the test will be conducted, and all the necessary results will be recorded. All three experimental results will be studied and compared to find the best performance and lowest emission values. The engine is a Kirloskar single-cylinder, water-cooled direct injection diesel engine [22].

Figure 2 shows the actual experimental setup used to conduct the analysis. The engine is operated by well-trained labour to carry out all the experiments more smoothly. The engine specifications are also mentioned in Table 2. The accuracy of the results depends upon the accuracy of the instruments used and test specifications in errors. Hence, the engine is cleaned well before the experimentation, which takes two to three hours. The engine trial and report are carried out to check the accuracy of the results.

Uncertainty Analysis

The important term that determines the least possible errors from the calibration instruments and the nature of the atmospheric defects caused by these defects is measured with the help of uncertainty analysis [23]. The experiments will be carried out at least three times to check the accurate results. Through uncertainty analysis, the instruments used

Table 1. Properties of blends

PROPERTIES	UNITS	DIESEL ASTM D975	BIO DIESEL ASTM D6751	D100	50D25MO25PO + 20 ppm Zr_2O_3	70D15MO15PO + 40 ppm Zr_2O_3	80D10MO10PO + 60 ppm Zr_2O_3	50MO50PO + 80 ppm Zr_2O_3
Density	kg/m ³	800-867	800-860	818	848	830	821	798
HHV	kJ/kg	44118	44910	43910	44979	42191	42245	39523
Viscosity	cs	1-4.5	1 -4.5	2.79	3.21	2.88	1.94	1.24
LHV	Kj/kg	43005	43124	42515	43422	40181	39441	38355
Flash point	°C	49-110	42-63	38	37	32	31	34
Fire point	°C	41-315	46-210	60	53	46	42	38
Cetane	-	43-65	46-55	67	59	58	53	42

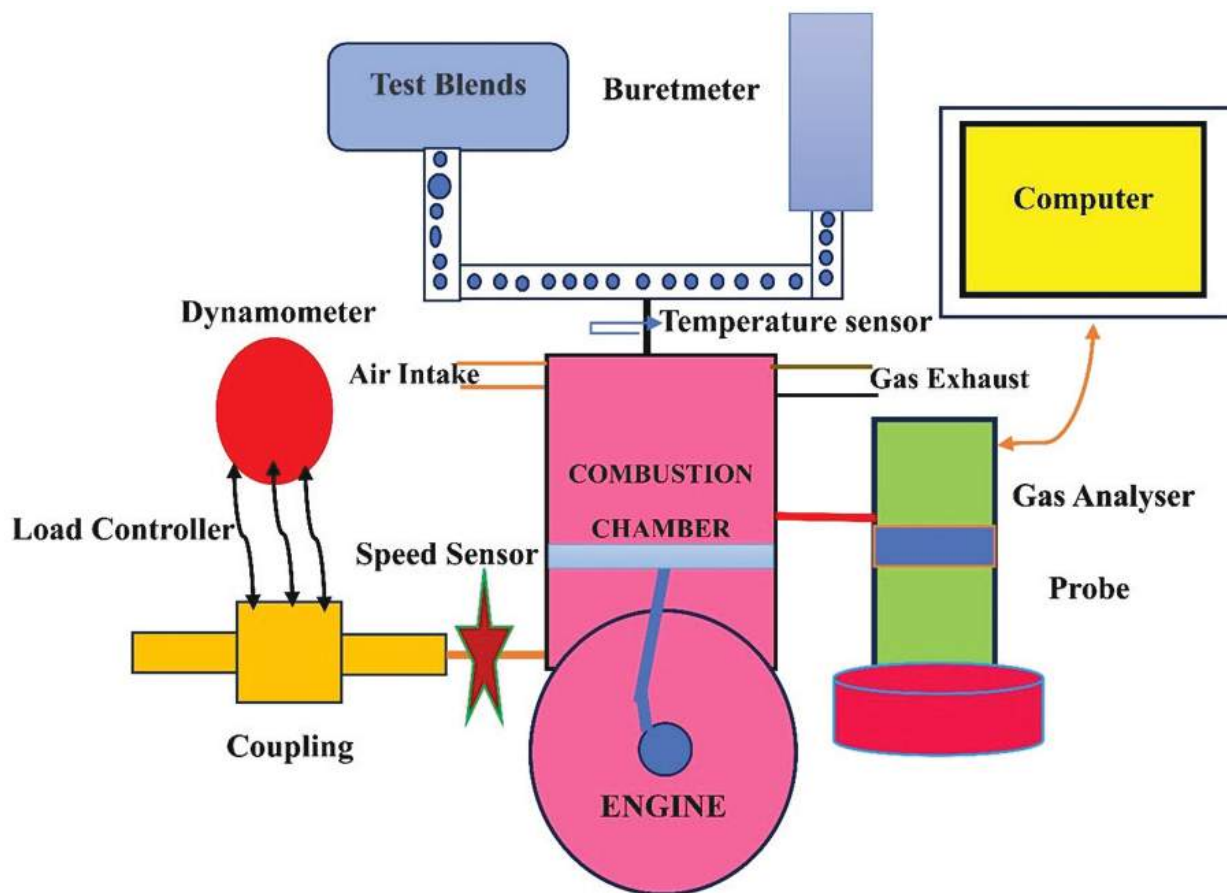


Figure 1. Schematic experiment set up.



Figure 2. Actual experimental setup.

Table 2. Engine specifications

S.no	Type of description	Make of specifications
1	Strokes type	4 stroke
2	Injection type	Direct injection
3	Compression ratio	18.5:1
4	length -Rod (connecting)	238 mm
5	The radius of the Crank	70 mm
6	Cubic capability Capacity	662 cc
7	Aspiration	Natural
8	Power	5.2 kW @ 1550 rpm
9	Bore & Stroke	88.5 & 110 mm

in the experiment were thoroughly inspected for repeatability to confirm the final results [24]. Table 3 shows the accuracy of the different instruments used.

RESULTS AND DISCUSSION

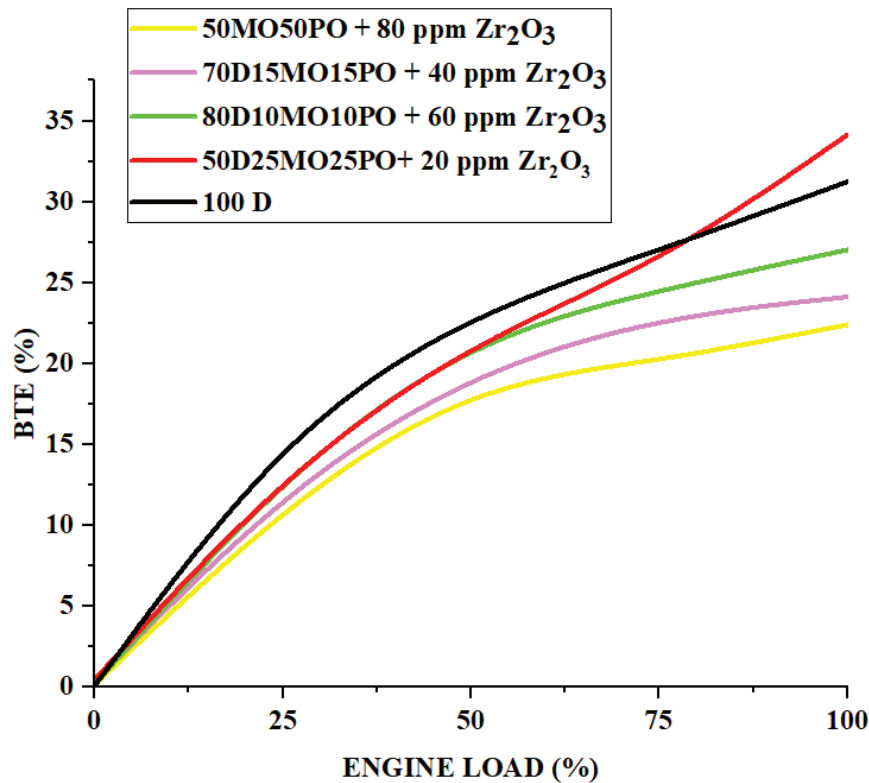
Brake Thermal Efficiency

The power generated from the fuel supplied is called BTE. In other words, the engine performance can be measured by the maximum power generated for the given amount, called BTE [25].

An engine's thermal efficiency of brakes is a measure or ratio indicating the actual brake horsepower achieved

Table 3. Measure of the uncertainty of data

S.no	Instrument	Accuracy	Range	Uncertainty %
1	Smoke meter	± 0.1	0-10 BSU	1
2	Stopwatch	$\pm 0.5s$	-	0.2
3	Manometer	± 2 mm	0-210 mm	1.1
4	Tachometer	± 10 rpm	0-11000 rev/min	0.11
5	Temperature of Exhaust gas	± 1 °C	0-900 °C	0.12
6	The encoder of Crank angle	± 1 °deg	250-420 deg	0.13
7	Loading indicator	± 2 kg	0-100 kg	0.2

**Figure 3.** Brake thermal efficiency.

throughout an engine's whole conversion of the highest amount of heat the combustion product delivered [26]. Figure 3 illustrates the brake thermal efficiency of the engine load. 50D25MO25PO+ 20 ppm Zr_2O_3 exhibited the highest brake thermal efficiency of 34.2% compared to all the remaining blends. This is owing to the superior calorific value offered by 50D25MO25PO+ 20 ppm Zr_2O_3 , and it was seen that 50MO50PO + 80 ppm Zr_2O_3 fuel attained the lowest BTE followed by (25.5%) due to lower calorific values; this results from the presence of elevated oxygenation concentration and a substantial calorific value [27]. The cheapest brake thermal efficiency found for 50MO50PO + 80 ppm Zr_2O_3 was recorded to possess the lowest calorific value compared to the other blends. Consequently, the 50D25MO25PO+ 20 ppm Zr_2O_3 mix exhibits superior brake thermal efficiency compared to other blends [28]. The higher calorific value leads to a higher brake thermal efficiency. The Zirconium oxide boosts the properties of the blends and acts as the best accelerator for attaining the highest efficiency [29].

Brake Specific Fuel Consumption

The actual energy use of brake power is determined by brake-specific fuel usage. The proportion represents the required fuel consumption for each watt of braking force [30]. Figure 4 illustrates the relationship between individual fuel usage and engine load. The lowest brake specific fuel consumption (BSFC) is seen with 50D25MO25PO+ 20 ppm Zr_2O_3 at full capacity, followed by 0.20 kg/kWh because of

its superior calorific value, resulting in enhanced output for the specified fuel quantity. Following 50D25MO25PO+ 20 ppm Zr_2O_3 , the fuel model exhibited the lowest brake-specific fuel consumption (BSFC), being 31.43 % less than diesel because zirconium acts as good boosting catalysts to improve the calorific values of the blends [31]. These results are because of superior economic power generation per kW/h achieved by utilizing a minimal amount of fuel for 50D25MO25PO+ 20 ppm Zr_2O_3 compared to diesel [32]. The lowest calorific value for other blends causes the highest fuel consumption in full-load conditions [33].

COMBUSTION CHARACTERISTICS

Inline Cylinder Pressures

The very important factor that describes the performance of the engine owing to the combustion parameters is significantly described by the term cylinder pressures. This parameter defines the working phenomenon of the gases subjected to different trials [34]. Figure 5 expresses the results of mango seed blends and pyrolysis oil subjected to adding zirconium catalysts. The highest inline cylinder pressures are attained for the blend 69 bar for blend 70D15MO15PO + 40 ppm Zr_2O_3 . Because of the good cetane number offered by the blends, it quickly evaporates and achieves the combustions subjected to peak pressures [35]. The other blends possess the lowest

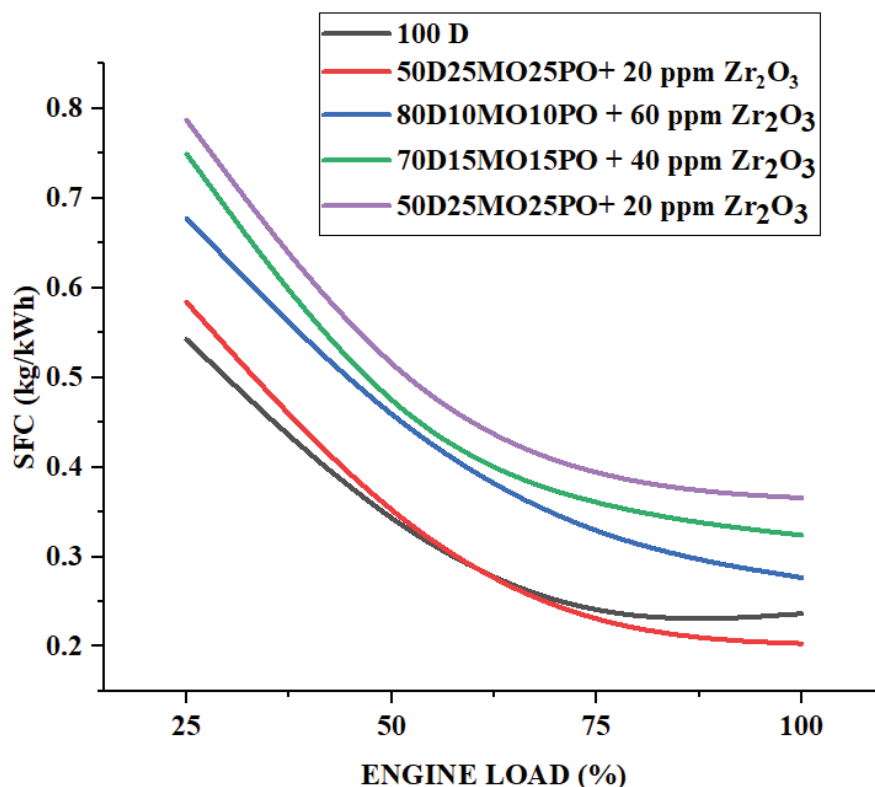


Figure 4. Specific Fuel Consumption.

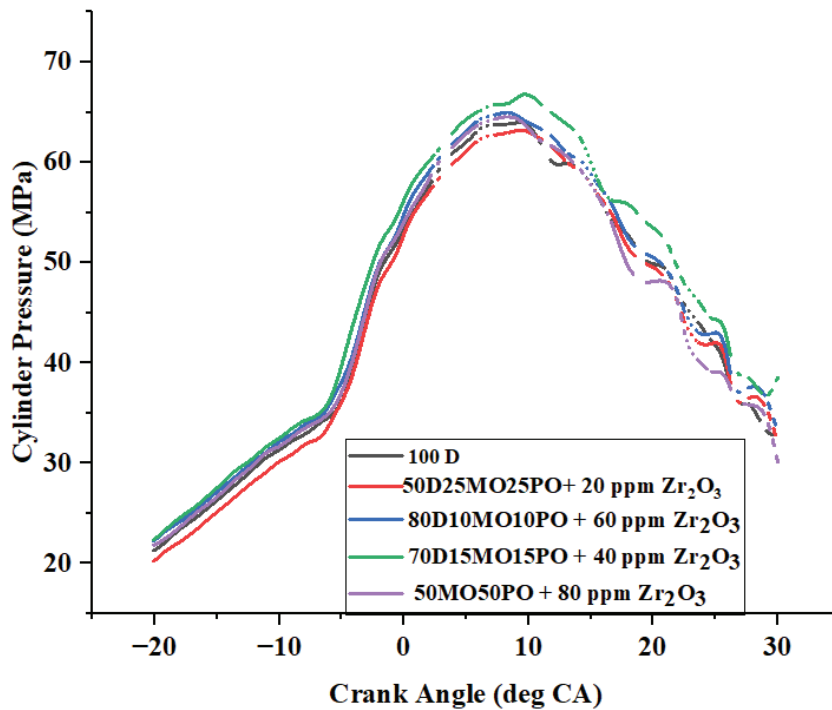


Figure 5. Cylinder pressures.

inline cylinder pressures compared to 70D15MO15PO + 40 ppm Zr_2O_3 and other blends because of the low surface tensions offered for these blends, resulting in low cylinder pressures at elevated pressures and temperatures [36]. The attainment of the lowest cylinder pressures concerning

different rpm tends to have the least performance for the lowest pressure rise blends compared to peak pressure rising blends [37].

Heat Release Rate

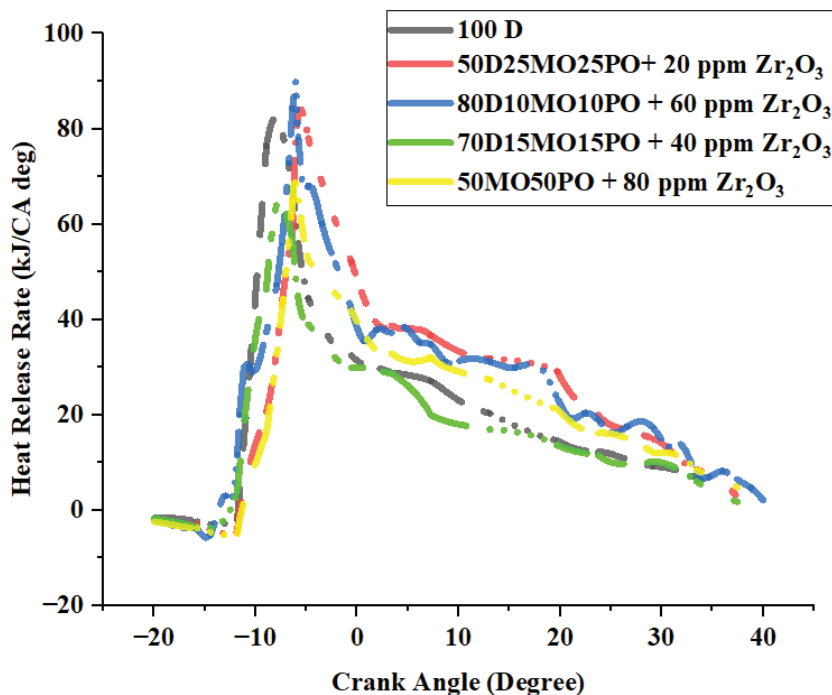


Figure 6. Heat release rates.

The amount of gases evolved during the different blends' peak and moderate loads. When the crank is subjected to different operating pressures and temperatures, it is technically expressed by heat release rates [38]. This term significantly determines the heat-evolving phenomenon for evaluating the combustion rates. Figure 6 demonstrates the Heat release rates offered by different blends. The maximum heat release occurred for the blend 80D10MO10PO + 60 ppm Zr_2O_3 , followed by 89 KJ/CA. The phenomenon of this blend occurring with the highest HRR is subjecting to higher surface tensions. Very short ignition delays offered by the blends tend to easy atomization and quicker releasing than other blends [39]. The lower the surface tensions, the lower the heat release rates, resulting in lower performance in terms of efficiency and higher consumption of the fuel [40].

EMISSIONS

CO Emissions

The result of improper combustion is improperly mixing carbon particles with oxygen particulates, which results in peak loads and CO emissions [41]. The continuous emissions of these particulates seriously affect the lungs, and almost 43.5% of people suffer from CO emissions. Figure 7 expresses the CO emissions from different

blends used in this analysis. The effect of blend followed by 70D15MO15PO + 40 ppm Zr_2O_3 has significantly less potential at 0.12% and for diesel at 0.18%; the decrease in CO formations is found for the blend 70D15MO15PO + 40 ppm Zr_2O_3 followed by 40% less than diesel. The reason is that adding 40 ppm Zr_2O_3 catalysts potentially decreases density, especially in pyrolysis oil and mango seed blends [42]. This reduced viscosity offers better combustion than diesel and offers low co-emission particulates [43]. The reason for achieving the lowest carbon emissions of all blends compared to diesel is that the catalyst zirconium blends act as the best reagent to react to combustion quickly with available oxygen and emitting particles that are very low compared to diesel [44].

CO₂ Emissions

The combustion occurs due to the inadequate mixing of carbon and oxygen proportions offered by different air-fuel ratios raised by the different blends subjected to load level and engine speed, called carbon dioxide proportions [45]. Figure 8 neatly expresses the CO₂ emissions offered by different blends. The lowest CO₂ emitted for the blend is 70D15MO15PO + 40 ppm Zr_2O_3 blends, followed by 5.6%, but for diesel, the emissions rates are 6.5% at peak loads. The difference in emissions is found by 14.78% for the blend 70D15MO15PO + 40 ppm Zr_2O_3 because the higher ignition delay offered by the blends tends to have

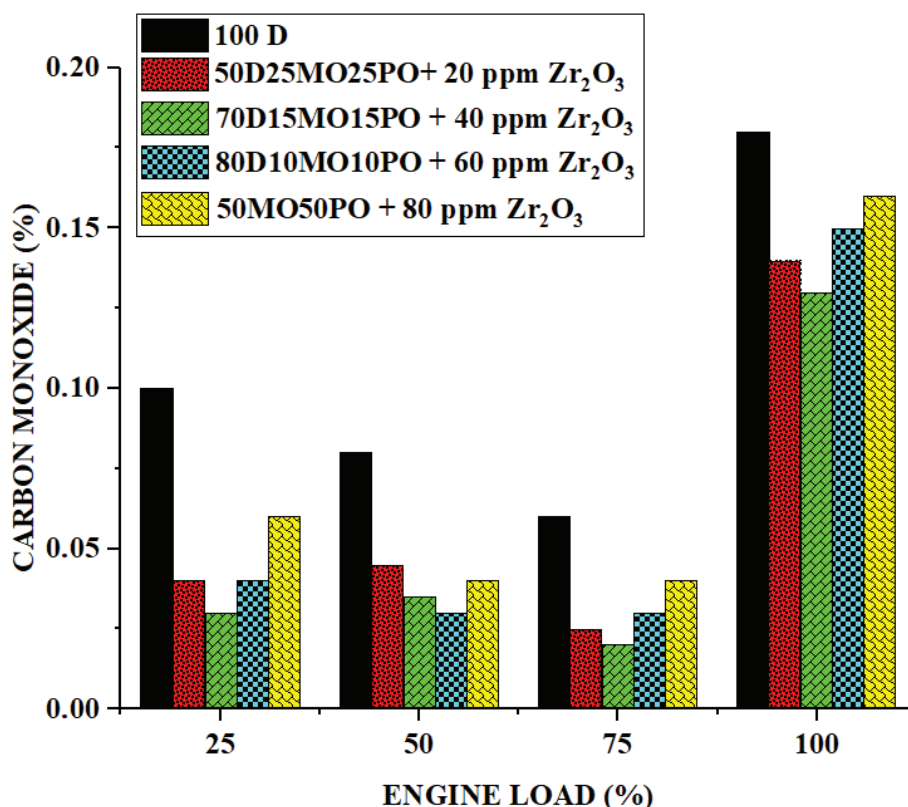


Figure 7. CO emissions.

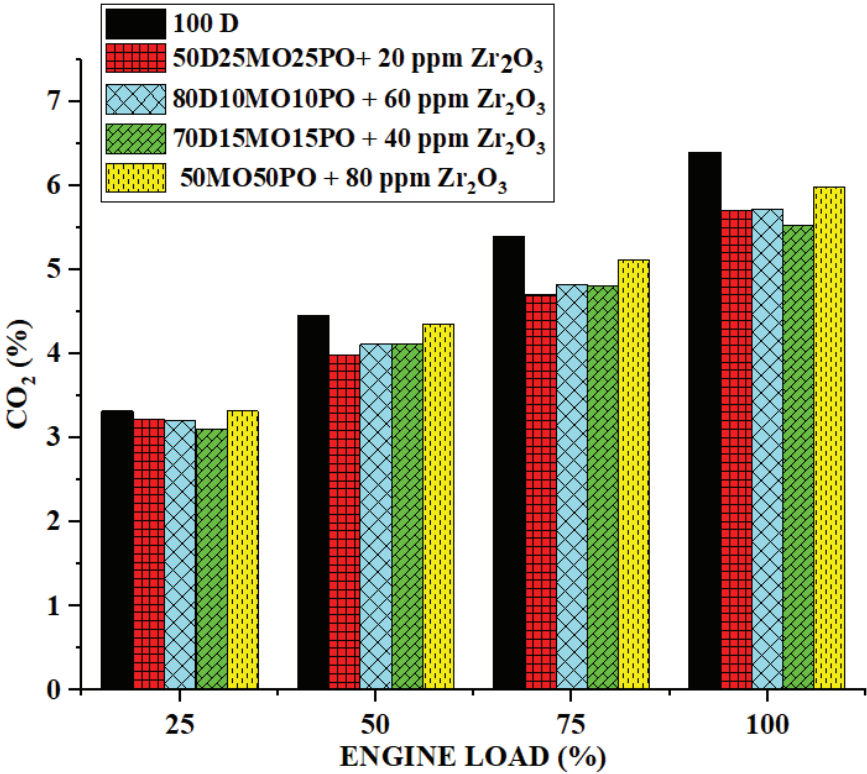


Figure 8. CO₂ emissions.

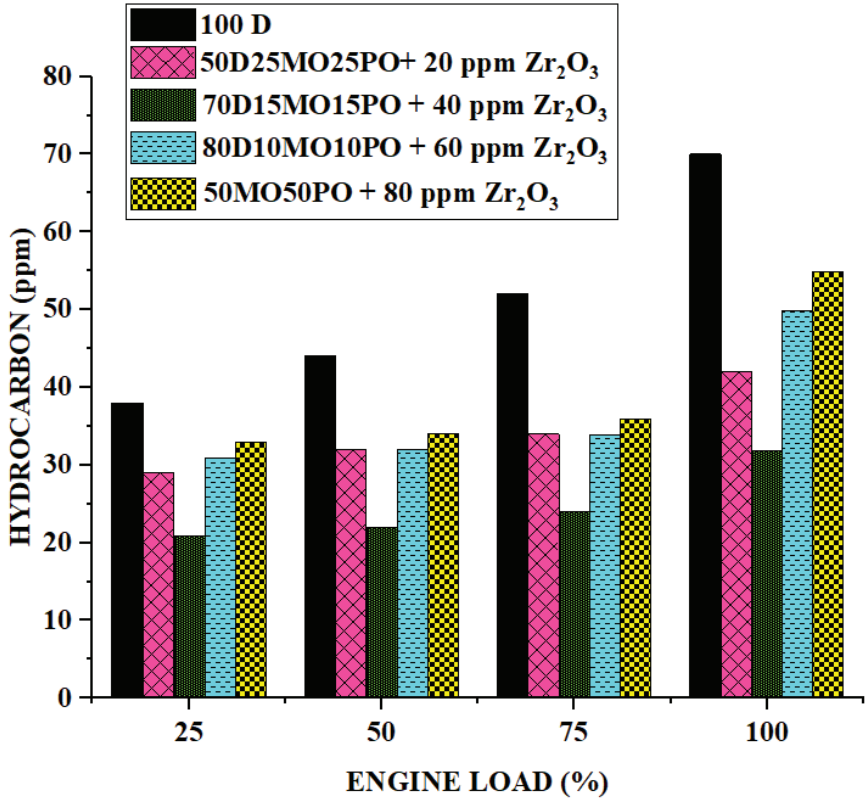


Figure 9. Hydro carbon emissions.

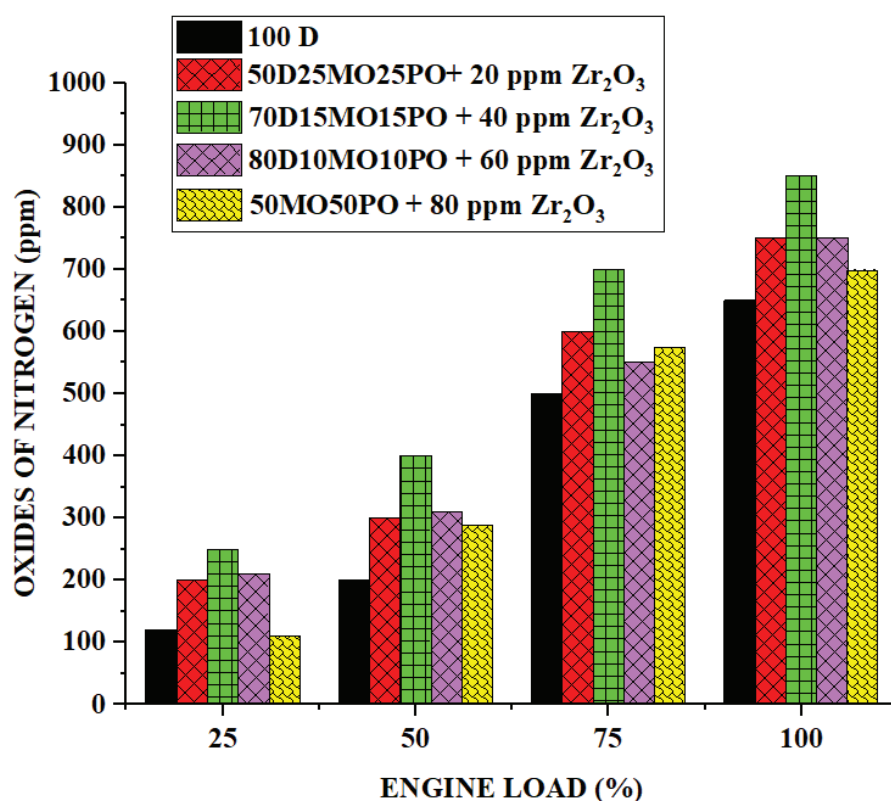


Figure 10. NO_x emissions.

more time for oxidizing the carbon atoms with the oxygen atoms, resulting in very least emissions than diesel [46]. The incomplete combustion due to inadequate proportions of oxygen supply emits the particles significantly faster and faster for the different blends than diesel [47].

HC Emissions

The particulates emitted continuously due to inadequate compilation of hydrogen molecules with the carbon molecules owing to the engine piston travelling from BDC to TDC at elevated pressures and temperatures are called hydrocarbon emissions [48]. Figure 9 represents hydrocarbon emissions emitted by the various blends combined with mango seeds and pyrolysis oil. The effect of HC emissions for the blend 70D15MO15PO + 40 ppm Zr₂O₃ is much less effective than diesel. It emits 31 ppm, but diesel emits 70 ppm. The 70D15MO15PO + 40 ppm Zr₂O₃ blend offers very few emissions, followed by 77.7% more than diesel, because of the superior oxidation offered by 40 ppm Zr₂O₃ catalysts to evaporate and atomize the blends very superior way than diesel [49]. Adding the zirconium catalyst with higher proportions up to 60 ppm pushes the combustion acceleration rate to achieve the lowest emissions, and the presence of sufficient oxygenated atoms in mango and pyrolysis blends emits the lowest particulates than diesel [50].

NO_x Emissions

The continuous emits of unburnt gases, which are very harmful to the environment, released in the temperature range of 1200 °C to 1300 °C, are called Nitrogen emissions. In other words, at the tail end pipe, the gases that evolved due to incomplete combustion in the temperature range of 1200°C to 1300°C are called NO_x emissions [51]. These emissions in minute forms will cause serious issues for very older adults, causing eye failures. Figure 10. illustrates the NO_x emissions released from the different blends used in this analysis. From the results, it is seen that 50MO50PO + 80 ppm Zr₂O₃ possesses the lowest NO_x emissions, followed by 700 ppm; the reason this blend has lower emissions compared to other blends is lowest viscosity offered by the blend causes it to oxidize and evaporate the blend, in addition to the catalyst of Zr₂O₃ reduces the viscosity for oxidizing the fuel and air very quickly combustion than other blends [52]. The higher temperature formations due to more ignition delay cause the rapid emission formation of nitrogen particles for all the blends compared to diesel.

CONCLUSION

The suitable proportions of mango pyrolysis oil concerned with adding zirconium catalyst with different proportions have been done. The analysis of performance, combustion, and particulate matter on single-cylinder

diesel engines has been studied thoroughly with the help of mango and pyrolysis blended with zirconium coatings. The addition of zirconium concentrations from 20, 40, 60, and 80 ppm increases the blends' density, affecting the performance and combustion characteristics. The following results were achieved by these test blends followed by

- ❖ 50D25MO25PO+ 20 ppm Zr_2O_3 exhibited the highest brake thermal efficiency of 34.2% compared to all the remaining blends. This is owing to the superior calorific value offered by 50D25MO25PO+ 20 ppm Zr_2O_3 , and it was seen that 50MO50PO + 80 ppm Zr_2O_3 fuel attained the lowest BTE followed by (25.5%) due to lower calorific values.
- ❖ The lowest Brake Specific Fuel Consumption (BSFC) is seen with 50D25MO25PO+ 20 ppm Zr_2O_3 at full capacity followed by 0.20 kg/kWh, because of its superior calorific value, resulting in enhanced output for the specified fuel quantity. Following 50D25MO25PO+ 20 ppm Zr_2O_3 , fuel model exhibited the lowest brake-specific fuel consumption (BSFC), being 31.43 % less than diesel.
- ❖ The highest inline cylinder pressures are attained for the blend 69 bar for blend 70D15MO15PO + 40 ppm Zr_2O_3 . Because of the good cetane number offered by the blends, it tends to evaporate and achieve the combustions subjected to peak pressures easily
- ❖ The maximum heat release occurred for the blend 80D10MO10PO + 60 ppm Zr_2O_3 , followed by 89 KJ/CA. The phenomenon for this blend occurring the highest HRR is subjecting to higher surface tensions and very short ignition delays offered by the blends tend to easy atomization and quicker releasing than other blends
- ❖ 70D15MO15PO + 40 ppm Zr_2O_3 has significantly less potential at 0.12%, and for diesel at 0.18%; the decrease in CO formations is found for the blend 70D15MO15PO + 40 ppm Zr_2O_3 followed by 40% less than diesel.
- ❖ The lowest CO_2 emitted for the blend is 70D15MO15PO + 40 ppm Zr_2O_3 blends, followed by 5.6%. However, for diesel, the emissions rates are found to be 6.5% at peak loads. The difference in emissions is 14.78% for the blend 70D15MO15PO + 40 ppm Zr_2O_3 because of the higher ignition delay offered by the blends.
- ❖ The 70D15MO15PO + 40 ppm Zr_2O_3 blend offers fewer HC emissions, followed by 77.7%, than diesel. Because of the superior oxidation offered by 40 ppm Zr_2O_3 catalysts, the blends evaporate and atomize more efficiently than diesel.
- ❖ 50MO50PO + 80 ppm Zr_2O_3 possesses the lowest NO_x emissions, followed by 700 ppm. This blend's lower emissions are due to the lowest viscosity offered, which causes it to oxidize and evaporate.

Future Recommendation

The authors were trying to include other additives, such as decanal and titanium oxide coatings, to determine the

performance, combustion, and emission parameters. They were also planning to do work on varying compression ratios. The authors were also willing to do tests concerning propanol as an alcohol with pyrolysis + mango and palm blends with the latest nano additives followed by various proportions.

NOMENCLATURE

BDC	Bottom dead centre
BTE	Brake thermal efficiency
BSFC	Brake specific fuel consumption
TDC	Top dead centre
CA	Crank angle
CE	Conventional Engine
CO	Carbon oxide
CO_2	Carbon dioxide
HC	Hydro carbons
PPM	Parts per minute
Zr_2O_3	Zirconium oxide
50MO50PO	50% Mango oil + 50% Pyrolysis oil
50D25MO25PO	50% Diesel + 25% Mango oil + 25% Pyrolysis oil
70D15MO15PO	70% Diesel + 15% Mango oil + 15% pyrolysis oil
80D10MO10PO	80% Diesel + 10% Mango oil + 10% pyrolysis oil
NO_x	Nitrogen dioxide

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.

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Research Article

Energy and exergy analysis of a novel solar-biomass tri-evaporator multigeneration plant integrated with fuel cell and electrolyzer

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ABSTRACT

This study presents the development and thermodynamic assessment of an environmentally benign multigeneration system utilizing solar and biomass resources. The system integrates parabolic trough collectors (PTC), biomass gasification, Organic Rankine Cycle (ORC), Kalina Cycle (KC), vapor absorption system (VAS), electrolyzer, and fuel cell to produce hydrogen, electricity, heating, cooling, and hot water. Thermodynamic analysis is conducted using the first and second laws, focusing on energy and exergy performance. The system is simulated using engineering equation solver (EES), with R245fa and an ammonia–water mixture as working fluids in the ORC and KC, respectively. The proposed system consists of PTC and biomass capacities of 1295 kW and 553.6 kW, respectively, and generates a total power output of 964 kW, including 276.03 kW from the ORC and 145.28 kW from the KC. Post-expansion cooling from the KC provides 129.6 kW at –33.5 °C with 12.15 kW of exergy, while the VAS yields 18.4 kW at 5 °C with 0.24 kW of exergy. The PTC supplies 311.4 kW and 683.8 kW of thermal input to the ORC and KC, respectively. Overall, the system attains energy and exergy efficiencies of 52.2% and 16.14%. The study demonstrates the feasibility and sustainability of integrating solar energy into biomass gasification, supporting its potential for clean energy generation.

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INTRODUCTION

The continuous rise in global energy demand, closely linked to national development, has intensified the urgency to transition from conventional fossil-based power systems to renewable energy solutions [1,2]. This shift is driven by

a combination of factors, including climate change, greenhouse gas (GHG) emissions, fossil fuel depletion, and volatile energy prices [3]. Renewable energy sources such as solar, wind, biomass, hydropower, and geothermal have emerged as sustainable alternatives, offering the potential to

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reduce carbon emissions, mitigate environmental degradation, and enhance energy security through resource diversification [4]. In this context, multigeneration systems also referred to as polygeneration systems represent a progressive evolution in energy conversion technologies. Unlike conventional cogeneration or trigeneration systems that typically produce electricity and heating/cooling, multigeneration systems can simultaneously deliver electricity, thermal energy, desalinated water, hydrogen, cooling, and other by-products. These systems enhance overall efficiency by utilizing waste heat and integrating multiple energy outputs within a unified configuration, thereby minimizing fuel consumption and improving sustainability metrics.

Recent global trends underscore the momentum behind renewable energy and hydrogen technologies. By the end of 2024, global concentrated solar power (CSP) capacity reached approximately 6.8 GW, while global bioenergy capacity rose to 150.8 GW, with electrical capacity alone contributing 83.8 GWel across nearly 4,971 operating biomass power plants [5,6]. Electrolytic hydrogen production capacity has surpassed 5 GW, and global hydrogen output is projected to reach 97 million tons in 2024. These figures reflect the expanding role of renewables in supporting clean, integrated energy systems. Solar energy, characterized by its wide availability and scalability, continues to be a cornerstone of the energy transition [7]. Its integration with other renewables, such as biomass, allows for hybrid configurations that offer complementary benefits. Biomass, derived from organic materials like forestry residues and municipal waste, provides a stable energy supply that compensates for the intermittency of solar power [8]. The efficiency of biomass gasification is influenced by factors such as feedstock type, particle size, and operational parameters. Studies have explored related phenomena in solid fuel combustion and transformation, including unburned carbon dynamics [9], briquetting techniques [10], tar characterization [11], grindability indices [12], and grinding energy optimization [13]. Hybrid renewable systems that couple solar and biomass sources offer improved system reliability and resource flexibility. Advanced thermodynamic cycles such as the Organic Rankine Cycle (ORC) and Kalina Cycle (KC) have proven effective for converting low- and medium-temperature heat from these sources into useful energy [14]. ORC systems, in particular, excel in recovering waste heat from industrial and renewable sources, contributing to both energy efficiency and environmental performance [15]. For instance, a solar-biomass hybrid system using Stirling engines and anaerobic digesters was shown to generate 5.6 GWh/year while reducing CO₂ emissions by 7.8 kton/year, with an energy cost of \$0.075/kWh [16].

Numerous studies have modeled and optimized hybrid renewable systems to enhance energy resilience. In Libya, [17] used the System Advisor Model (SAM) to evaluate a PV/wind system with pumped hydro storage, demonstrating reliable load coverage and minimal levelized cost of energy (LCOE). In Gaza, [18] employed HOMER-Pro to

optimize a PV-wind-diesel system for a COVID-19 quarantine center, achieving 100% load reliability and reduced emissions. Similarly, [19] explored a hybrid wind-Stirling dish system for Hurghada, Egypt, and [20] assessed a biogas-based system tailored to the terrain and waste profile of Al-Jabal Al-Gharbi, achieving 86% CO₂ reduction with an LCOE of 3.5 ¢/kWh [21]. Hydrogen integration into multigeneration systems offers further decarbonization potential. As detailed by [22], hydrogen and fuel cell technologies can stabilize renewable energy systems by storing excess electricity and providing clean fuel for multiple end uses. Water electrolysis using surplus renewable energy is central to this approach, particularly in regions with variable solar and wind resources.

Other subsystems, such as Vapour Absorption Systems (VAS), contribute to sustainable energy applications by replacing mechanically intensive vapor compression systems with thermal-energy-driven alternatives [23]. VAS can utilize waste heat from industrial or renewable processes, improving system sustainability for cooling and heating loads. Additionally, Proton Exchange Membrane Fuel Cells (PEMFCs), operating in low (60–80°C) or high (120–200°C) temperature ranges, present efficient pathways for clean electricity generation from hydrogen [24]. Multigeneration systems have already been deployed in industrial sectors such as pulp and paper, salt, and district heating, particularly in Europe. For example, Finland and Sweden have pioneered the use of forest residues and roundwood for electricity and heat generation [25]. Meanwhile, several researchers have proposed novel configurations integrating multiple renewable inputs and thermodynamic cycles. [26] developed a solar-biomass-wind polygeneration system incorporating gasification, CO₂ capture, ammonia synthesis, and ORC technologies. [27–30] examined variations of solar-based multigeneration systems utilizing combinations of Kalina cycles, Brayton cycles, supercritical CO₂ cycles, and desalination technologies. Recent innovations include [31]’s solar tower system that integrates Brayton, transcritical, and organic Rankine cycles with PEM electrolysis and ammonia synthesis. The system produced 4938 kW of net power and demonstrated notable hydrogen and ammonia outputs with energy and exergy efficiencies of 26.71% and 26.16%, respectively. [32] reported a CSP-based system producing 3.62 MW of power and 0.0072 kg/s of hydrogen. Similarly, [33] evaluated a solar-geothermal hybrid system generating 56.6 kg/day of hydrogen and 1.47 kg/s of freshwater with enhanced power generation from thermoelectric integration. [34] compared basic and regenerative ORC cycles coupled with reverse osmosis, highlighting performance sensitivity to turbine and condenser pressure settings.

The following evaluation objectives were taken into consideration while conducting the above thorough literature review: the architecture of the multigeneration system, the system’s primary energy resources and how they are converted, the effective utilization of energy sources for

each subsystem, the method of producing hydrogen, the decarbonization of power production, the system, and its primary end products. It is found that a multigeneration energy system powered by a combination of biomass and solar energy sources with PTC system running on a hybrid nanofluid (MWCNT- Al_2O_3 / therminol VP1), incorporating and utilising the tri-evaporator system has not been investigated by integrating a unique ORC system combined with Kalina Cycle, Vapour absorption system, Proton exchange membrane, and a fuel cell, using minimal energy of the configured system to generate Electricity, Heating, Cooling, hot water, and Hydrogen. Furthermore, the performance of linked subsystems in such a multigeneration system must be investigated under various operational conditions. It is anticipated that these systems will open up several new possibilities for using biomass and solar energy sources.

Inspired by these gaps in the literature, this study proposes an integrated energy system that generates clean energy from renewable sources. A key innovation lies in the use of a single working fluid to drive three interconnected subsystems. By strategically bleeding the fluid during turbine expansion, the same medium is repurposed for cooling production. This integrated design shortens the thermodynamic process path, reduces system complexity, and minimizes the number of components required. As a result, it enhances overall performance, boosts energy utilization, and achieves a higher cooling-to-power ratio—an especially noteworthy aspect of this study. Furthermore, the paper develops models for an innovative cycle featuring a tri-evaporator system. This includes a conventional evaporator in the vapour absorption system (VAS) and unconventional evaporators in both the organic Rankine cycle (ORC) and Kalina cycle (KC) subsystems. The tri-evaporator configuration provides numerous benefits, such as increased cooling capacity, improved heat recovery, enhanced flexibility, reduced temperature differentials, greater system reliability, better heat transfer, and improved overall energy efficiency. By enabling heat recovery from multiple sources, minimizing thermal stress, and offering operational redundancy, this configuration represents a significant advancement in integrated energy systems.

The objectives of this research include the development of a new renewable energy-based integrated system capable of producing various utilities. To achieve this, the study involves a thorough investigation using thermodynamic balance equations for each system component, analyzing how state properties and varying operating conditions impact individual subunit performance as well as the system's overall performance.

In this paper, we further present a comprehensive analysis starting with the system description, including detailed insights into each component such as the parabolic trough collector (PTC), Kalina cycle (KC), organic Rankine cycle (ORC), vapour absorption system (VAS), proton exchange membrane (PEM) electrolyzer, biomass combustor, and a

fuel cell. Then the energy and exergy analyses, are discussed providing balance equations for different components to evaluate performance.

MATERIALS AND METHODS

Materials

The National Aeronautics and Space Administration (NASA) database provided daily meteorological data on solar irradiation for the Otuoke University community for 15 years. Furthermore, the biomass is woody waste of different wood species from tropical hardwood timber harvesting/logging in Otuoke. The average data served as the foundation for selecting simulation data for the planned multigenerational system and its components. Furthermore, the system modelling was conducted using Engineering Equation Solver (EES), and secondary data was sourced from publicly available literature and scientific sources.

System Assumptions

The whole system operated at steady state with negligible changes in potential and kinetic exergy. The ambient temperature and pressure at inlet to the turbine were taken as 25°C and 1.013 bar, unless otherwise indicated during parametric analysis. The remainder of the input data for simulation across the entire plant is summarised and presented in Table 9.

System Description

The proposed multigeneration architecture is shown in Figure 1. The system overview shows that the parabolic trough collector (PTC), powered by the sun, is the integrated energy system's primary energy source, the Kalina cycle (KC) provides power and cooling, the organic Rankine cycle (ORC) also provides power and cooling, the vapour absorption system (VAS) provides cooling, a proton exchange membrane (PEM) electrolyzer for hydrogen production powered by a biomass combustor and a fuel cell powered by hydrogen from the PEM electrolyzer. Starting with the PTC, a hybrid Nanofluid (Multi-walled carbon nanotube and Aluminium dioxide (MWCNT - Al_2O_3 / therminol-VPI) which is the heat transfer fluid of the subsystem, enters the solar collector as stream 54, and its energy is raised. The superheated exiting stream 52 from the PTC flows first into the vapour generator (VG for KC) for the Kalina cycle and transfers some of its thermal energy to the KC working fluid. Exiting as stream 53, it enters the vapour generator of the organic Rankine cycle (VG for ORC) and secondly transfers some more of its heat energy to the ORC working fluid. The hybrid Nano fluid returns to its initial state 54, and the cycle goes on continuously. Superheated steam-ammonia ($\text{NH}_3\text{-H}_2\text{O}$), the working fluid (stream 1) of the Kalina cycle leaves the vapour generator (VG for KC) and enters a separator (S1) and flashes into stream 2 and stream 3. Stream 2 is a

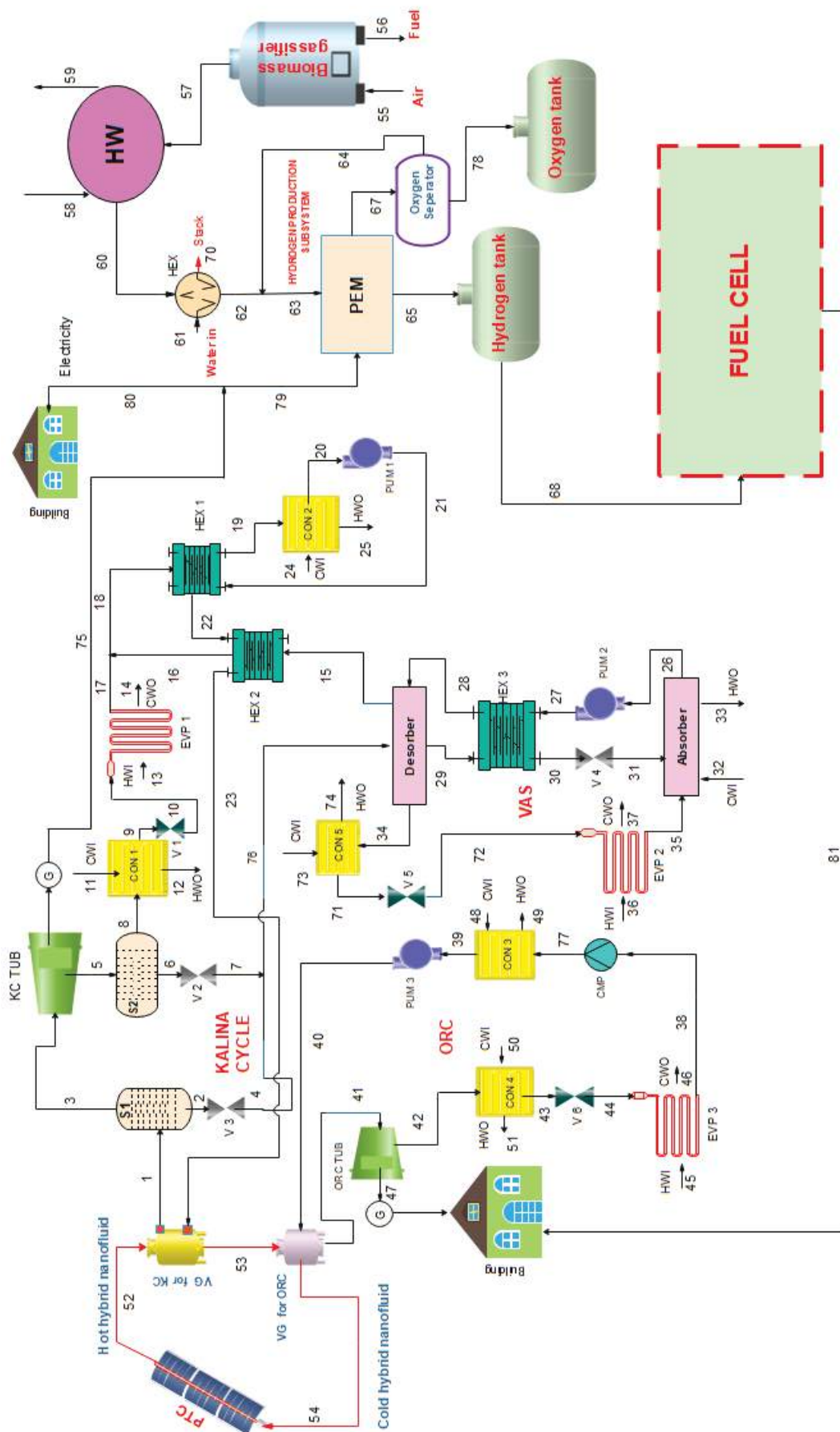


Figure 1. The proposed solar-biomass-based multigeneration system.

high-pressure, temperature condensate that flows through a throttling valve resulting in low-pressure stream 4. Stream 3, which is a high temperature and pressure steam ammonia passes and expands through the Kalina turbine converting some thermal energy into Mechanical shaft work that drives an alternator to generate electricity. The expanded vapour (stream 5) flows into a separator (S2) and splits into a lower-pressure condensate (stream 6), and another lower-pressure and temperature vapour (stream 8). Stream 6 is throttled into lower pressure stream 7 which mixing uniformly with stream 4 results in stream 76. Stream 8 is condensed into a lower temperature stream 9 by a high-pressure condenser (CON1), and then throttled into a lower pressure stream 10, it gains heat by the cooling process in the evaporator (EVP1), exits as stream 17 and mixes uniformly with stream 16 to yield stream 18. This transfers its heat energy in the heat exchanger (HEX1), exiting as stream 19, it further loses its heat in the low-pressure condenser (CON 2) and exits as stream 20, whose pressure is increased in the pump (PUM 1) to become stream 21 that gains heat in HEX1 into stream 22, and further gains more heat in the heat exchanger (HEX2) to exit as stream 23 which returns to the VG for KC and the cycle continues in that order. The working fluid (R245fa) of the organic Rankine cycle (ORC) subsystem receives its heat energy from VG for ORC and then flows as stream 41 to expand in the turbine, converting some of its heat energy to shaft work that drives an alternator to produce electric power. Exiting as stream 42 and entering into the high-pressure condenser (CON4), it loses some of its energy and exits as stream 43, whose pressure is further lowered when throttled to stream 44. It gains heat energy through the cooling process in the evaporator (EVP3) and leaves as stream 38, which is compressed by the compressor (CMP) into stream 77. Stream 77 flows into the low-pressure condenser (CON3), releasing some of its heat energy and flows out as a low-temperature, low-pressure stream 39, which is pumped by a pump (PUM3) back to VG for ORC as stream 40. This cycle continues in this manner. The ammonia-water fluid fraction (stream 76) from the Kalina cycle separators (S1 & S2) transfers heat to the Lithium Bromide water (LiBr-H₂O) solution inside the desorber and exits as stream 15. When the weak LiBr-H₂O solution (stream 28) is heated, a portion of the water absorbed by the LiBr is released as vapour (stream 34). Due to the removal of water from the remaining solution, it becomes more concentrated (stream 29). The water vapour coming from the generator/desorber is then condensed (CON 5) to a saturated liquid state in a condenser (stream 34 to stream 71), where the heat rejected is utilized for space heating/hot water production. The condensed liquid is then throttled to a lower pressure. This pressure is determined by setting the evaporator temperature to be the saturation temperature and determining the corresponding saturation pressure. After throttling, the refrigerant is in the saturated liquid-vapour state

(stream 72). Afterwards, the refrigerant is passed through the evaporator, where the refrigerant absorbs heat from the space being cooled/air-conditioned. After the evaporator, the refrigerant is sent to the absorber (stream 35). The strong solution exiting from the desorber is passed through a heat exchanger (HEX3) and recovers some heat (stream 29 to stream 30) that was provided to the solution in the desorber. And then, the heat is transferred to the incoming weak solution (stream 27 to stream 28). After the heat exchanger, the strong solution is then throttled to a lower pressure in the absorber (stream 31). The strong solution (stream 31) coming into the absorber is cooled by rejecting some heat into the atmosphere. The lower temperature of the solution increases its capability to absorb water vapour, and therefore, the vapour incoming from the evaporator is absorbed to make a weak LiBr-H₂O solution (stream 26). This weak solution is then pumped.

(PUM 2) to a higher pressure to the desorber (stream 27) and is passed through the heat exchanger (HEX3) to gain some heat (stream 28) and return to the desorber. The cycle continues in this order. The heat released in the form of flue gases (stream 57) from the combustion of fuel obtained from biomass gasification flows to the hot water generating unit and transfers its heat energy to the hot water generator for domestic hot water supply. Exiting (stream 60) into a heat exchanger (HEX) and transferring energy to the incoming water (stream 61). The spent flue gases (stream 70) leave HEX through the stack, while heated water (stream 62) and hot water by-product (stream 64) from the proton exchange membrane (PEM) electrolyzer mix uniformly as stream 63. A portion of the electric power generated from the Kalina cycle (stream 79) is used by electrolysis reaction to decompose the heated water (stream 63) in the PEM electrolyzer, which yields hydrogen (stream 65) going into the H₂ storage tank, oxygen and hot water mixture (stream 67) which goes through the O₂ separator to split into oxygen (stream 78) going into the O₂ storage tank and hot water (stream 64) going as a portion of the input to feed the PEM electrolyzer. The chemical energy of the fuel, hydrogen (stream 68) and oxygen from the air, is converted into electricity (stream 81) in the fuel cell unit through a pair of redox reactions.

Thermodynamic Modelling and Analysis

This research work deployed thermodynamic modelling, which specifically dwelt on mass, energy, and exergy balances of each component of the subsystems of the multigeneration system through developed codes in the EES software [35].

Energy Modelling

The system is modelled using the energy flow equation at steady state conditions for a control volume. Thus, for the k^{th} component, the energy balance, since the potential and kinetic energy effects are negligible, is generally obtained as follows from Rajput [36]:

$$\dot{Q}_k + \sum_k |H_i + KE_i + PE_i| = \dot{W}_k + \sum_k |H_j + KE_j + PE_j| \quad (1)$$

The terms in Eq. (1) are represented as: $\dot{Q}_k$ = heat rate to the k th component, $\dot{W}_k$ = rate of work output from the k th component, while H_i , KE_i , and PE_i , represent the enthalpy, kinetic energy, and potential energy, respectively, at the component inlet. Similar expressions are used for the control volume at the outlet.

However, since the potential and kinetic energy effects are negligible, Equation (1) is reduced to Equation (2) as:

$$\dot{Q}_k + \sum_k |H_i| = \dot{W}_k + \sum_k |H_j| \quad (2)$$

Equation (2) is applied to all components of the system; similarly, the mass balance is obtained at the component level with the term [36]:

$$\sum \dot{m}_i = \sum \dot{m}_j \quad (3)$$

Table 1. Summary of energy balances for the system

Component	Energy balance
PTC	$\dot{m}_{54} h_{54} + E_{sun} = \dot{m}_{52} h_{52}$
ORC vapour generator	$\dot{m}_{40} h_{40} + \dot{m}_{53} h_{53} = \dot{m}_{41} h_{41} + \dot{m}_{54} h_{54}$
ORC turbine	$\dot{m}_{41} h_{41} = \dot{m}_{42} h_{42} + W_{ORC_{TURB}} + W_{ORC_{PUMP}} + W_{COMP}$
ORC condenser 4	$\dot{m}_{50} h_{50} + \dot{m}_{42} h_{42} = \dot{m}_{51} h_{51} + \dot{m}_{43} h_{43}$
ORC pump	$W_{ORC_{pump}} + \dot{m}_{39} h_{39} = \dot{m}_{40} h_{40}$
ORC valve 6	$\dot{m}_{43} h_{43} = \dot{m}_{44} h_{44}$
ORC evaporator 3	$\dot{m}_{44} h_{44} + \dot{m}_{45} h_{45} = \dot{m}_{38} h_{38} + \dot{m}_{46} h_{46}$
ORC compressor	$\dot{m}_{38} h_{38} + W_{comp} = \dot{m}_{77} h_{77}$
Kalina vapour generator	$\dot{m}_{52} h_{52} + \dot{m}_{23} h_{23} = \dot{m}_1 h_1 + \dot{m}_{53} h_{53}$
Kalina separator 1	$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3$
Kalina separator 2	$\dot{m}_5 h_5 = \dot{m}_6 h_6 + \dot{m}_8 h_8$
Kalina turbine	$\dot{m}_3 h_3 = \dot{m}_5 h_5 + W_{Kal_{TURB}} + W_{PEM}$
Kalina valve 3	$\dot{m}_2 h_2 = \dot{m}_4 h_4$
Kalina valve 2	$\dot{m}_6 h_6 = \dot{m}_7 h_7$
Kalina valve 1	$\dot{m}_9 h_9 = \dot{m}_{10} h_{10}$
Kalina condenser 1	$\dot{m}_8 h_8 + \dot{m}_{11} h_{11} = \dot{m}_9 h_9 + \dot{m}_{12} h_{12}$
Kalina evaporator 1	$\dot{m}_{10} h_{10} + \dot{m}_{13} h_{13} = \dot{m}_{14} h_{14} + \dot{m}_{17} h_{17}$
Kalina condenser 2	$\dot{m}_{19} h_{19} + \dot{m}_{24} h_{24} = \dot{m}_{20} h_{20} + \dot{m}_{25} h_{25}$
Kalina HEX 1	$\dot{m}_{18} h_{18} + \dot{m}_{21} h_{21} = \dot{m}_{19} h_{19} + \dot{m}_{22} h_{22}$
Kalina HEX 2	$\dot{m}_{15} h_{15} + \dot{m}_{22} h_{22} = \dot{m}_{16} h_{16} + \dot{m}_{23} h_{23}$
Kalina pump 1	$W_{Kal_{pump}} + \dot{m}_{20} h_{20} = \dot{m}_{21} h_{21}$
VAS desorber	$\dot{m}_{76} h_{76} + \dot{m}_{28} h_{28} = \dot{m}_{15} h_{15} + \dot{m}_{29} h_{29} + \dot{m}_{34} h_{34}$
VAS condenser 5	$\dot{m}_{34} h_{34} + \dot{m}_{73} h_{73} = \dot{m}_{71} h_{71} + \dot{m}_{74} h_{74}$
VAS valve 5	$\dot{m}_{71} h_{71} = \dot{m}_{72} h_{72}$
VAS valve 4	$\dot{m}_{30} h_{30} = \dot{m}_{31} h_{31}$
VAS evaporator 2	$\dot{m}_{36} h_{36} + \dot{m}_{72} h_{72} = \dot{m}_{37} h_{37} + \dot{m}_{35} h_{35}$
VAS HEX 3	$\dot{m}_{27} h_{27} + \dot{m}_{29} h_{29} = \dot{m}_{28} h_{28} + \dot{m}_{30} h_{30}$
VAS pump 2	$W_{VAS_{pump}} + \dot{m}_{26} h_{26} = \dot{m}_{27} h_{27}$
VAS absorber	$\dot{m}_{31} h_{31} + \dot{m}_{35} h_{35} + \dot{m}_{32} h_{32} = \dot{m}_{33} h_{33} + \dot{m}_{26} h_{26}$
Biomass combustor	$\dot{m}_{55} h_{55} + \dot{m}_{56} h_{56} = \dot{m}_{57} h_{57}$
Water heater	$\dot{m}_{57} h_{57} + \dot{m}_{58} h_{58} = \dot{m}_{59} h_{59} + \dot{m}_{60} h_{60}$
PEM HEX	$\dot{m}_{60} h_{60} + \dot{m}_{61} h_{61} = \dot{m}_{62} h_{62} + \dot{m}_{70} h_{70}$
PEM Electrolyser	$\dot{m}_{63} h_{63} + W_{PEM} = \dot{m}_{67} h_{67}$
Oxygen separator	$\dot{m}_{67} h_{67} = \dot{m}_{64} h_{64} + \dot{m}_{78} h_{78}$

Parabolic Trough Collector Subsystem

The Parabolic Trough collectors (PTCs) receive energy from the sun as one of the energy sources of the multigeneration energy system. PTCs are eligible to deliver high temperatures up to 400 °C for heat production or electricity generation and are available in the market with several types. The energy balance in the PTC, for Equation (1), is obtained as:

$$\dot{m}_{54}h_{54} + E_{sun} = \dot{m}_{52}h_{52} \quad (4)$$

The Actual useful heat gain from a concentrating type collector is defined with the Hottel–Whillier equation from Kalogirou [37] and Tyagi et al. [38]:

$$E_{sun} = F_r A_a [C(\rho\alpha)S_t - U_L(T_{col} - T) - \varepsilon\sigma(T_{col}^4 - T_o^4)] \quad (5)$$

where, F_r is the collector heat removal factor, A_a is the aperture area, C is the concentration ratio, ρ , α , ε , and σ are reflectivity, absorptivity, and emissivity and Stephan Boltzman constant, U_L is the overall heat loss coefficient and can be directly obtained or determined using equations in Tyagi et al. [38], S_t is global solar radiation, and T is the collector temperature. The heat removal factor of the collectors can be defined as follows:

$$F_r = \frac{\dot{m}c_p}{A_r U_L} \left[1 - e^{-\frac{A_r U_L F}{\dot{m}c_p}} \right] \quad (6)$$

Here, the unknowns F , $\dot{m}$, c_p , and A_r are collector efficiency, working fluid mass flow rate, specific heat and reflector area, respectively. The connection between the aperture and reflector area can be written in terms of concentration ratio as follows [38]

$$C = \frac{A_r}{A_a} \quad (7)$$

The definition for aperture area is [38]:

$$A_a = (\omega - D_c)L \quad (8)$$

where L is the collector length, ω is the collector width and D_c is the collector cover diameter.

From Equation (1), the related energy equations for each of the components and subsystems are formulated for the multigeneration system in Table 1. The properties of the different hardwood species [39] used as biomass are presented in Table 2

Exergy Modelling

The exergy modelling of the integrated energy system is performed at steady state conditions using the second law of thermodynamics. Exergy-based modelling has the potential to identify components in the system with large irreversibilities and quantify them. The analysis considers each component as a control volume while applying the general exergy balance expression in Equation. (39) from Bejan and Tsatsaronis [40] as:

Table 2. Elemental composition and combustion properties of some the hardwood species [30]

Wood Species	Common Name (English)	C (%)	H (%)	O (%)	N (%)	S (%)	GCV (MJ/kg)	Density (g/cm ³)	MC (%)	Ash (%)	FVI
Acacia sp	Acacia	49.05	5.46	45.13	0.32	0.03	18.944	1.125	9.68	5.38	4092.19
Azela africana	African mahogany / Apa	48.26	5.27	46.15	0.29	0.03	18.356	0.792	10.00	6.28	2315.00
Celtis sp	Hackberry	45.06	5.04	49.62	0.25	0.03	16.652	0.658	9.84	4.73	2354.11
Brachystegia eurycoma	African walnut (locally)	47.72	5.74	46.35	0.18	0.02	18.673	0.735	9.52	5.35	2694.64
Bombax bounopozense	Red silk cotton tree	45.38	5.53	48.94	0.14	0.01	17.372	1.007	10.17	5.78	2976.02
Blighia sapida	Akee apple	46.66	5.16	47.88	0.28	0.03	17.511	0.582	9.84	7.24	1430.51
Piptadeniastrum africanum	Dahoma	42.67	5.60	51.48	0.24	0.02	16.272	0.843	10.17	4.87	2769.59
Cleistopholis partens	Salt tree	49.11	5.33	45.34	0.20	0.02	18.792	0.751	10.53	5.85	2290.98
Triplochiton scleroxylon	Obeche / African whitewood	46.28	5.46	48.07	0.17	0.02	17.692	1.032	9.09	6.78	2962.52
Albizia sp	West African albizia	41.07	5.53	53.16	0.22	0.02	15.475	1.056	8.62	7.33	2586.38

Table 3. Summary of component exergy balances for the system

Component	Component exergy balance	Exergy of fuel	Exergy of product
PTC	$\dot{E}_{54} + \dot{E}_{Q_{sum}} = \dot{E}_{52} + \dot{E}D$	$\dot{E}_{Q_{sum}}$	$\dot{E}_{52} - \dot{E}_{54}$
ORC vapour gen.	$\dot{E}_{40} + \dot{E}_{53} = \dot{E}_{41} + \dot{E}_{54} + \dot{E}D$	$\dot{E}_{54} - \dot{E}_{53}$	$\dot{E}_{40} - \dot{E}_{41}$
ORC turbine	$\dot{E}_{41} = \dot{E}_{42} + \dot{E}_{W_{ORC_{TURB}}} + \dot{E}_{W_{ORC_{PUMP}}} + \dot{E}_{W_{COMP}} + \dot{E}D$	$\dot{E}_{41} - \dot{E}_{42}$	$\dot{E}_{W_{ORC_{TURB}}} + \dot{E}_{W_{ORC_{PUMP}}} + \dot{E}_{W_{COMP}}$
ORC cond. 4	$\dot{E}_{50} + \dot{E}_{42} = \dot{E}_{51} + \dot{E}_{43} + \dot{E}D$	$\dot{E}_{42} - \dot{E}_{43}$	$\dot{E}_{51} - \dot{E}_{50}$
ORC pump	$\dot{E}_{39} + \dot{E}_{W_{ORC_{pump}}} = \dot{E}_{40} + \dot{E}D$	$\dot{E}_{W_{ORC_{pump}}}$	$\dot{E}_{40} - \dot{E}_{39}$
ORC valve 6	$\dot{E}_{43} = \dot{E}_{44} + \dot{E}D$	$\dot{E}_{43}$	$\dot{E}_{44}$
ORC evap. 3	$\dot{E}_{44} + \dot{E}_{45} = \dot{E}_{38} + \dot{E}_{46} + \dot{E}D$	$\dot{E}_{45} - \dot{E}_{46}$	$\dot{E}_{38} - \dot{E}_{44}$
ORC comp.	$\dot{E}_{38} + \dot{E}_{W_{comp}} = \dot{E}_{77} + \dot{E}D$	$\dot{E}_{W_{comp}}$	$\dot{E}_{77} - \dot{E}_{38}$
Kalina vap. gen.	$\dot{E}_{52} + \dot{E}_{23} = \dot{E}_1 + \dot{E}_{53} + \dot{E}D$	$\dot{E}_{52} - \dot{E}_{53}$	$\dot{E}_1 - \dot{E}_{23}$
Kalina sep. 1	$\dot{E}_1 = \dot{E}_2 + \dot{E}_3 + \dot{E}D$	$\dot{E}_1$	$\dot{E}_2 + \dot{E}_3$
Kalina sep. 2	$\dot{E}_5 = \dot{E}_6 + \dot{E}_8 + \dot{E}D$	$\dot{E}_5$	$\dot{E}_6 + \dot{E}_8 + \dot{E}D$
Kalina turb.	$\dot{E}_3 = \dot{E}_5 + \dot{E}_{W_{Kal_{TURB}}} + \dot{E}_{W_{PEM}} + \dot{E}D$	$\dot{E}_3 - \dot{E}_5$	$\dot{E}_{W_{Kal_{TURB}}} + \dot{E}_{W_{PEM}}$
Kalina valve 3	$\dot{E}_2 = \dot{E}_4 + \dot{E}D$	$\dot{E}_2$	$\dot{E}_4$
Kalina valve 2	$\dot{E}_6 = \dot{E}_7 + \dot{E}D$	$\dot{E}_6$	$\dot{E}_7$
Kalina valve 1	$\dot{E}_9 = \dot{E}_{10} + \dot{E}D$	$\dot{E}_9$	$\dot{E}_{10}$
Kalina cond. 1	$\dot{E}_8 + \dot{E}_{11} = \dot{E}_9 + \dot{E}_{12} + \dot{E}D$	$\dot{E}_8 - \dot{E}_9$	$\dot{E}_{12} - \dot{E}_{11}$
Kalina eva. 1	$\dot{E}_{10} + \dot{E}_{13} = \dot{E}_{14} + \dot{E}_{17} + \dot{E}D$	$\dot{E}_{13} - \dot{E}_{14}$	$\dot{E}_{17} - \dot{E}_{10}$
Kalina condenser 2	$\dot{E}_{19} + \dot{E}_{24} = \dot{E}_{20} + \dot{E}_{25} + \dot{E}D$	$\dot{E}_{19} - \dot{E}_{20}$	$\dot{E}_{25} - \dot{E}_{24}$
Kalina HEX 1	S	$\dot{E}_{18} - \dot{E}_{19}$	$\dot{E}_{22} - \dot{E}_{21}$
Kalina HEX 2	$\dot{E}_{15} + \dot{E}_{22} = \dot{E}_{16} + \dot{E}_{23} + \dot{E}D$	$\dot{E}_{15} - \dot{E}_{16}$	$\dot{E}_{23} - \dot{E}_{22}$
Kalina pump 1	$\dot{E}_{W_{Kal_{pump}}} + \dot{E}_{20} = \dot{E}_{21} + \dot{E}D$	$\dot{E}_{W_{Kal_{pump}}}$	$\dot{E}_{21} - \dot{E}_{20}$
VAS desorber	$\dot{E}_{76} + \dot{E}_{28} = \dot{E}_{15} + \dot{E}_{29} + \dot{E}_{34} + \dot{E}D$	$\dot{E}_{76} + \dot{E}_{28}$	$\dot{E}_{15} + \dot{E}_{29} + \dot{E}_{34}$
VAS cond. 5	$\dot{E}_{34} + \dot{E}_{73} = \dot{E}_{71} + \dot{E}_{74} + \dot{E}D$	$\dot{E}_{34} - \dot{E}_{71}$	$\dot{E}_{74} - \dot{E}_{73}$
VAS valve 5	$\dot{E}_{71} = \dot{E}_{72} + \dot{E}D$	$\dot{E}_{71}$	$\dot{E}_{72}$
VAS valve 4	$\dot{E}_{30} = \dot{E}_{31} + \dot{E}D$	$\dot{E}_{30}$	$\dot{E}_{31}$
VAS evap. 2	$\dot{E}_{36} + \dot{E}_{72} = \dot{E}_{37} + \dot{E}_{35} + \dot{E}D$	$\dot{E}_{36} - \dot{E}_{37}$	$\dot{E}_{35} - \dot{E}_{72}$
VAS HEX 3	$\dot{E}_{27} + \dot{E}_{29} = \dot{E}_{28} + \dot{E}_{30} + \dot{E}D$	$\dot{E}_{29} - \dot{E}_{30}$	$\dot{E}_{28} - \dot{E}_{27}$
VAS pump 2	$\dot{E}_{W_{VAS_{pump}}} + \dot{E}_{26} = \dot{E}_{27} + \dot{E}D$	$\dot{E}_{W_{VAS_{pump}}}$	$\dot{E}_{27} - \dot{E}_{26}$
VAS absorber	$\dot{E}_{31} + \dot{E}_{35} + \dot{E}_{32} = \dot{E}_{33} + \dot{E}_{26} + \dot{E}D$	$\dot{E}_{31} + \dot{E}_{35}$	$\dot{E}_{26} + \dot{E}_{33} - \dot{E}_{32}$
Biomass combustor	$\dot{E}_{55} + \dot{E}_{56} = \dot{E}_{57} + \dot{E}D$	$\dot{E}_{55} + \dot{E}_{56}$	$\dot{E}_{57}$
Water heater	$\dot{E}_{57} + \dot{E}_{58} = \dot{E}_{59} + \dot{E}_{60} + \dot{E}D$	$\dot{E}_{57} - \dot{E}_{60}$	$\dot{E}_{59} - \dot{E}_{58}$
PEM HEX	$\dot{E}_{60} + \dot{E}_{61} = \dot{E}_{62} + \dot{E}_{70} + \dot{E}D$	$\dot{E}_{60} - \dot{E}_{62}$	$\dot{E}_{70} - \dot{E}_{61}$
PEM Electrolyser	$\dot{E}_{63} + \dot{E}_{W_{PEM}} = \dot{E}_{67} + \dot{E}_{65} + \dot{E}D$	$\dot{E}_{63}$	$\dot{E}_{67} + \dot{E}_{65}$
Oxygen separator	$\dot{E}_{67} = \dot{E}_{64} + \dot{E}_{78} + \dot{E}D$	$\dot{E}_{67}$	$\dot{E}_{64} + \dot{E}_{78}$

$$\dot{E}_{Q_k} + \sum_k \dot{E}_i = \dot{E}_{W_k} + \sum_k \dot{E}_j + \dot{E}_{D_k} \quad (9)$$

Where $\dot{E}_{D_k}$ is the exergy destruction rate, $\dot{E}_{Q_k}$ is the exergy flow rate associated with heat transfer, $\dot{E}_{W_k}$ is the rate of work done within the control volume, $\sum_k \dot{E}_i$ and $\sum_k \dot{E}_j$ are the sums of the exergy flow rate in and out of the control volume, respectively. The subscript 'k' represents the kth component. The exergy associated with work and heat is defined as [40]:

$$\dot{E}_{W_k} = \dot{m}_i |h_j - h_i| \quad (10)$$

$$\dot{E}_{Q_k} = Q_k \left| 1 - \frac{T_0}{T_k} \right| \quad (11)$$

The exergy destruction can also be expressed in terms of product and fuel exergy as:

$$\dot{E}_{D,k} = \dot{E}_{F,k} - \dot{E}_{P,k} \quad (12)$$

The exergy efficiency, ψ_k , and the exergy destruction ratio, $Y_{D,k}$, are equally defined for the kth component as [40]:

$$\psi_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} \quad (13)$$

$$Y_{D,k} = \frac{E_{D,k}}{\dot{E}_{F,total}} \quad (14)$$

$$\varphi = \varphi_1 + \varphi_2 \quad (18)$$

Parabolic trough collector subsystem

The exergy balance in the PTC, for Equation (9), is obtained as:

$$\dot{E}_{54} + \dot{E}_{\dot{Q}_{sun}} = \dot{E}_{52} + \dot{E}_D \quad (15)$$

The amount of solar heat ratio collected by the solar collectors and transferred to the working fluid can be defined as [37]:

$$\dot{Q}_{sun} = F_r A_a S_t \eta_r C_r \quad (16)$$

Here, C_r represents the number of collector rows and η_r represents receiver efficiency. Equation (16) is used to calculate heat exergy

input to the system since the collector and heat exchanger efficiencies are not included in Equation (5)

From Equation (9) the related exergy balance, the exergy of fuel, and the exergy of product around the state points in each component of the subsystems of the multigeneration system are expressed below in Table 3.

Thermophysical Properties of the Hybrid Nanofluid

The equations used to calculate the hybrid nanofluid characteristics are provided in this section and are adapted from the source, Takabi & Salehi [41].

In the equations below, “1” represents a Multi-walled carbon nanotube (MWCNT) nanoparticle, “2” represents Aluminium Oxide (Al_2O_3) nanoparticle, “bf” represents the base fluid Therminol VP1, and “hnf” represents the hybrid nanofluid.

The density of the hybrid nanofluid is derived below (Equation (17)) as an extended form of single-particle nanofluid, which showed great agreement with experimental results [41]:

$$\rho_{hnf} = \varphi_1 \rho_1 + \varphi_2 \rho_2 + (1 - \varphi) \rho_{bf} \quad (17)$$

φ is the total volume concentration of two types of nanoparticles scattered in a hybrid nanofluid, computed as [41]:

Equation (19) below is an extended form of the single-particle nanofluid used in specifying the heat capacity of the hybrid nanofluid [41]:

$$C_{hnf} = \frac{\varphi_1 \rho_1 C_1 + \varphi_2 \rho_2 C_2 + (1 - \varphi) \rho_{bf} C_{bf}}{\rho_{hnf}} \quad (19)$$

In addition, the thermal conductivity of hybrid nanofluid, which is defined according to the Maxwell Model, is derived below (Equation (20)) as an extended form of single-particle nanofluid, which showed great agreement with experimental results [41]:

$$\frac{k_{hnf}}{k_{bf}} = \left(\frac{(\varphi_1 k_1 + \varphi_2 k_2)}{\varphi} + 2k_{bf} + 2(\varphi_1 k_1 + \varphi_2 k_2) - 2\varphi k_{bf} \right) \times \left(\frac{(\varphi_1 k_1 + \varphi_2 k_2)}{\varphi} + 2k_{bf} - (\varphi_1 k_1 + \varphi_2 k_2) + \varphi k_{bf} \right)^{-1} \quad (20)$$

Plant Performance Indices

Several plant performance indices were considered in the plant, including the Kalina turbine and pump work,

Table 4. Summary of key system performance indices

Performance index	Expression
Kalina turbine work	$m_3 h_3 - h_5 $
Kalina power requirement	$m_{20} h_{21} - h_{20} $
Kalina evaporator of cooling	$m_{17} h_{17} - h_{10} $
Kalina exergy of cooling	$ ex_{10} - ex_{17} $
ORC turbine output	$m_{41} h_{41} - h_{42} $
ORC pump work	$m_{39} h_{40} - h_{39} $
ORC compressor work	$m_{77} h_{77} - h_{38} $
ORC evaporator cooling	$m_{38} h_{38} - h_{44} $
ORC evaporator exergy of cooling	$ ex_{44} - ex_{38} $
PTC heat rate	$m_{52} h_{52} - h_{54} $
PTC exergy of heat	$ ex_{52} - ex_{54} $
VAS cooling rate	$m_{72} h_{72} - h_{35} $
VAS exergy of cooling	$ ex_{72} - ex_{35} $

Table 5. Model validation of present work with (Parikhani et al. [42] for Kalina cycle subsystem

Parameter	Reference [42]	Present work
System working fluid	Ammonia-water	Ammonia-water
Mass flow rate (Kg/s)	2.917	2.917
Ammonia concentration at turbine outlet (%)	99.97	83.06
Turbine inlet Temperature (°C)	160.2	160.2
Turbine inlet pressure (bars)	30	30
Turbine outlet pressure (bars)	2.986	2.986
Net power output (kW)	886.3	941.3

exergy of refrigeration, both for the Kalina and ORC, as well as ORC turbine work. All performance indices and the corresponding developed relationships are shown in Table 4, following the nomenclature in Figure 1.

RESULTS VALIDATION

To validate the thermodynamic simulation results for the proposed Multigeneration system, a suitable code is written in Engineering Equation Solver (EES) software to compare the obtained findings to those in the literature. In this study, four case studies from the literature are chosen, and each is simulated in a constant situation. These case studies include the KC, ORC, VAS subsystem, and PEM electrolyzer system. Validation for each case study is performed below.

Validation of Kalina Cycle Subsystem

In the first case study, a KC is selected from Parikhani et al. [42], and simulated under constant input parameters. In this simulation, some of the KC design input parameters of [42] were plugged into the KC of the present work model. It can be expressed that the results of the current investigation properly corroborate the results of [42], with some improvements in performance indices. According to Table 5, a performance parameter (i.e. net output power) is selected, and the calculated value is compared with the literature. There is a good agreement between the obtained result in the present model and those reported in the literature.

Validation Of Organic Rankine Cycle Subsystem

For the second case study, the results of the model for the Organic Rankine Cycle system are presented in Table 6. Some of the ORC design input parameters of [43] were plugged into the ORC of the present work model. It can be stated that the results of the current investigation properly verify the results of Ozturk & Dincer [43], with some enhancements in the performance indices. From Table 6, two performance parameters (i.e., Turbine Power Output and Pump work) are selected, and the calculated values are compared with the literature. This comparison of the different parameters validates well with the literature.

Validation of Vapour Absorption Subsystem

Presented in Table 7 is the third case study, which is the vapor absorption system (VAS). In the simulation, some of the VAS design input parameters of Karaca and Dincer [44], were substituted into the VAS of the present work model. From Table 7, two performance parameters (i.e. evaporator cooling rate and energetic COP) are selected and the calculated values are compared with the literature. This comparison of the different parameters, agrees well with the literature.

Validation of PEM Electrolyzer Subsystem

The final but not least significant verification step is using a PEM electrolyzer. In the mathematical modelling of the PEM electrolyzer, the operating temperature is set at 90°C, the power supplied is 4.07kW, and the oxygen and hydrogen pressures are fixed at 1 bar. Based on Table 8, a performance indicator (i.e., hydrogen production rate) is

Table 6. Model validation of the present work with Ozturk & Dincer, [43] for the ORC generation subsystem

Parameter	Reference [43]	Present work
System working fluid	Isobutane	Isobutane
Mass flow rate (Kg/s)	34.24	34.24
Turbine inlet Temperature (°C)	146.8	146.8
Turbine inlet pressure (bars)	32.5	32.5
Turbine outlet pressure (bars)	4.10	4.10
Turbine Power Output (kW)	2436	3110
Pump work (kW)	234	182.2

Table 7. Model validation of present work with Karaca & Dincer [44] for the VAS subsystem

Parameter	Reference [44]	Present work
System working fluid	Ammonia-water	Ammonia-water
Mass flow rate (Kg/s)	0.0089	0.0089
Pressure across the evaporator (bars)	4.62	4.62
VAS cooling rate (kW)	5.08	4.546
Energetic COP	0.96	0.8783

Table 8. Model validation of the present work with Khanmohammadi et al. [45] for PEM electrolyzer

Parameter	Reference [45]	Present work
Electrolyzer hot water Mass flow rate (Kg/s)	-	1.99
Electrolyzer temperature (°C)	95	95
Electrolyzer Pressure (bar)	1	1
Electricity feed to electrolyzer (kW)	4.07	4.07
Hydrogen production rate (kg/hr)	0.245	0.255

selected, and the calculated value is compared with the literature. It can be expressed that the results of the current investigation substantiate the results of Khanmohammadi et al. [45] as seen in Table 8, with some improvements in performance indices. A minor discrepancy is attributed to the different design operating conditions, like the unavailable or varying mass flow rate of water. Since the mass flow rate of water affects hydrogen production if it is insufficient, but once sufficient, the hydrogen flow rate increases with an increase in power supplied, as long as efficiency losses at high power are managed.

RESULTS AND DISCUSSION

This paper investigates the thermodynamic performance of a solar power-biomass-assisted multigeneration system for the production of electricity, cooling, heating, hydrogen, and hot water from the energetic and exergetic analysis perspectives. The design data and thermodynamic

computation of this modelled investigation at the component level were simulated utilizing a generated source code in EES. Table 9 shows the design data used for the analysis. In conducting the energy and exergy assessments of the solar-biomass integrated multigeneration system, values of mass flow rate, temperature, pressure, specific enthalpy, specific entropy and exergy are serially determined for each state of the system as tabulated in Table 10. These were considered and evaluated using the design conditions in Table 9.

As obtained in Table 4, the major performance indices of the plant are shown in Table 11, and comprises the power output and cooling from both the ORC and KC subsystems. The system has a total power output of 964 kW, with 276.03 kW from the ORC and 145.28 kW from the Kalina system. Also, cooling from the Kalina system after turbine expansion and separation resulted in 129.6 kW of cooling with an evaporator temperature of -33.5 °C. Since the analysis involved the second law analysis, the exergy of cooling in

Table 9. System initial design data

Parameter	Schematic label	Unit	Value
PTC heat rate	QPTC	kW	1295
ORC TIT	t_{41}	°C	130
ORC TIP	P_{41}	Bar	25
ORC intermediate pressure	P_{44}	Bar	0.671
ORC condenser pressure	P_{42}	Bar	4.79
ORC refrigerant mass flow rate	$\dot{m}_{41}$	kg/s	2
Kalina TIT	t_3	°C	150
Kalina TIP	P_3	Bar	20
Ammonium water concentration	x_{21}	%	28
Kalina evaporator pressure	t_{10}	Bar	-34
Kalina condenser pressure	P_8	Bar	6.993
Ammonium water mass flow rate	$\dot{m}_{21}$	kg/s	1.3
VAS generator pressure	P_{34}	Bar	0.07904
VAS generator temperature	t_{34}	°C	80
VAS absorber pressure	P_{35}	Bar	0.008635
PEM electrolyzer temperature	t_{63}	°C	95
Biomass flow rate	$\dot{m}_{56}$	kg/s	0.75

Table 10. The systems' operating thermodynamic properties

S/No	$\dot{m} \left(\frac{kg}{s} \right)$	$t \text{ (}^{\circ}\text{C)}$	$P \text{ (bar)}$	$h \left(\frac{kJ}{kg} \right)$	$s \left(\frac{kJ}{kg \cdot K} \right)$	$E \text{ (kW)}$
1	1.3	150	20	583.3	2.094	161.5
2	1.215	150	20	498.9	1.87	115
3	0.08468	150	20	1796	5.315	41.55
4	1.215	61.9	0.8	498.9	2.053	48.45
5	0.08468	108.8	6.993	1610	5.315	25.87
6	0.005263	108.8	6.993	322.2	1.411	0.2296
7	0.005263	56.7	0.8	322.2	1.476	0.1273
8	0.07942	108.8	6.993	1697	5.578	25.59
9	0.07942	18.8	6.993	-28.4	0.2503	14.63
10	0.07942	-33.5	0.8	-28.4	0.349	12.3
11	1.311	25	1.013	104.3	0.3651	0
12	1.311	50	1.013	208.8	0.7018	5.446
13	5.379	25	1.013	298.4	5.695	0
14	5.379	1	1.013	274.3	5.611	5.518
15	1.221	51.9	0.8	293	1.431	23.61
16	1.221	41.9	0.8	116.5	0.8817	8.093
17	0.07942	53.9	0.8	1604	6.341	0.1462
18	1.3	42.9	0.8	207.4	1.216	12.9
19	1.3	32.9	0.8	34.48	0.6622	2.714
20	1.3	25	0.8	-73.38	0.3073	-0.03079
21	1.3	25.1	20	-71.24	0.3073	2.743
22	1.3	40.1	20	-6.966	0.5177	4.796
23	1.3	55.1	20	57.31	0.7183	10.67
24	1.342	25	1.013	104.3	0.3651	0
25	1.342	50	1.013	208.8	0.7018	5.572
26	7.237	35	0.008635	85.05	0.2104	2.085
27	7.237	35	0.07905	85.08	0.2104	2.313
28	7.237	64	0.07905	144.1	0.394	33.81
29	7.037	80	0.07905	181.8	0.4761	63.6
30	7.037	45	0.07905	111.3	0.2657	8.349
31	7.037	35	0.008635	91.56	0.2025	2.062
32	12.7	25	1.013	104.3	0.3651	0
33	12.7	35	1.013	146.1	0.5031	8.711
34	0.2	80	0.07905	2649	8.452	27.02
35	0.2	5	0.008635	2510	9.029	-35.25
36	2.289	25	1.013	298.4	5.695	0
37	2.289	33	1.013	306.5	5.722	0.2185
38	2	5.5	0.671	408.4	1.748	-13.52
39	2	25	1.478	232.3	1.113	13.16
40	2	25.8	25	234	1.113	16.67
41	2	130	25	389.7	1.556	63.73
42	2	61.4	4.79	377.9	1.556	40.2
43	2	61.4	4.79	282.1	1.27	19.34
44	2	5.5	0.671	282.1	1.295	4.205
45	11.97	25	1.013	298.4	5.695	0
46	11.97	4	1.013	277.3	5.622	9.338

Table 10. The systems' operating thermodynamic properties (continued)

S/No	$\dot{m} \left(\frac{kg}{s} \right)$	$t \text{ (}^{\circ}\text{C)}$	$P \text{ (bar)}$	$h \left(\frac{kJ}{kg} \right)$	$s \left(\frac{kJ}{kg \cdot K} \right)$	$E \text{ (kW)}$
47	-	-	-	-	-	-
48	3.647	25	1.013	104.3	0.3651	0
49	3.647	50	1.013	208.8	0.7018	15.14
50	1.835	25	1.013	104.3	0.3651	0
51	1.835	50	1.013	208.8	0.7018	7.618
52	3.854	180	1.813	903	-	731.6
53	3.854	101	19.6	664.1	-	306.8
54	3.854	65	19.4	567.1	-	153.4
55	4.433	25	1.013	298.4	5.695	0
56	0.75	25	1.013	-8442	1.362	16273
57	5.183	450	1.013	738.4	6.607	871.6
58	2.034	25	1.013	104.3	0.3651	0
59	2.034	90	1.013	376.4	1.191	52.87
60	5.183	350	1.013	631.6	6.448	563.5
61	1.847	25	1.013	104.3	0.3651	0
62	1.847	95	1.013	397.5	1.249	55.18
63	1.847	95	1.013	397.5	1.249	55.18
64	1.846	95	1.013	397.5	1.249	55.16
65	0.00006612	95	1.013	4937	56.4	0.006792
66	-	-	-	-	-	-
67	1.847	95	1.013	397.5	1.249	0.00346
68	-	-	-	-	-	-
69	-	-	-	-	-	-
70	5.183	250	1.013	527.1	6.266	304.1
71	0.2	55	0.07905	2602	8.313	25.84
72	0.2	54.1	0.008635	2602	9.334	-35.01
73	0.06503	25	1.013	104.3	0.3651	0
74	0.06503	60	1.013	250.6	0.8294	0.5187
75	-	-	-	-	-	-
76	1.221	61.9	0.8	498.1	2.051	48.57
77	2	25.2	1.478	422.8	1.752	13.25

Kalina evaporator was recorded as 12.15 kW. In the VAS, at an evaporator temperature of 5 °C, 18.4 kW of refrigeration was achieved, with a corresponding 0.24 kW of exergy of cooling. The PTC contributes about 311.4 kW and 683.8 kW of heat to power the vapour generators of the ORC and KC systems, respectively. The total net output of the system was simulated as 964.9 kW, with 93.32 kW of exergy, while the system's net heat input from the biomass combustor was 3827 kW. The energy and exergy efficiencies of the plant was recorded as 52.2 % and 16.14 %, respectively.

Parametric Study

The surrounding temperature influences most thermodynamic systems in their operation. For example, changes

in the surrounding temperature might occasionally lead to an increase or decrease in system performance. The impact of variations in ambient temperature on the total energy and exergy inputs into the integrated energy system is shown in Figure 2. The results indicate that the total energy input is 1848 kW. This remained relatively constant when ambient temperature varied from 290 to 310 °K for the reason that energy input is not affected by varying environment temperature conditions. Whereas for exergy input, at the same ambient temperature range from 290 to 310 °K, it decreases linearly from 587.8 to 564.1 kW as exergy loss increases.

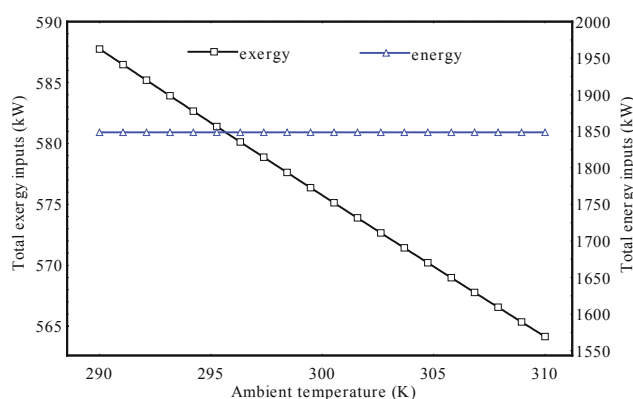
Figure 3 illustrates the impact of ORC turbine inlet temperature on the ORC turbine output. Between 120 to 150 °C, the turbine output power increased steadily from 17.74

Table 11. Summary of performance indices of the energy system

Performance index	Unit	Value
Kalina turbine work	kW	15.68
Kalina pump power requirement	kW	2.775
Kalina evaporator of cooling	kW	129.6
Kalina exergy of cooling	kW	12.15
ORC turbine output	kW	23.53
ORC pump work	kW	3.515
ORC compressor work	kW	28.89
ORC evaporator cooling	kW	252.5
ORC evaporator exergy of cooling	kW	17.73
PTC heat rate	kW	1295
PTC exergy of heat	kW	578.1
VAS cooling rate	kW	18.4
VAS exergy of cooling	kW	0.2358
Total energy output	kW	964.9
Total exergy output	kW	93.32
Total energy input	kW	1848
Total exergy input	kW	578.1
Energy efficiency	%	52.2
Exergy efficiency	%	16.14

to 59.13 kW. The sharp rise is due to the rise in the values of input enthalpies and exergies. Although the turbine power output increases with turbine inlet temperatures, it can be inferred that an optimum TIT of 137.5 °C exists corresponding to a turbine inlet pressure of 25 bar.

Figure 4. shows the impact of Kalina turbine inlet temperature (TIT) on both total energy and exergy efficiencies of the energy system. It is observed that an increase in Kalina turbine inlet temperature from 150 to 170 °C, results in an increase in both overall energy and exergy efficiencies.

**Figure 2.** Ambient temperature effect on system total heat input.

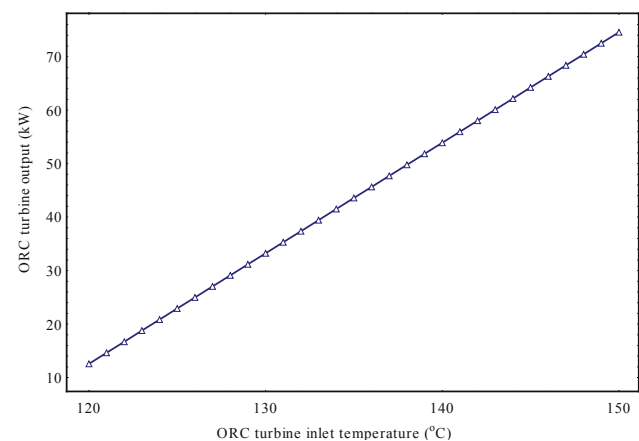
Whereas the energy efficiency increased steadily from 52.2 to 78.4%, the exergy efficiency increased steadily to a peak of 18.42% at 162.9 °C, beyond this temperature, any further increase results in a decrease in power output due to operational disturbances.

Figure 5 demonstrates the impact of ORC turbine outlet pressure on the system performance index. By increasing the backpressure of the turbine from 3 to 6 bar, the exergy and energy efficiencies decrease by about 1% and 4.5%, respectively. However, at a very low turbine back pressure, the fluid can attain a two-phase zone after expansion with a resultant effect on turbine blades.

The influence of ORC turbine back pressure on turbine output is demonstrated in Figure 6. High turbine back pressures result in minimal turbine expansion with truncated turbine work. It was found that when the ORC turbine backpressure was increased from 3 to 6 bar, the ORC turbine power output decreased from 33.83 to 18.92 kW. The results demonstrate the validity of the model linking the operating parameters in the ORC and its components.

The impact on the total exergy and energy efficiencies due to changes in the Kalina turbine outlet pressure is shown in Figure 7. In the design simulation, the Kalina turbine outlet pressure was set at 6.993 bar. However, for this analysis, the Kalina turbine back pressure was varied between 5 and 8 bar; and within this range, the total system energetic and exergetic efficiencies decreased by about 1 % and 2.5 %, respectively. The results are consistent with the behavior of the energy generation system with regards to the value of turbine back pressure against the turbine power output capacity.

The results of the effect of generator temperature on VAS evaporator cooling is shown in Figure 8. The cooling rate with regard to energy and exergy considerations increases at a constant generator pressure of 0.07904 bar. The higher vapour generator temperature increases the heat input to the generator; thus, the refrigerant can absorb heat energy. This results in a higher concentration of refrigerant vapour

**Figure 3.** Effect of ORC TIT on ORC turbine output.

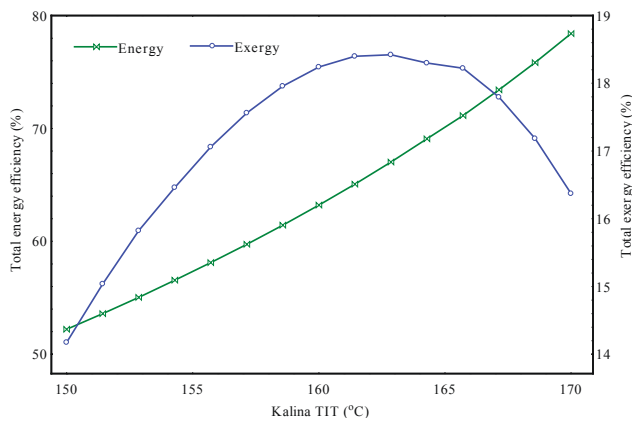


Figure 4. Effect of Kalina TIT on system's total efficiencies.

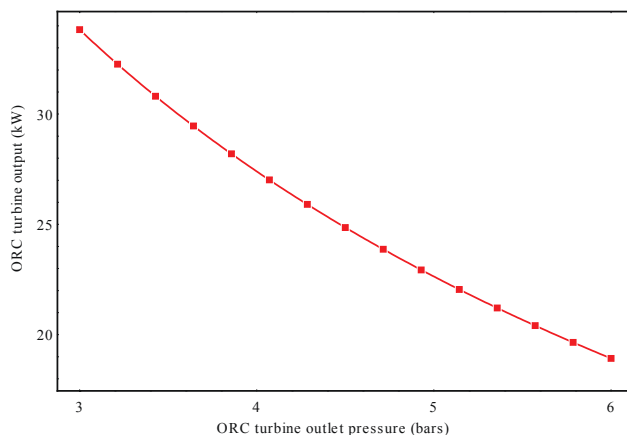


Figure 6. Effect of ORC turbine outlet pressure on ORC turbine power output.

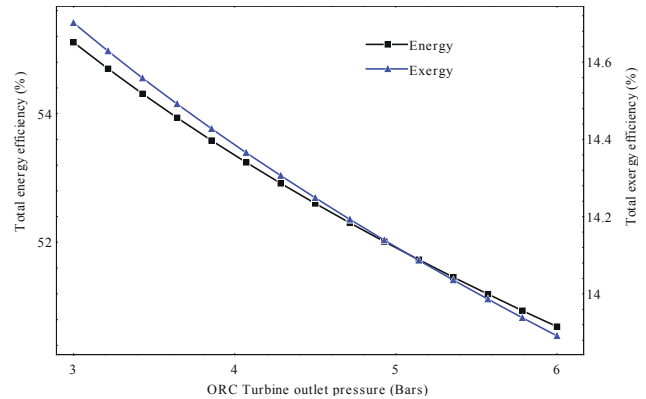


Figure 5. Effect of ORC turbine outlet pressure on total energy and exergy efficiencies.

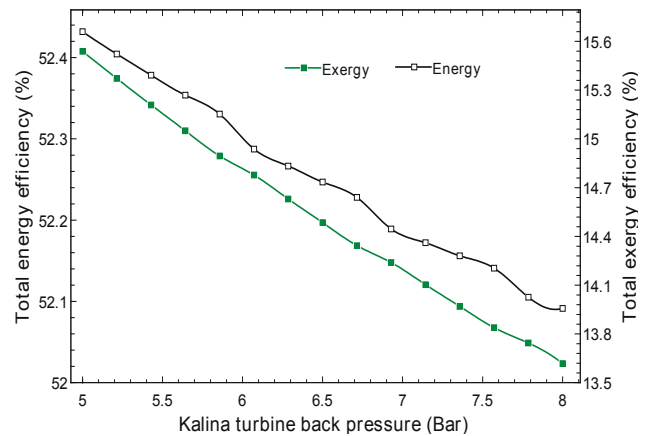


Figure 7. Effect of Kalina turbine back pressure on total energy and exergy efficiencies.

entering the condenser. Since the VAS relies on the absorption and desorption processes to create cooling, the higher concentration of refrigerant vapour in the condenser results in an increased cooling rate in the system. The magnitude of such an increase is depicted in Figure 8, where an additional 40 °C generator temperature increase resulted in an additional 20 kW of cooling in the evaporator with constant operating pressure.

Vapour generator pressure on the cooling rate of the VAS evaporator was investigated with the result shown in Figure 9. Increasing the pressure in the vapour generator raises the boiling point of the refrigerant. This means that the refrigerant will need more energy to change from a liquid to a vapour state. As a result, the refrigerant is not able to absorb enough heat in the evaporator. The reduction of the cooling rate, both in energy and exergy, is thus experienced with higher vapour generator pressures.

The impact of the cell operating temperature of the PEM electrolyzer on the daily hydrogen production rate of the plant is seen in Figure 10. At increasing temperatures, the

PEM becomes more conductive, allowing for easier proton transport. This leads to a reduction in membrane resistance and improves the overall cell efficiency, thereby enhancing the hydrogen production rate. However, the results also show a decrease in the cell voltage when temperature was increased. The cell voltage dropped marginally to 1.15 volts for a cell temperature increase of 40 K. This drop can be attributed to the membrane connectivity due to increased proton migration triggering a voltage drop in the process.

Figure 11 demonstrates the energy balance around the domestic hot water heater, with hot water temperature fixed at 90 °C, gave an indication of the required biomass combustor outlet temperature and the quantity of hot water that can be produced. The quantity of hot water at 90 °C can be high with higher combustor outlet temperature as shown in Figure 11. The system can deliver up to 6000 liters of hot water at 90 °C for a combustor temperature of 425 °C.

The result of hybrid nano fluid source temperature on the overall energy and exergy output of the system was investigated and shown in Figure 12. Without an additional

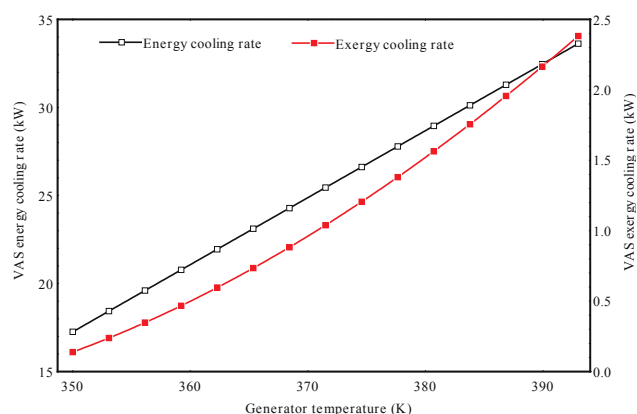


Figure 8. Effect of VAS generator temperature on evaporator cooling rate.

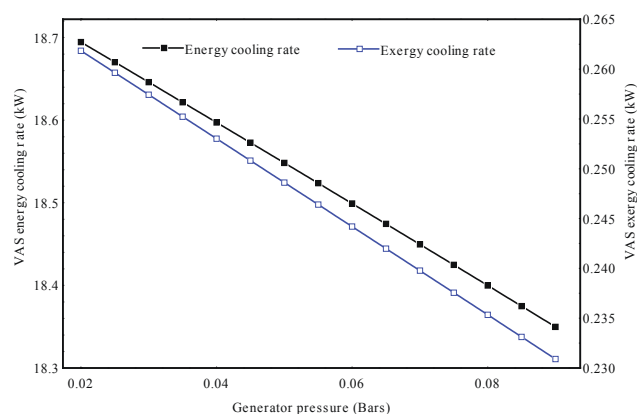


Figure 9. Effect of VAS generator pressure on evaporator cooling rate.

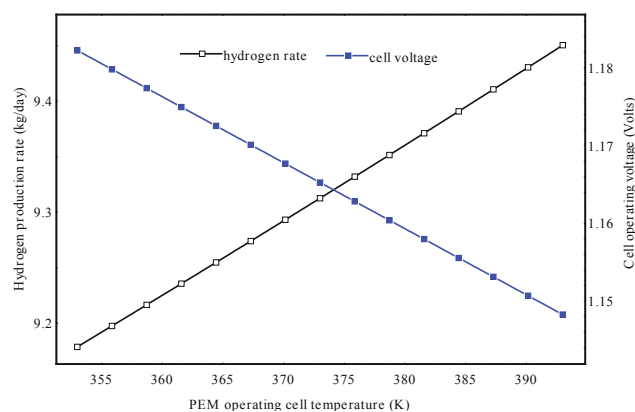


Figure 10. Effect of PEM temperature on cell voltage and hydrogen rate.

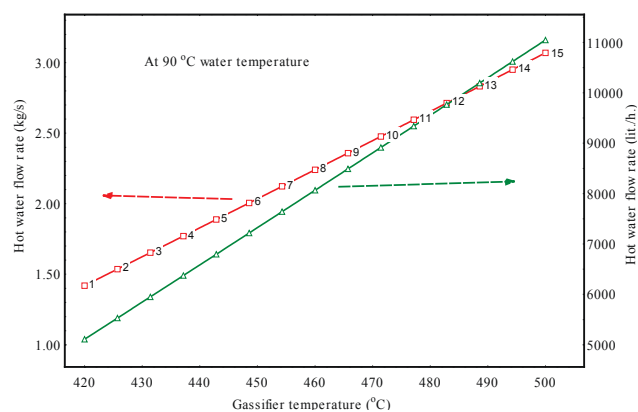


Figure 11. Effect of combustor temperature on domestic hot water production.

re-configuration of the system to reflect changes in turbine inlet parameters for the ORC and KC, an increase in the heat transfer of the hybrid nanofluid will result in additional heat transfer in the vapour generators of the two power

cycles. This is shown in Figure 12. Therefore, an increase in the nano operating temperature will overall result in lower energy and exergy efficiencies with large exergy destruction from the system.

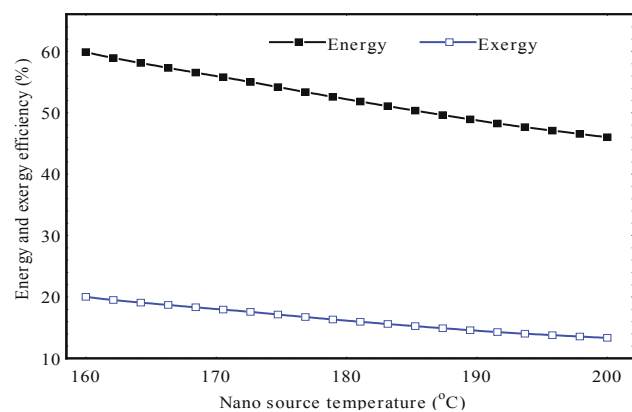


Figure 12. Nano source temperature effect on system total efficiencies.

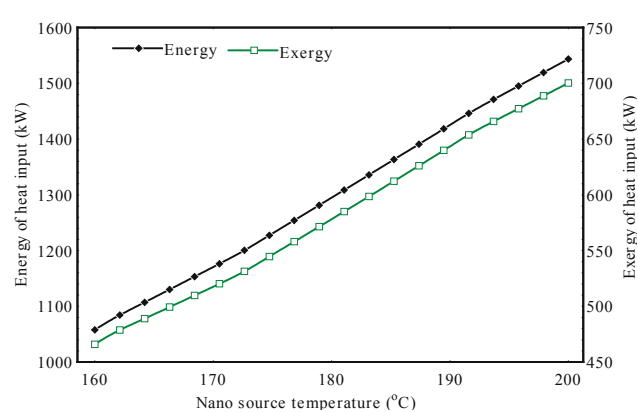


Figure 13. Nano source temperature effect on system total heat inputs.

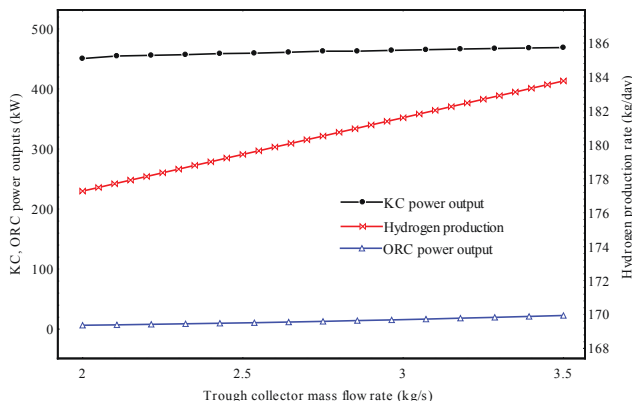


Figure 14. Variation of the collector mass flow rate with KC, ORC power outputs and hydrogen production rate.

The parametric effect of the hybrid Nano fluid source temperature was investigated on the overall energy and exergy inputs of the system in Figure 13. An increase in the Nano source temperature results in an increase in the overall energy input of the integrated energy system. Between the nano source temperatures of 155 °C to 180 °C, total energy input increased significantly from 1549 to 1848 kW. For the same source temperature range, the total exergy also increased from 435.8 to 578.1 kW. This increasing energy and exergy inputs with respect to higher source temperatures is so because both of these types of heat energy are functions of temperature since it is theoretically agreed upon that an increase in thermal energy yields an increase in temperature.

The effect of varying the trough collector mass flow rate on KC net power output, ORC net power output, and hydrogen production rate is demonstrated in Figure 14. As Parabolic Trough Collector (PTC) mass flow rate increases from 2 kg/s to 3.5 kg/s, the Kalina cycle net power output rises linearly by 4.1% from 450.8 kW to 469 kW, while the Organic Rankine Cycle (ORC) net power output increases non-linearly by 251% from 6.475 kW to 22.69 kW, accelerating beyond 2.75 kg/s. Concurrently, daily hydrogen production from the Proton Exchange Membrane (PEM) electrolyser grows steadily by 4.1% from 176.2 kg/day to 183.4 kg/day. These increases are attributed to enhanced heat transfer and improved thermal energy input, highlighting the benefits of optimizing PTC mass flow rate for improved performance in the multigeneration energy system.

CONCLUSION

This research paper examines the thermodynamic performance of a novel solar-biomass power based plant for hydrogen generation, from an energetic and exergetic viewpoints. This integrated energy system is designed for different purposes such as hot water, heating, electricity, cooling,

and hydrogen productions. The parametric studies have been conducted with regard to a number of factors, including ambient temperature, the turbines inlet temperatures and the turbines output pressures. Here is a quick summary of the primary findings from the thermodynamic analysis:

- The ORC and Kalina subsystems are sustainable when run with R245fa and ammonium water as the respective working fluid.
- The system has a total power output of 964 kW, with 276.03 kW from the ORC and 145.28 kW from the Kalina system. Also, cooling from the Kalina system after turbine expansion and separation resulted in 129.6 kW of cooling with an evaporator temperature of -33.5 °C.
- The PTC contributes about 311.4 kW and 683.8 kW of heat to power the vapour generators of the ORC and Kalina systems, respectively. The total net output of the system was simulated as 964.9 kW, with 93.32 kW of exergy, while the system's net heat input from the bio-mass combustor was 3827 kW. The energy and exergy efficiencies of the plant was recorded as 52.2 % and 16.14 %, respectively.

Overall, the developed integrated energy systems model may easily replace conventional energy systems due to its higher energy and exergy efficiency and greater environmental performance. It could be an effective technique for incorporating solar energy into biomass gasification and reforming, allowing for the prediction and comparison of energetic and exergetic performance.

Further research will examine the dynamic behavior of the developed multigeneration plant, as the current work was carried out using a steady-state model, and it was observed that solar irradiance and biomass variations considerably affect efficiency. In order to determine appropriateness from a wider sustainability standpoint, the cost-effectiveness and eco-friendliness will also be assessed utilizing exergoeconomic and exergoenvironmental assessments.

NOMENCLATURE

Symbol	
A_a	Aperture area (m ²)
A_r	Reflector area (m ²)
C	Concentration ratio
C_{hmf}	Heat capacity of the hybrid nanofluid (kJ/K)
c_p	Specific heat (J/kg·K)
C_r	Number of collector rows
D_c	Collector cover diameter (m)
$\dot{E}_{Wk}$	Exergy associated with work (kJ)
$\dot{E}_{Qk}$	Exergy associated with heat (kJ)
$\dot{E}_{Dk}$	Exergy destruction rate (kJ/s)
F	Collector efficiency (%)
F_r	Collector heat removal factor
H	Enthalpy (kJ)
h	Specific enthalpy (kJ/kg)

k	Thermal conductivity (kW/m.K)
L	Collector length (m)
$\dot{m}$	Working fluid mass flow rate (kg/s)
P	Pressure
$\dot{Q}_k$	Heat rate to the kth component (kJ/s)
s	Specific entropy (kJ/kg.K)
S_t	Global solar radiation (kW/m ²)
T	Collector temperature (K)
t	Temperature (°C)
U_L	Overall heat loss coefficient (kW/m ² K)
$\dot{W}_k$	Rate of work output from the kth component (kJ/s)
$Y_{D,k}$	Exergy destruction ratio

Greek symbols

α	- absorptivity
ε	- emissivity
ρ	- reflectivity
ρ_{bf}	- Density of the base fluid (kg/m ³)
ρ_{hnf}	- Density of the hybrid nanofluid (kg/m ³)
ϕ	- Total volume concentratio
σ	- Stephan Boltzman constant (kW/m ² K ⁴)
ω	- Collector width (m)
ψ_k	- exergy efficiency (%)
η_r	Receiver efficiency ((%))

Subscripts and Superscripts

bf	Base fluid
hnf	Hybrid nanofluid
i	Inlet
k	Element k , component k
j	Outlet

Abbreviations and Acronyms

Al ₂ O ₃	Aluminium dioxide
C	Carbon
CHP	Combined heat and power
CMP	Compressor
CO ₂	Carbon dioxide
CON	Condenser
COP	Coefficient of performance
CSP	Concentrated Solar Power
EES	Engineering Equation Solver
EVP	Evaporator
FVI	Fuel Index Values (FVI)
GCV	Gross Calorific Values
H	Hydrogen
HEX	Heat exchanger
KC	Kalina cycle
LCOE	Levelized cost of energy
LiBr	Lithium Bromide water
MC	Moisture Content
MWCNT	Multi-walled carbon nanotube
N	Nitrogen
NASA	National Aeronautics and Space Administration
NH ₃	Ammonia

O	Oxygen
ORC	Organic Rankine cycle
PEM	Proton exchange membrane
PTC	arabolic trough collector
PUM	Pump
PV	Photovoltaics
S	Separator & Sulphur
SAM	System Advisor Model software
SGT	Solar-driven gas turbine
V	Valve
VAS	Vapour absorption system
VG	Vapour generator

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.

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

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Research Article

Design and parametric evaluation of proton exchange membrane fuel cell for enhanced performance

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ABSTRACT

The objective of the study is to explore the influence of various operating conditions on the performance of the Proton exchange membrane fuel cell (PEMFC) using COMSOL software. The effects of gas diffusion layer porosity, membrane conductivity, electrolyte thickness and operating temperature are studied. A structured hexahedral mesh was employed for the simulations. Essential geometric and operational factors are calculated analytically. The computational analysis of the PEMFC is conducted using a coupled approach that integrates free and subsurface flow, species transport and current distribution modules. The governing equations used in the simulation included conservation of energy, momentum, mass and species. The Joule heating, Butler-Volmer and the Nernst equation were incorporated to enable the modeling of reaction kinetics and cell performance. The uniqueness of this study lies in its comprehensive examination of the interactions between gas diffusion layer, gas diffusion electrode and membranes under various conditions, an area that has been less thoroughly investigated in the existing literature. Results reveal that higher gas diffusion layer porosity, increased electrolyte conductivity and reduced membrane thickness enhance fuel cell performance. A 26.54% increase in power density is observed when electrolyte conductivity is raised from 5.05 to 10.6, and a 21.19% improvement is achieved by decreasing electrolyte thickness from 254 μm to 127 μm . This data is vital for applications requiring high power output, such as in transportation or portable devices. Higher conductivity permits for better ion transport within the electrolyte, which can lead to more efficient reactions at the electrodes. This characteristic makes such fuel cells suitable for applications that demand rapid response times and high energy outputs. This study presents a novel approach to optimizing the performance of PEMFCs by systematically investigating the effects of critical operating conditions.

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INTRODUCTION

Proton Exchange Membrane Fuel Cells (PEMFCs) have fascinated substantial interest as an innovative clean energy solution, compact structure, valued for their high efficiency and competence to reduce greenhouse gas emissions. The hydrogen and oxygen is supplied to PEMFCs and it produces electricity through electrochemical reactions along with water and heat as byproducts. These qualities make PEMFCs great options for use in transportation, portable power and energy systems that are spread out. They support the worldwide movement toward sustainable and renewable energy sources. To achieve the best performance and durability, careful design choices are important. It is also necessary to understand the complex interactions among the cell components. The performance of a PEMFC is affected by several important factors. These include the design and materials of the membrane, gas diffusion electrode (GDE) and gas diffusion layer (GDL) as well as operational factors like porosity, temperature and reactant flow rates. The catalyst layer is very important for PEMFC. It is made of platinum based materials and helps to speed up the reactions that involve hydrogen and oxygen which are essential for the fuel cell to work. Platinum is expensive. Improving how well the catalyst layer works is a big challenge in designing PEM fuel cells. Recent advancements in computational modeling have greatly enhanced the study of factors affecting fuel cells in controlled virtual environments. These improvements allow for detailed analysis of various parameters. COMSOL Multiphysics software enables simulations that can capture the intricacies of species transport, current distribution and the effect of various structural and operational conditions on fuel cell output. These models offer invaluable insights into optimizing PEMFC performance by evaluating factors like electrolyte conductivity, membrane thickness, and GDL porosity. PEMFCs have the potential to serve as an economical power source across various sectors, including stationary, mobile and automotive applications [1]. The materials used for the main components of PEMFC are discussed and highlighted that the low power density is one of the problem with the PEMFC [2]. A two dimensional analysis of PEMFC was carried out using FORTRAN and the effects of porosity and pressure on the electrolyte potential were studied. The performance of the fuel cell increased with porosity [3]. A two dimensional analytical model for high temperature PEMFC in the range of 160°C was developed and the results obtained from the model were validated [4]. The influence inlet gas flow rate of the cathode, humidification temperature at cathode inlet, cell temperature on the performance of PEM fuel cells with interdigitated flow field and conventional flow field were studied from the experiments. The results reveal that the performance of the fuel cell increased with the increase of these chosen parameters [5]. The effects of channel depths and anode flow rates on the performance of small three pass serpentine flow field PEMFC were studied

with 200, 400, 600µm channel depth and noticed that the cell with 400µm yielded better performance [6]. The effect of channel path length in flow fields on the performance of PEMFCs was studied experimentally [7]. Commercial flow solver STAR-CD 3.26 was employed and studied the influence of serpentine flow-fields with various rib/ channel's cross section areas on the effectiveness of PEMFC. The numerical analysis discovered that narrower channel with wider rib spacing gives higher performance for stationary condition and the trend is different for automotive condition [8]. The experimental study on PEMFC revealed that the performance of the fuel cell was improved with the decrease of the thickness of the GDL and PEM thickness [9]. The effect of working pressure and temperature on the effectiveness of a PEM fuel cell is investigated using a 3D computational model [10]. They noticed that the performance was improved with increased cell temperature and operating pressure. An experimental analysis was carried out to study the self-regulating capability of open-cathode PEM fuel cells at different fan operating conditions and noticed that the fan speed remarkably affects the thermal and water management of the PEMFC [11]. The PEM fuel cell system thermal stability of UAVs was analyzed using MATLAB/Simulink at different altitudes and noticed the variations of the cooling load with altitude [12]. The hydrogen chilling efficiency for aviation PEM fuel cells was studied with simulation and experimental approach. The results of this study revealed that that effective hydrogen cooling is critical for maintaining thermal stability in aviation applications [13]. A system-level dependability model of the hybrid marine power plant with four PEMFC systems and a battery has been developed for 500 kW capacities [14]. The techno-economic aspects of flow channel design were investigated using Nafion membranes [15]. Compared the S-shaped and two-stage micro channel designs with standard designs and optimum one is identified [16]. The study highlighted that, how ejector configurations can contribute to enhanced performance in large-scale applications [17]. A human memory optimizer algorithm was presented to improve the modeling and characterization of PEM fuel cells and sensitivity and uncertainty analysis also carried out [18]. The artificial rabbits optimization procedure was used for parameter extraction in PEM fuel cells and their study emphasized the substantial role of parameter optimization in improving fuel cell performance [19]. The significance of cathode design for PEM fuel cells in transport applications was deliberated and presented analysis of cathode material choice for maximizing power density [20]. Various applications and types of fuel cells, the importance of the PEM fuel cell and the current status and challenges of the fuel cells are discussed [21]. The authors opined that hydrogen fuel cells show great promise in transforming clean energy resolutions and justifying the injurious properties of outdated fossil fuel-based systems on the atmosphere. PEMFCs have a relatively high specific power above 1,000 W/kg and are used in transportation, Aerospace,

stationary power applications. PEM fuel cells play a vital role in Sustainable energy systems and ecofriendly sustainability. Current developments in mathematical modeling of Proton Exchange Membrane (PEM) fuel cells have presented inventive iterative methods for parameter estimation, improving the correctness of voltage-current characteristic estimates [22, 23]. The previous investigators focused their studies on the materials and designs of components such as the gas diffusion layer (GDL), gas diffusion electrode (GDE) and membrane and channel designs. The effect of porosity, temperature and reactant flow rates, humidification levels on performance of fuel cell are studied experimentally. The interactions between GDL, GDE and membrane under varying conditions remain underexplored. There are gaps in realizing the interactions under varying operational conditions. The integration of advanced computational models for multi-scale phenomena is still emerging with limited studies focusing on parametric analysis using COMSOL Multiphysics software. Therefore, comprehensive studies are required to understand the interactions among PEMFC components. The aim of the current study is to evaluation of the influence of the operating conditions on the performance of the PEM fuel cell using COMSOL software with the intension of recognizing optimal operating conditions which enhances the performance of fuel cell.

MATERIALS AND METHODS

The platinum is chosen as the catalyst for the electrode surface. The design parameters for the PEMFC gas diffusion layer and catalyst layer were chosen based on established thumb rules, ensuring a balance between performance and durability. The PEMFC is designed based on the empirical relations available in the literature [24]. Three equations are used to calculate the catalyst layer. Equation (1) defines the catalyst loading, Equation (2) relates the geometric area to the platinum-to-carbon ratio and Equation (3) calculates the active surface area based on particle size and density.

$$L_a = \frac{m_{Pt}A_o}{A_c} \quad (1)$$

$$A_o = 2.2779 * 10^6 \left(\frac{P_t}{C}\right)^3 - 1.5857 * 10^6 \left(\frac{P_t}{C}\right)^2 - 2.01538 * \left(\frac{P_t}{C}\right) - 1.595 * 10^6 \quad (2)$$

$$A_c = \frac{6}{d_{Pt} \times \rho_{Pt}} \quad (3)$$

The thumb rules were applied to select the design parameters based on existing literature [13, 14]. Diameter of platinum particles ranges from 2 nm to 7 nm for pure platinum and 3.5 nm to 15 nm for platinum alloy. Platinum Carbon ratio ranges from 0.1 to 0.6. The mass loading of platinum is between 0.033 mg/cm² to 0.15 mg/cm². The porosity of the catalyst layer varies from 0.1 to 0.5. The thickness of the

gas diffusion layer is in the range of 0.11mm to 0.37mm. The porosity of the gas diffusion layer ranges from 0.78 to 0.8. The anode side of the fuel cell is designed with a linear gas channel, the dimensions of which are determined analytically based on the catalyst layer's characteristics. The specific dimensions used for creating the fuel cell geometry including the GDL, GDE and electrolyte are outlined in Table 1. Figure 1 illustrates the three-dimensional model of the designed fuel cell, which integrates these components to simulate the fuel cell's operational environment.

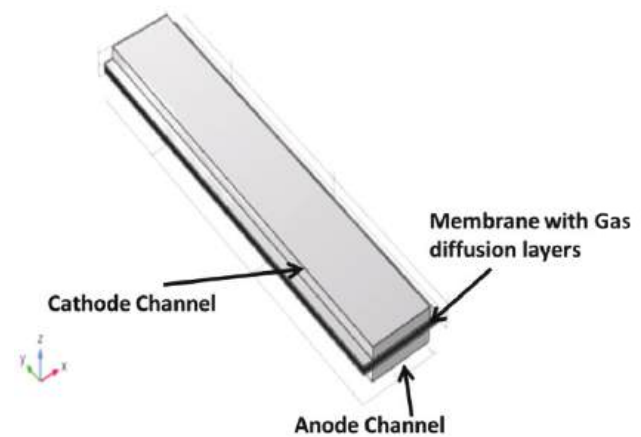


Figure 1. Model of the designed fuel cell for parametric study.

Table 1. Geometry and Operating parameters

S No.	Parameter	Value
1	Length of the channel (mm)	50
2	Channel height (mm)	1
3	Channel width (mm)	8
4	Rib Width (mm)	2.5
5	Thickness of GDL(μm)	190
6	Thickness of Membrane (μm)	127
7	Thickness of GDE (μm)	100
8	GDL Porosity	0.8
9	Conductivity of GDE (S/m)	222
10	Conductivity of Membrane (S/m)	9.825
11	Operating Temperature	800C
12	Inlet mass flow rate of H ₂ (kg/s)	0.1576
13	Inlet mass flow rate of O ₂ (kg/s)	0.6255
14	Mole fraction of H ₂ at inlet	0.96
15	Mole fraction of O ₂ at inlet	0.20
16	Mole fraction of H ₂ O at inlet	0.034
17	Mole fraction of N ₂ at inlet	0.76
18	Anode stoichiometry	2
19	Cathode stoichiometry	4



Figure 2. Meshed image of the modeled fuel cell.

In COMSOL multi-physics software, the computational domain of PEMFC is discretized using hexahedral elements and the mesh generated is depicted in Figure 2. The use of hexahedral mesh for the simulation provides good convergence and good results to capture the flow, heat and mass transfer physics. The computational domains include the air and fuel flow channels, as well as the GDL, GDE, and membrane. The geometry of fuel cell is a rectangular geometry and it is convenient to employ the structured mesh in computational domain. The number of elements used for fuel cell in computational domain is 1876.

Three-dimensional computational analysis is carried out by assuming steady flow conditions and the inlet gases are considered as ideal gases. The flow is presumed to be smooth and the fluid is immiscible. In the analysis of the PEM fuel cell, the flow is assumed to be laminar, considering the operating conditions and the dimensions of the flow channels. The working fluid is modeled as incompressible, as density variations due to pressure changes are negligible. The porous GDL, catalyst layers, and membrane are

treated as isotropic, implying uniform material properties in all directions. Additionally, the thermal properties of the materials are considered constant throughout the analysis and the membrane, catalyst layers, GDL are assumed to be homogeneous, ensuring uniform structural characteristics for simplification and consistency in the numerical model. The computational analysis of the PEMFC was conducted using a coupled approach integrating the free and subsurface flow, species transport, and current distribution modules. The free and subsurface flow module simulated the flow dynamics of the species under specified boundary conditions such as pressure and velocity, while the species transport module accounted for chemical reactions based on a defined diffusivity matrix. The current distribution module calculated the current density generated from the electrochemical reactions. A coupled analysis of these modules was performed to deduce the power density from the polarization curve. The numerical solution employed governing equations that encompassed the conservation of mass, momentum, energy and species. The solver also utilized the Butler-Volmer equation, Joule heating equation, and Nernst equation. This framework helped the modeling of reaction kinetics and cell performance [25-31]. The study takes into account the species transfer tool, current distribution node, open and permeable medium flow plugin, and fuel- cell component. The flow component is focused on the movement of substances in accordance to specific model parameters like pressure, velocity. The fuel-cell engine is in responsibility of connecting the electrochemical interactions with the other programs boundary conditions. The chemical reactions that take place for the selected diffusion matrix are under the control of the species transport module. The energy distribution element determines how much current density is produced by the process. As a consequence, a connected analysis of every module in the application was performed and the characteristic plot is used to evaluate the overall output of fuel cell. The materials selected for the Proton Exchange Membrane Fuel Cell (PEMFC) include copper for the current collectors due to its excellent electrical conductivity, which ensures efficient current collection and transport. For GDLs and GDEs, porous graphite is chosen for its high porosity, enabling efficient gas distribution and water removal. Additionally, platinum is coated on the GDE to

Table 2. Parameters varied for analysis

S No.	Parameter	Value
1	GDL Porosity	0.6 to 0.8 in the steps of 0.1
2	Operating Temperature	60 °C to 80 °C in the steps of 10 °C
3	Membrane Conductivity	5 S/m to 11 S/m
4	Membrane Thickness	Nafion 115,117 and 1110 membranes
5	Catalyst layer thickness	100µm to 60µm in the steps of 20µm

enhance catalytic activity for both the hydrogen oxidation reaction and oxygen reduction reaction. The membrane material selected is Nafion 117, a sulfonated tetrafluoroethylene-based copolymer, which offers superior proton conductivity, chemical stability, and mechanical strength, making it ideal for PEMFC applications. A set of parameters, as outlined in **Table 2**, has been chosen for a systematic study of the fuel cell's performance under various operating conditions, enabling an in-depth analysis of its efficiency and overall effectiveness.

RESULTS AND DISCUSSION

Validation

The following are the geometrical parameters used for the geometric model of the PEMFC. The width and depth of the anode and cathode flow channels are 50mm and height is 10mm. The thickness of gas diffusion layer and gas diffusion electrode is 190 μm and 10 μm respectively. Thickness of the membrane is 127 μm . Thickness of the current collectors is 2 mm. The PEMFC model was created in COMSOL software for validation is depicted in Figure 3.

The computed V-I characteristics of the PEMFC were compared with the experimental data [9] at 80°C and RH 80% as shown in Figure 4. The characteristic curve demonstrates the current density variation with the voltage for both the experimental and computational results. The results establish a strong agreement with the experimental findings. However, the minor deviations are noticed due to the stimulation over potentials govern at low current densities because of slow electrochemical reaction rates. This validated computational approach was used for the subsequent parametric analysis.

Polarization curve

The plot depicted in Figure 5 is a polarization curve of a fuel cell, shows the dependency between the current

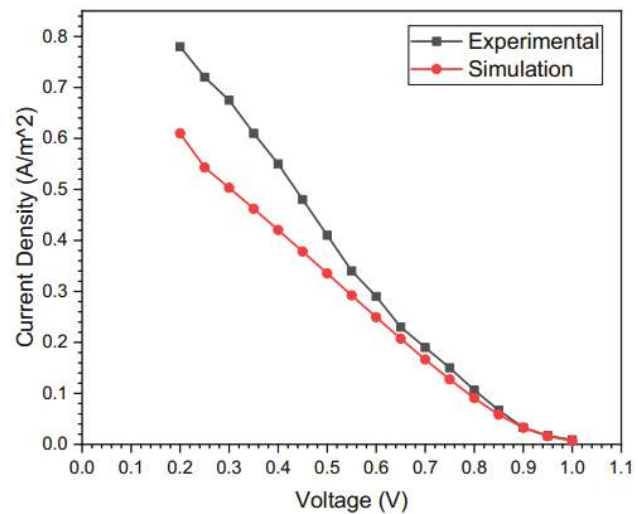


Figure 4. Validated V-I Characteristics.

density and voltage within the operational voltage range of a fuel cell. The data indicates that the proposed fuel cell generates higher power outputs specifically within the voltage range of 0.3 to 0.5 V. The slope of the curve is more at low current densities due to activation losses caused by sluggish reaction rates. The voltage decreases slowly with the increase in current densities and this trend indicates that the prevalence of ohmic losses arising from resistance to ion transport in the electrolyte and electron flow in the cell [3].

Electrode potential

The contours of electrode potential are depicted in Figure 6. The cell voltage differential is greater at the middle of the cell than the ribs. This is because the interaction of the reagents occurs at the center of the cell, whilst the reaction at the ribs is gradual.

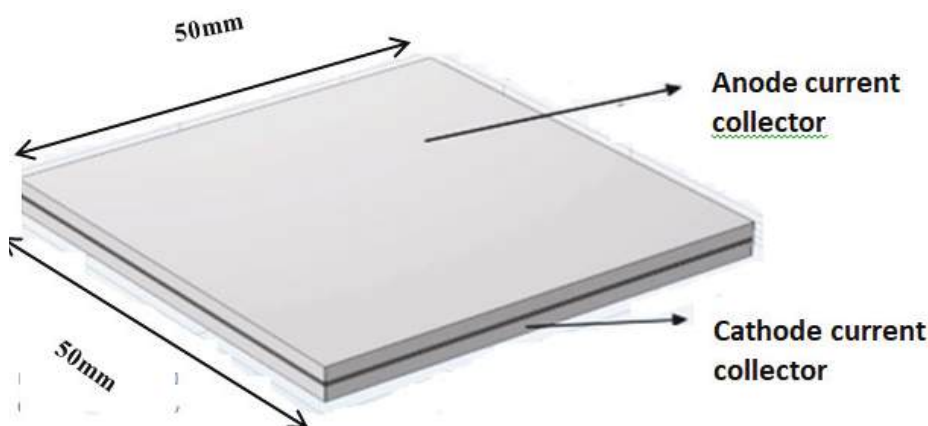


Figure 3. Fuel cell model employed for validation.

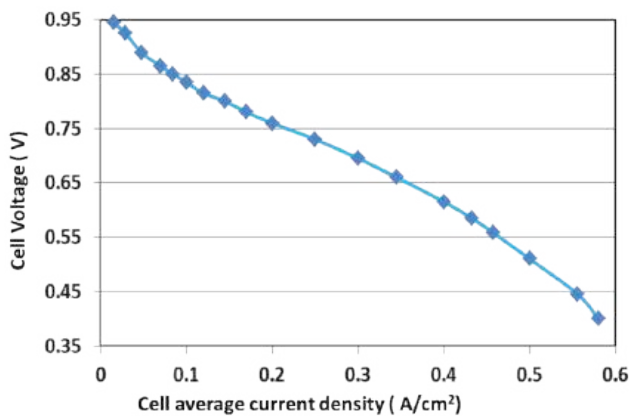


Figure 5. V-I Characteristics of the designed fuel cell.



Figure 6. Contours of electrode potential of the fuel cell.

Mole fraction of hydrogen and oxygen

The hydrogen mole fraction decreases progressively from inlet to outlet of the fuel cell. This reduction is attributed to the consumption of hydrogen in electro-chemical reactions on the anode side, where it contributes to generating electricity. As hydrogen molecules react, protons move across the electrolyte membrane and the electrons are directed along the external circuit an electric current is produced. Figures 7 and 8 illustrate the hydrogen and oxygen mole fractions through the cell. The drop in hydrogen mole fraction in Figure 7 confirms its utilization during the reaction process. Figure 8 shows the oxygen mole fraction trends, which mirror those of hydrogen on the cathode side. Oxygen consumption increases towards the output as it reacts with protons and electrons to form

water, completing the electrochemical process and confirming effective reaction kinetics within the cell.

Velocity profiles

The simulated flow within the fuel cell operates under laminar conditions, as evidenced by the velocity contours shown in Figure 9 and Figure 10 at cell voltage is 0.5 V.

These contours illustrate that velocity reaches its peak at the center of the flow channel, where the reactant gases have unobstructed movement. This high central velocity facilitates effective passage of reactants to the reaction sites, crucial for sustaining the electrochemical reactions. At the channel edges, however, the velocity of the reactant gases drops to zero, adhering to the no-slip condition imposed by the channel

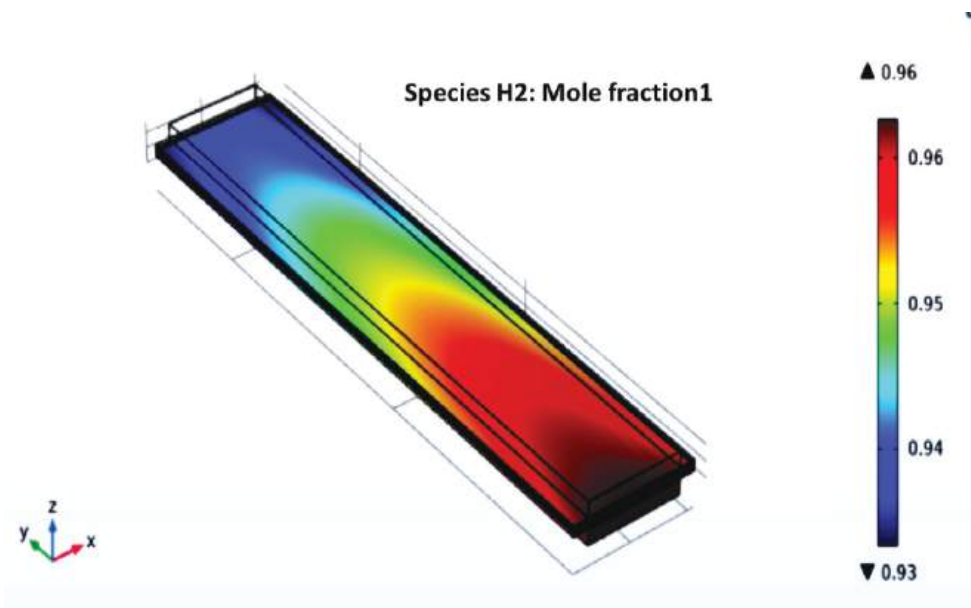


Figure 7. Hydrogen mole fraction contour of designed fuel cell.

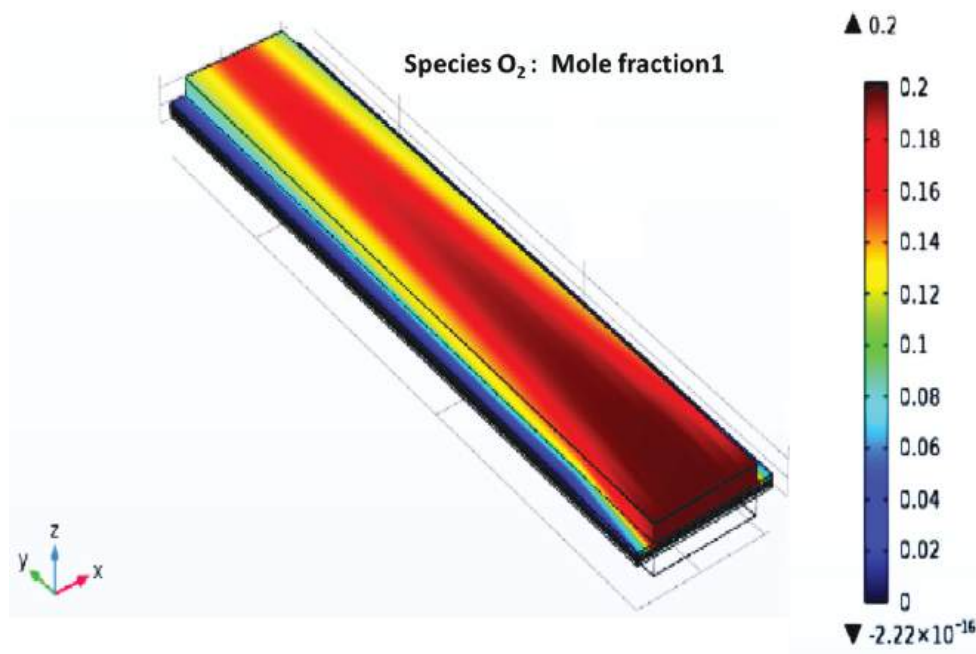


Figure 8. Oxygen mole fraction contour of designed fuel cell.

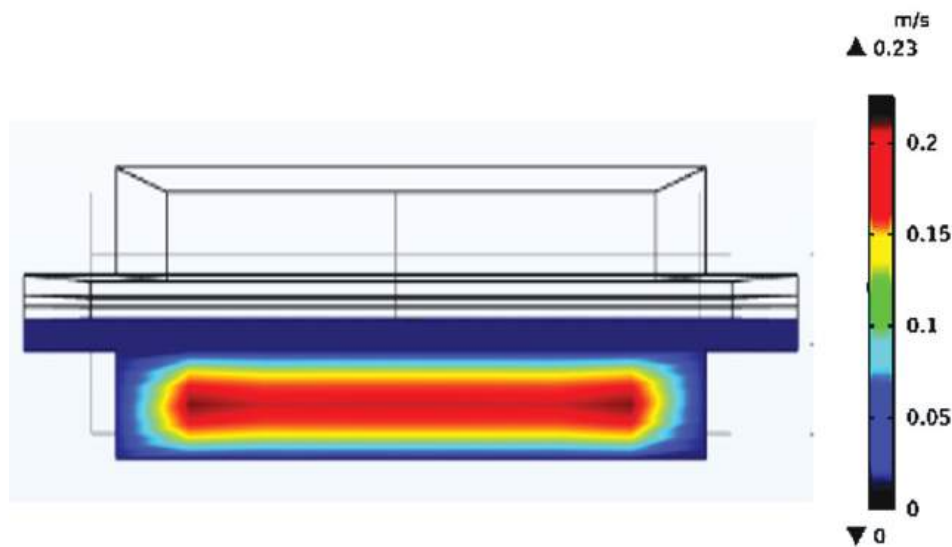


Figure 9. Velocity contours of fuel cell on anode side.

walls. This boundary condition leads to a steady reduction in velocity from the channel center to the walls, characteristic of laminar flow profiles. The resulting velocity gradient not only impacts reactant distribution but also influences mass transfer rates, highlighting the role of channel geometry and flow conditions in optimizing fuel cell performance.

Effect of GDL porosity

Figure 11 presents the fluctuation of current density with voltage, demonstrating that fuel cell effectiveness is

positively influenced by higher porosity values of the GDL, particularly at lower operating voltages. At these voltages, the effect of GDL porosity is more pronounced, as greater porosity facilitates enhanced gas diffusion. However, it leads to uneven distributions temperature inside the cell. These changes influence the local kinetics of reaction, as higher temperatures enhance electrochemical reactions rate, while lower temperatures can decrease reactant consumption and the performance decreases. Thus, the interaction between GDL porosity and temperature changes is important for

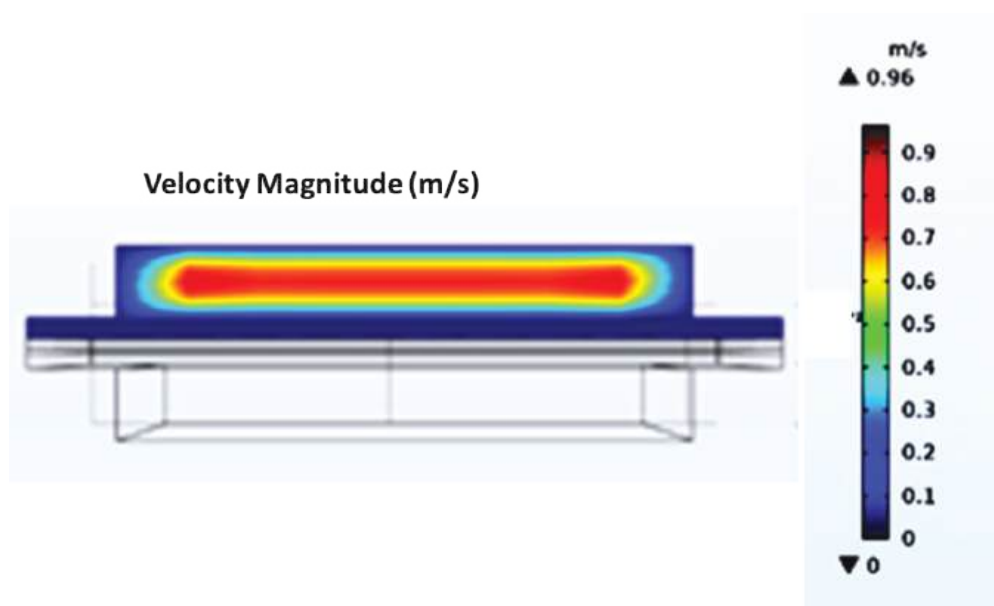


Figure 10. Velocity profile of fuel cell cathode side.

optimizing cell performance This improved diffusion allows for a more consistent and ample supply of reactant gases, such as hydrogen and oxygen, at the reaction zones, thus increasing the generation of electrons and improving overall cell efficiency. This phenomena also noticed by Larbi et al [3]. This is likely due to the reduced role of diffusion limitations at higher voltages, where other factors such as ohmic resistance and reaction kinetics become more dominant. Consequently, optimizing GDL porosity is especially beneficial at lower voltages, where it plays a crucial role in maintaining efficient fuel cell operation by ensuring a steady supply of reactants to the electrodes.

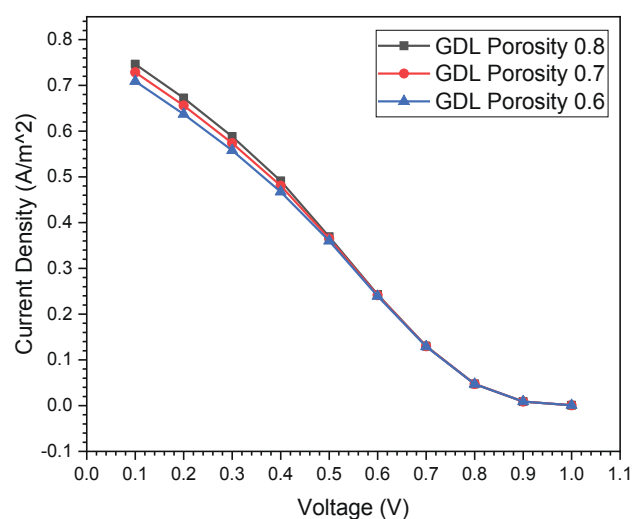


Figure 11. Polarization graph at different GDL porosity.

Effect of Operating Temperature

Figures 12 and 13 illustrate the voltage-current (V-I) characteristics and power density variation with voltage across a temperature range of 60 to 80°C. The V-I curves and power density profiles display only slight changes over this temperature range, indicating that the PEM fuel cell (PEMFC) maintains stable performance within these operational temperatures. This consistency suggests that moderate temperature variations between 60 and 80°C do not substantially affect the electrochemical reaction rates or the overall efficiency of the PEMFC. As a result, the fuel cell can function effectively within this temperature range

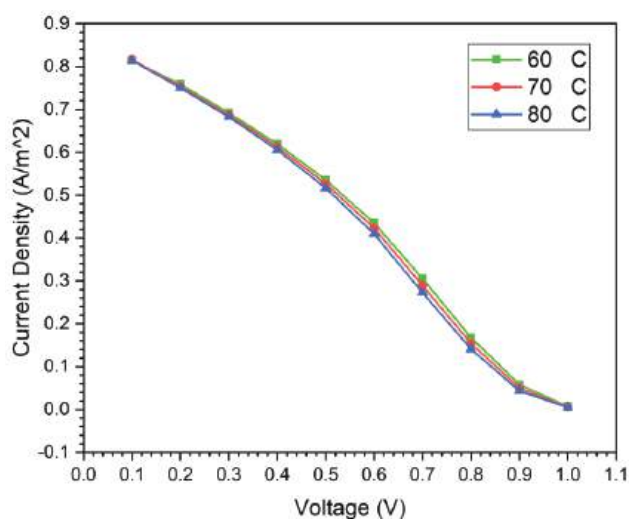


Figure 12. Polarization graph for the effect of operating temperature.

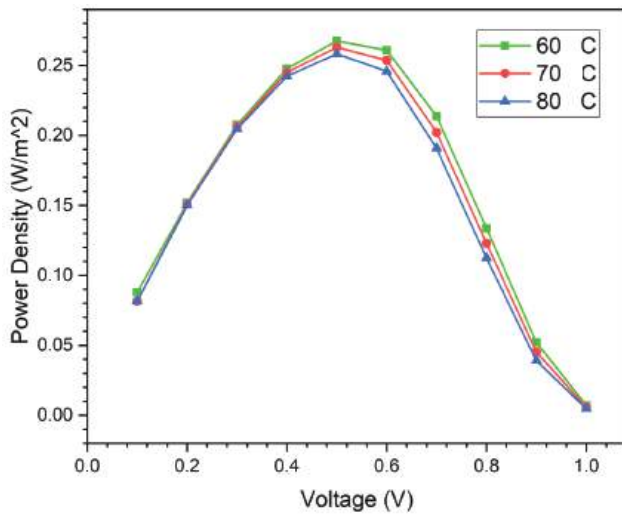


Figure 13. Power density curve for the effect of operating temperature.

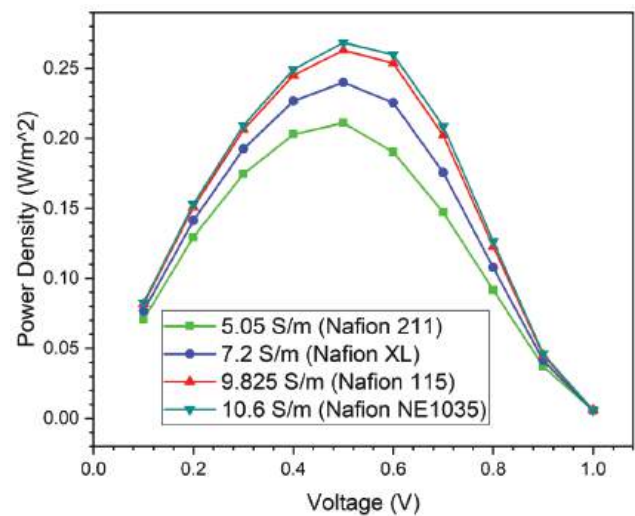


Figure 15. Power density curve for the effect of membrane conductivity.

without the need for intensive thermal regulation. This stability is advantageous for practical applications, where maintaining precise temperature control may be challenging, reinforcing the robustness of PEMFCs in various operating environments.

Effect of Electrolyte Conductivity

The variation of current density with voltage of different Nafion membranes is typically analyzed to realize the dependence of ionic conductivity on fuel cell effectiveness. The conductivity of Nafion membranes is a critical factor because it influences the ionic transport of protons from positive electrode to negative electrode, a key process in

fuel cell operation. Higher conductivity means better proton conductivity, which allows for more efficient charge transfer and improved fuel cell performance. The comparison of the Nafion membranes shows a clear trend depicted in Figure 14 and Figure 15. Higher ionic conductivity leads to better fuel cell performance. The fuel cell demonstrates improved performance as conductivity increases, due to enhanced proton mobility that reduces internal resistance and improves power output. This trend is consistent with findings [11,12,14]. Therefore, selecting a membrane with higher conductivity is essential for optimizing fuel cell efficiency, especially in applications requiring high power densities.

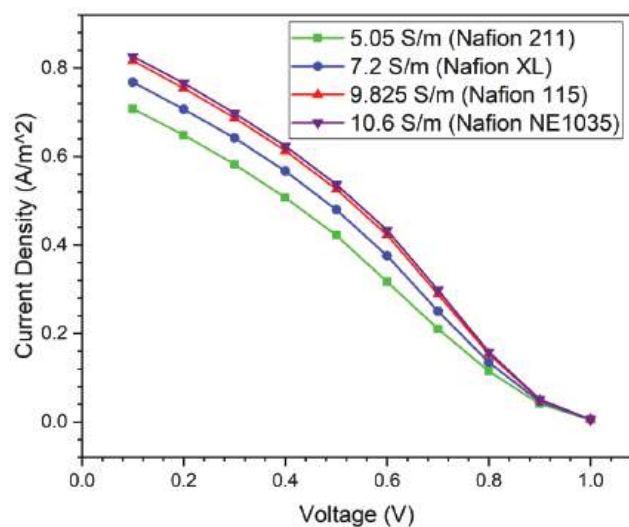


Figure 14. Polarization graph for the effect of membrane conductivity.

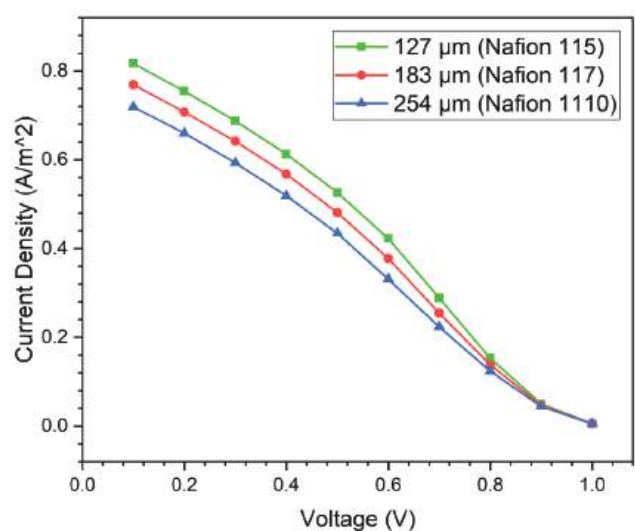


Figure 16. Polarization graph for the effect of membrane thickness.

Effect of Electrolyte Thickness

The performance characteristics demonstrated in Figure 16 show that thinner electrolytes provide better fuel cell performance and the same trend was observed by Kösters et al. [13]. Because an increase in thickness causes an increase in resistivity, a barrier is formed against flow of protons from the positive electrode to negative electrode. Thus the effectiveness of the cell decreases.

Effect of Catalyst Layer Thickness

Figure 17 and Figure 18 depicts the characteristics curves. The thickness of the catalyst layer has very negligible effect. As a result, this is not a major parameter for improving fuel cell performance. The plot illustrates the fluctuation of current density with voltage for three different catalyst layer thicknesses of 100 μm , 80 μm and 60 μm .

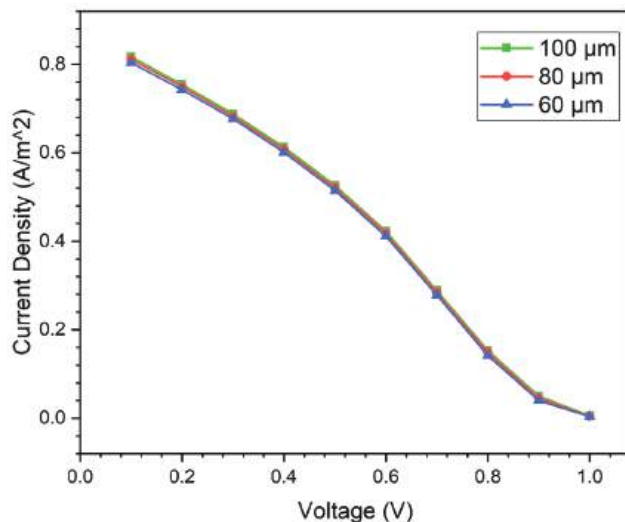


Figure 17. Catalyst layer thickness influence on V-I curve.

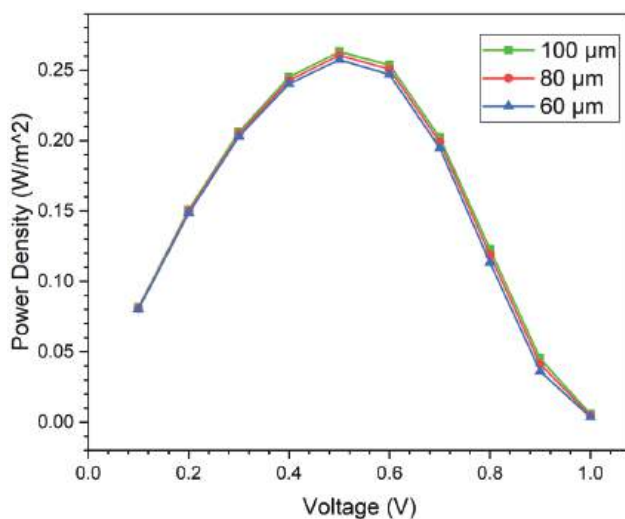


Figure 18. Variation of power density with voltage.

The present study offers some different contributions in the performance analysis of fuel cell. COMSOL Multiphysics software is employed for the systematic parametric study of fuel cell. This study provides the understandings into their specific effects on fuel cell performance. The computational analysis considered the design variables and evaluated their combined influence, which has not been extensively addressed in prior research. The results indicate that, a substantial 26.54% increase in power density with improved electrolyte conductivity, providing new insights into the optimization of fuel cell efficiency.

CONCLUSION

The following are the key conclusions from the parametric analysis of PEM fuel cell.

This research work provides valuable insights for designers, emphasizing electrolyte conductivity, thickness, and gas diffusion layer porosity as the most significant parameters for optimizing PEM fuel cell performance.

An increase in electrolyte conductivity from 5.05 to 10.6 was noticed to improve power density by 26.54%, while a decrease in electrolyte thickness from 254 μm to 127 μm resulted in a 21.19% development.

A gas diffusion layer with higher porosity is recommended to enhance fuel cell performance.

The fuel cell achieves peak performance at an operating voltage of 0.5V under all parameters studied

The operating temperature and catalyst layer thickness have a minimal impact on performance.

This study enables proton exchange membrane fuel cell in sustainable transportation, portable power systems, renewable energy integrated systems, marine and aeronautical and space applications.

This study contributes to improving PEM fuel cell performance, crucial for achieving global sustainability targets, as it encourages the adoption of clean energy technologies for efficient and pollution-free power generation.

NOMENCLATURE

L_a	Catalyst loading, mg/cm^2
m_{pt}	Mass of platinum catalyst, g
A_c	Active surface area of the platinum catalyst, mm^2
A_o	Geometric area of the catalyst layer, mm^2
d_{pt}	Diameter of the platinum particles, nm
ρ_{pt}	Density of the platinum, kg/m^3
P_t/C	Platinum to Carbon ratio

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

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Research Article

Experimental evaluation of hybrid electric-heat operation for reliable residential water heating technology using photo-thermal system

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ABSTRACT

Photovoltaic (PV) is equipped with additional cooling system to reduce the cell temperature and enhance its efficiency. Numerous studies have addressed the issue using various methods such as thermoelectric, passive, and active cooling. In this work, the excess heat is taken as additional energy input considering the trend of PV direct heating system. The present study introduces PV and thermal collector as one compact system for heating application. PV output is connected to heating element while thermal collector harvests the waste heat from PV module. It allows the system to accumulate higher effective heat energy, which is useful for residential water heating applications. Experimental work is performed using 50 Wp monocrystalline PV by varying the filler material (sand and grease) inside the thermal collector and flowrate of the working fluid. The effect of filler material is observed as rapid heat dissipation, which is obtained by sand. Also, coolant flowrate affects the maximum temperature output and the accumulated heat energy from PV module. Despite the variation, the cooling effect from thermal collector reduces the PV temperature up to 30.1% and enhances the electric generation by 15.1%. The collected heat energy reaches a maximum value of 129.3 Watts, while peak electricity output only 46.4 Watts. The heated water from PV output reaches maximum temperature of 75 °C with maximum coolant temperature of 48.7 °C. It shows that the system is highly reliable for residential heating, offering a compact PV-thermal collector with higher power output.

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INTRODUCTION

Despite population growth on global scale, utilization of fossil fuel transforms into cleaner energy. It shows serious

acts to mitigate severe climate change. The engineering process for utilizing biomass and renewable sources [1–3] indicates positive achievement on higher portion of renewable

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sources in energy mix. However, global projection reveals that annual energy consumption tends to increase [4], so further effort is required to maximize the potential of clean energy sources, particularly for building sector. The sector is ideal for being role model in the application and utilization of clean energy technology to meet the local demand, such water heating. Moreover, residential heating consumes enormous amounts of energy [5]. It makes additional improvements in supplying clean energy for the sector is essential.

Solar energy is essential for supplying hot water in residential housing through solar thermal technology [6]. It uses specific thermal collector to enhance the energy intake from the sun [7]. The system reliability is enhanced by coupling the collector with thermal storage, which utilizes latent and sensible heat of the material [8–10], making the system operate at longer duration with higher capacity. Moreover, recent trend indicates alternative option to operate photovoltaic (PV) as input energy for heating purposes. The work [11] assessed the potential of PV for the heating sector, implying that around 48% of the required energy for heating purposes can be supplied by PV. The concept offers one significant advantage regarding novel configuration to store the excess energy production from PV. Different assessment was taken [12] for the idea, emphasizing the positive contribution of PV for supplying heating demand in residential sector under suitable control process. Thus, it adds a new option for applying PV system specifically designed for hot water generation.

PV operates using photoelectric, which converts sunlight into electrical energy. The process increases the surface temperature (SUT) of the PV. It is considered waste heat for the system operation [13]. It disrupts the conversion process, significantly decreasing the PV efficiency. The system requires additional cooling system to maintain the optimum working temperature of PV. The basic method is passive cooling, which operates without working fluid [14]. It is located at backsheet (BS) assembly of PV module. M. B. Elsheniti et al. developed heat management under passive mode operation for PV modules, resulting in a higher power output around 5.75% compared to the reference module [15]. Increasing the cooling capacity was conducted using dual model cooling system, achieving a better thermal and electrical efficiency at 24.3% and 0.16% [16]. It shows that heat management is essential to prevent inefficient output from PV modules.

Another method is developed specifically concerned with the operating condition of PV cooling system. H. Chen et al. evaluated intermittent cooling for PV modules with nanofluids, indicating that electric generation efficiency increases 19.7% [17]. Q. Wang et al. employed heat pipes and iron-oxide nanofluids for PV cooling system, showing the SUT reduces around 20.4 °C [18]. Adjustment of the type of nanofluid and operating system contributes notably to enhancing the capability of the system, as summarized in this work [19]. The heat exchange is the key

aspect to ensuring the effective cooling process. Installing additional fin at BS enhances the heat exchange, particularly under suitable fin orientation [20]. Also, different cooling mechanism is proposed by employing magnetic field. Numerical optimization was performed here [21] by introducing an additional magnetic field for the system, revealing a better heat distribution corresponding to higher cell efficiency. It demonstrates the urgency of performing a cooling mechanism to maintain the effective output of PV modules. Despite that, the usability of nanofluids and extensive equipment may result in high installment cost, reducing the net energy balance and potentially bringing negative drawbacks at night [22].

The accumulated waste heat from PV operation is another form of solar energy potentially harvested as additional energy input. Thus, several works focus on utilizing the energy through thermoelectric (TE). This study [23] conducted numerical analysis for PV cooling system with TE, demonstrating the potential of cell temperature decrement about 2.9 °C with increment of power generated by TE. Combined PV-TE was evaluated numerically [24], which showed the potential of cell temperature decrement for the proposed model and the ability to increase PV efficiency slightly (1.82%). Different work [25] conducted experimental process for TE concept as PV cooling method, showing the cell temperature drops by 3% with improvement in the system efficiency by 13.9%. Comprehensive numerical and experimental assessment [26] indicated that cascaded PV-TE offers higher cooling function with suitable power production, making the system ideal for alternative PV-TE assembly. Nevertheless, employing TE is still in development phase since the net produced power is insignificant (less than 5%) [27]. Therefore, effort to maximize the potential of waste heat from PV while simultaneously reducing the system temperature is still developed to achieve a more feasible method.

The produced heat from PV can be harvested and stored to material with heat storage capability such phase change materials (PCM). It can store the heat under certain working temperature [28–30]. A. Hamada et al. investigated PV-thermal (PVT) with PCM, indicating the cumulative achievement of system efficiency using passive PVT-PCM reached 74.1%, which was higher than system with no PVT at 34.6% [31]. Advance modification was taken with nano-PCM, which improved thermal efficiency and electrical power by 83.3% and 44.5% [32]. Optimization for the PVT-PCM was done by incorporating composite material, showing reliable improvement for continuous operation of the system for suitable regions [33]. Moreover, a different model was studied numerically to accommodate different heat transfer from the surface. This work [34] employed dual PCM as passive heat management for PV modules. Numerical modeling demonstrated a significant achievement in the efficiency (35.8%). In general, positive achievement is obtained by integrating system with PCM.

Employing PCM offers reliability of the application PVT. The PCM is used as alternative storage medium. However, the system demands additional equipment to harvest the stored heat, which requires supplementary equipment for the operation. Also, thermodynamic limits come as serious barrier due to maximum temperature generated by PV surface. Thus, waste heat from PV might be used directly rather than stored in different mediums. It is also suitable for the current trend of utilizing PV heaters for heat generation, particularly for rich electricity regions [35]. A. Szajding et al. analyzed high capacity PV-heating system, indicating simplification operation for heat generation purpose using PV [36]. Optimization was done to adjust the suitable operating condition for hot water supply from PV system based on specific household demands [37]. Therefore, it is advisable to harvest heat from PVT, which can be combined with the PV-heating concept.

The possibility of employing PV-heating with waste heat accumulation as one combined system offers a new option for residential water heating development. Previous works imply that the system operates independently and is not considered one compact system. Thus, the present study proposes to evaluate the idea of combined PV-heating and waste heat accumulation. It is the main novelty of this work, which differs from previous studies. The work here focuses on harvesting waste heat from PV modules for hot water production by placing thermal collector (TC) at the BS area and simultaneously connecting the PV output to electric heater. As a result, compact PVTC (photovoltaic-thermal collector) is obtained as one robust assembly. Therefore, the objective is designed to introduce and provide preliminary evaluation of the operation of PVTC for hot water production. The mass flowrate (MF) and filler material (FM) of the TC are varied to understand the impact of the given aspect on the accumulated energy of the system. The temperature, electrical output, and accumulated energy are presented and discussed briefly to provide new perspective on the

development of this concept. We believe it offers significant contribution for the development of residential water heating and potentially maximizing the utilization solar energy for the sector.

MATERIALS AND METHODS

Manufacturing PVTC Unit

The specification of PV module is presented in Table 1. TC unit is an additional component placed at the BS surface of PV assembly. Locating the TC in the given region is aimed to absorb heat and transferring it to the cold water. The process causes cooling effect for PV cells at the surface area. In this work, TC unit was equipped with serpentine copper tube (OD/ID: 6.35 mm/5.23 mm) as coolant path. The serpentine form is chosen for its suitable performance [38] and technical simplicity.

Figure 1a shows the photograph of the serpentine tube. There is empty space between each pass which makes the assembly required FM. Considering actual implementation, the FM must be widely available, have excellent technical properties, and be economically feasible. Thus, this work employed sand and grease as FM for the TC assembly. Sand is taken as solid filler with an excellent nature conductivity compared to water [39–41], making it favorable to fill the

Table 1. Specification of PV modules

Model	50 Wp monocrystalline PV
Dimension (mm)	670 × 410 × 30
V_{mp} (Volt)	18.0
I_{mp} (Ampere)	2.78
V_{oc} (Volt)	22.4
I_{sc} (Ampere)	3.24

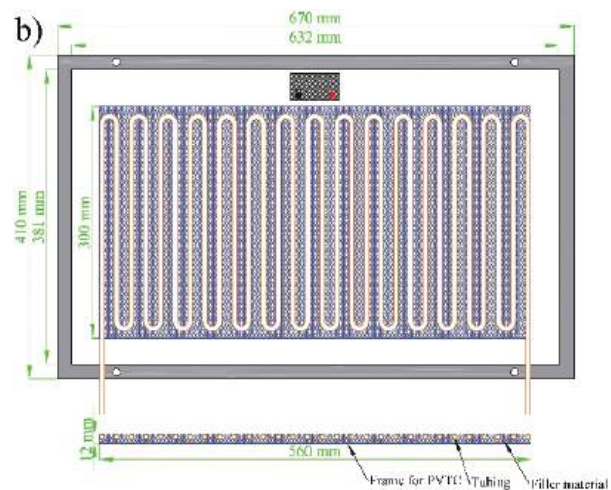


Figure 1. a) Photograph of serpentine tube and b) Dimension of the PVTC unit.

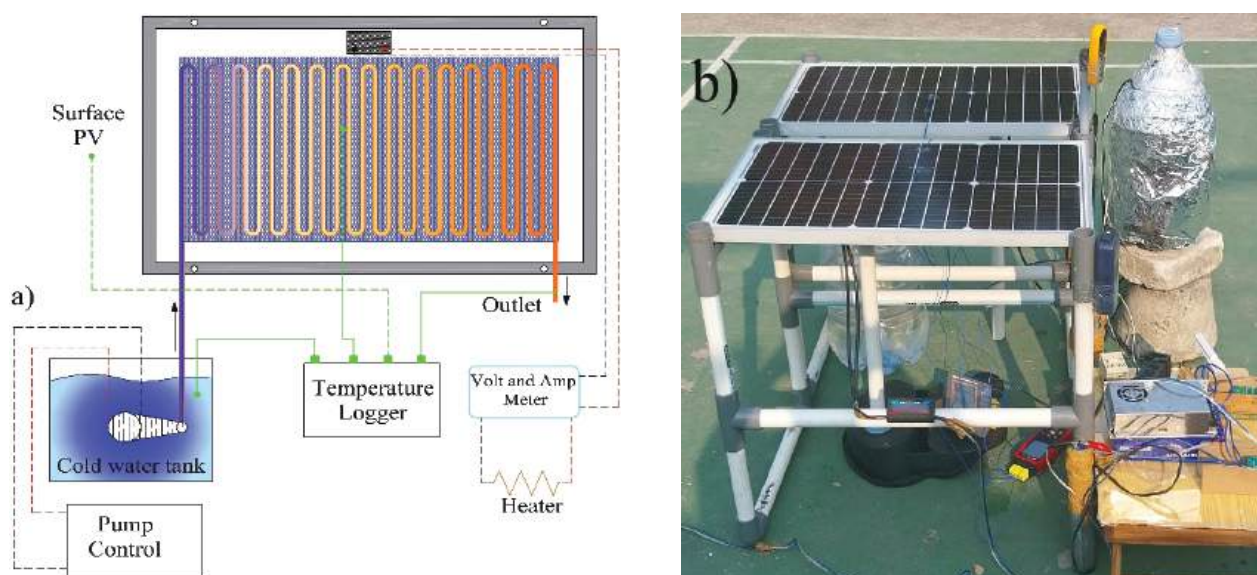


Figure 2. a) Schematic of the assessment and b) Photograph of the process.

empty space of the TC assembly. Sand with average mesh size 18 was used to maximize the filling ratio. Another FM for this work was thermal grease, which offers suitable contact area between each tube and BS layer [42]. In this work, grease class NLGI grade 2 was employed for the FM.

The detailed dimension of PVTC assembly is presented in Figure 1b. All component is placed in specifically designed frame made of acrylic. The tube was installed precisely to ensure direct contact with BS layer. The inlet/outlet tube was extended to accommodate the installation of additional hose for the working fluid. The system was examined for potential leakage and then installed at the BS layer to obtain one compact component as a PVTC.

Experimental Test

The assessment was conducted to observe the power output from PV and the amount of collected heat from TC. The schematic evaluation process is illustrated in Figure 2a. Cold water was distributed through TC assembly and collected heat from BS surface. Cold water temperature was maintained at 31 °C (± 1 °C). The temperature increment of the water after passing TC assembly was measured to

determine the input energy from TC. The SUT and BS temperature (BST) were recorded simultaneously during experiment. The PV was connected to heater, and the amount of produced electrical energy was recorded by I-V meter. The heater was placed in insulated storage with 7 liters of fresh water. The heated water was also measured throughout the process. Figure 2b shows the pictorial view of experimental process.

The evaluation was performed according to the variation of FM and MF of cold water. The designed name for each model is summarized in Table 2. Two systems were assessed simultaneously during each case. It was performed on October 2024 in West Java, Indonesia. The evaluated model, as presented in Table 2, is expected to provide additional information regarding the effect of FM and MF on the accumulated energy of system. Additionally, PVTC-SG was designed as mixture of sand/grease with ratio 75/25 (by volume) to observe the potential of the combined version of the two fillers.

On-field measurement was performed based on the schematic presented in Figure 2. The process was conducted

Table 2. Specific test model for evaluated system

System Name	Filler Material	MF ($\times 10^{-3}$ l/s)	Designated Case	Date
PVTC-S	Sand	4	Case 1	06/10/2024
PVTC-G	Grease	4		
PVTC-SL	Sand	2	Case 2	08/10/2024
PVTC-SG	Sand/Grease	2		
PVTC-SH	Sand	6	Case 3	10/10/2024
PV	-	-		

Table 3. Measurement properties of weather and apparatus

Aspect	Properties	Device	Accuracy	Range error (%)
Local weather	Solar radiation	SM206	0.1	5
	Air speed	UT363	0.1	3.5
	RH	HTC-2	1	8
	Air temperature		0.1	6
PV	Current	DC-150A	0.01	6
	Voltage			6
System	Temperature	Thermocouple (K)	0.1	2.2

from 08 AM to 03 PM (local time). As seen in Table 2, each test was taken at intervals one day due to weather condition on subsequent days. The local weather condition and all measurement equipment used in this work are summarized in Table 2. Each component has range error to indicate the uncertainty range from the measurement.

RESULTS AND DISCUSSION

Weather Conditions

The weather measurement was performed for each case since it affects the obtained result for evaluating the system (Table 2). The solar radiation profile (Fig. 3a) shows that peak radiation above 1000 W/m^2 occurs between 10:50–12:30. The average radiation for all cases is obtained between $769 - 827.2 \text{ W/m}^2$ with maximum radiation at 1273, 1178.5 and 1267 W/m^2 for evaluation at case 1, case 2 and case 3, respectively. The radiation values demonstrate the average condition for clear sky with intensity above 1000 W/m^2 [43]. The recorded air speed is considerably low during the test, with an average air speed less than 1 m/s .

Another weather parameter is ambient air temperature and RH (Fig. 3b). The average temperature for all cases ranges around $35.9 - 37.7^\circ\text{C}$, with peak temperature appearing in the midday along with peak solar intensity. The air was relatively dry during the experiment, according to the recorded RH. Despite the variation, the lowest RH is $29 - 31\%$, with average value between $34.4 - 38.5\%$. The recorded weather condition provides essential information to understand the obtained performance for the tested PV and PVTC. Moreover, the dynamic condition emphasizes that the actual measurement shows significant variation, which is more relevant for actual evaluation than in lab measurement. The error measurement from the metering device presented in Table 3 is provided to give additional consideration regarding the value obtained from this assessment.

PV Module Temperature

The direct exposure from the sun to PV module elevates its SUT notably. Figure 4 shows the recorded SUT, which clearly emphasizes the high temperature condition of PV module during the operation. The accumulated heat

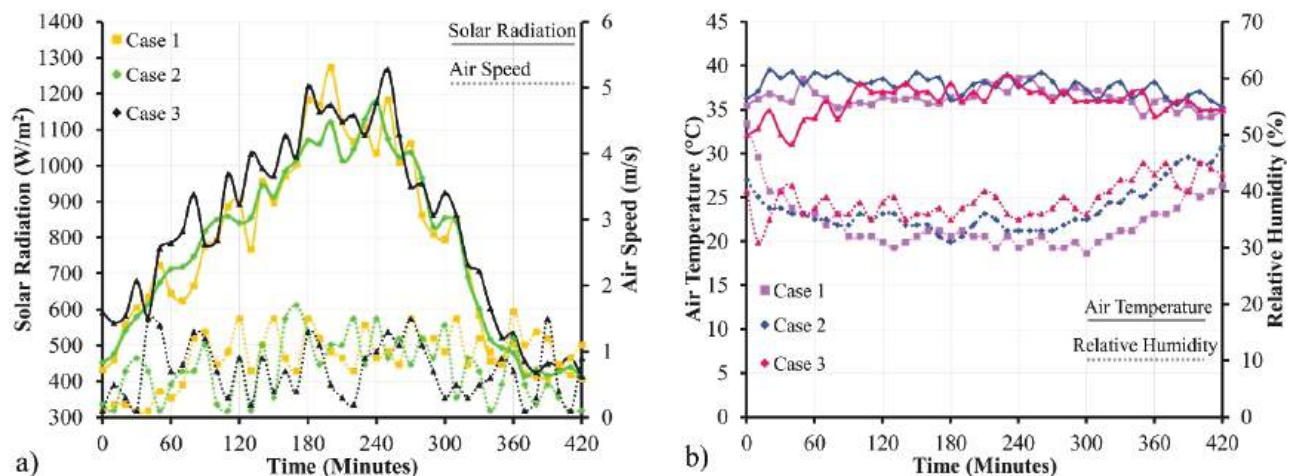


Figure 3. Weather condition on the experimental test: a) Solar radiation and air speed, and b) Air temperature and relative humidity.

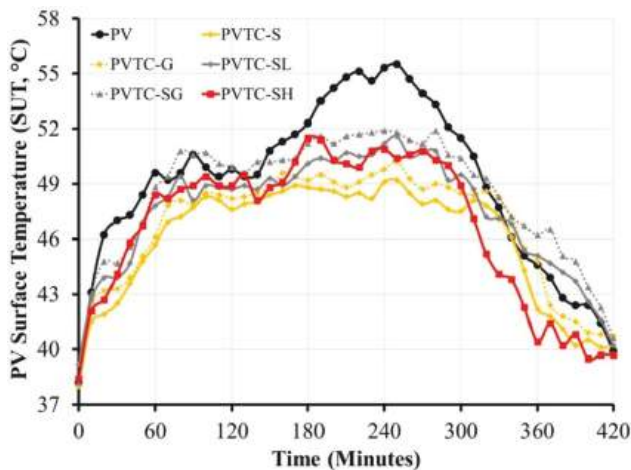


Figure 4. The SUT for all tested PV and PVTC assembly.

increases the SUT of PV module up to 55.5 °C with an average 49.1 °C. The specific increment in the initial operation is obtained at 4.7 °C/hour, then continues to increase steadily until the peak point at midday. The weather affects this high SUT, corresponding to the average ambient air temperature and low RH (Fig. 3b). The cooling effect from the ambient air is insufficient due to low air speed (Fig. 3a), making the average SUT higher (around 11.3 °C) than ambient temperature.

The presence of TC for the PVTC contributes positively to managing extreme SUT. The recorded SUT for PV modules decreases between 12.7–17.3% with TC. It confirms the benefit of using TC for PV, as it can achieve a lower SUT. Moreover, the FM plays different cooling effect for the PVTC system. PVTC with sand offers a higher heat dissipation from the PV module. It decreases the SUT by 8.8 °C (PVTC-S) and 9.6 °C (PVTC-SH). Lowering the MF for the same FM reduces the average heat dissipation around 8.1 °C. Moreover, the combined S/G for PVTC-SG has drawbacks due to the highest SUT (48.5 °C) among the other PVTC. In contrast, PVTC-G offers a relatively better SUT around 46.6 °C. The designed variation in this work shows relevant result to choose suitable FM and operating condition, making the aspects crucial to maximize the heat dissipation rate from PV modules.

The accumulated heat from the surface is transferred to the backsheet layer. As seen in Figure 5, the backsheet zone has remarkably high temperature, which is affected by the relatively thin layer assembly of typical PV modules [44]. Specifically for PV, the BST has identical temperature with the SUT, with average and maximum temperature of 46.7 °C and 55 °C. Adding additional TC effectively reduces the backsheet temperature, indicating a suitable cooling effect from the component. As observed, all cases show an optimum heat dissipation, which reduces the BST at lower temperature around 11.28–16.52% than PV.

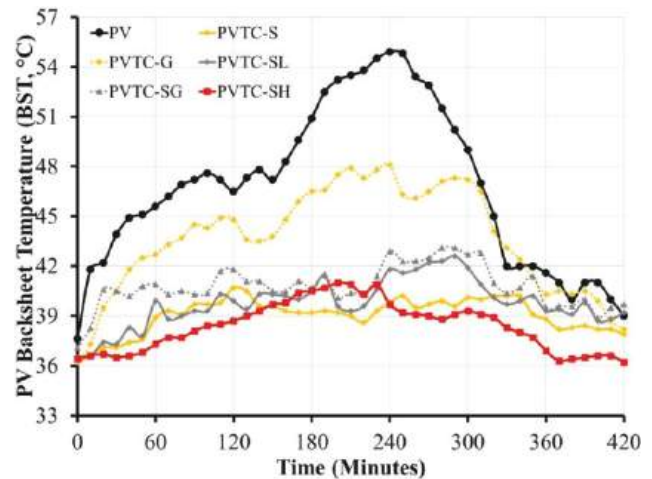


Figure 5. The BST for all tested PV and PVTC assembly.

Using different PVTC assembly directly affects the achievement of cooling effect for the BST. For example, PVTC with sand demonstrates suitable temperature decrement around 15.87% for PVTC-S. Modifying the MF for the same assembly (PVTC-SH) boosts the heat dissipation rate, reducing the temperature at relatively higher value (16.52%). It achieves the lowest recorded BST at 38.4 °C. Unfortunately, changing the filler with grease makes unfavorable cooling effect where the temperature at BST reduces only 11.3 °C relative to PV. It is minimized by adding sand (PVTC-SG), which can provide a higher heat dissipation, reducing the BST up to 14.2 °C even though it operates at lower MF. Regardless of the result, PVTC-G remains better at reducing the BST than PV without TC.

The ability to provide heat dissipation for the PVTC is observed according to the temperature deviation between SUT/BST (Fig. 6). PV has a lower gradient, indicating the heat transferred rapidly from surface to backsheet region.

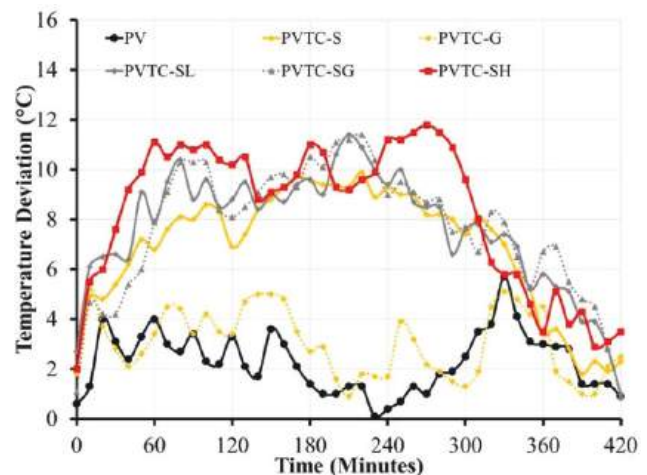


Figure 6. Temperature deviation between SUT/BST.

The low cooling effect for PVTC-G indicates the same behavior with average deviation of 3 °C, which varies insignificantly from PV (2.3 °C). Accordingly, it confirms the unsuitable heat dissipation rate, which might be affected by low conductivity of the grease. All PVTC with sand shows an excellent result, removing the heat effectively, which enhanced the temperature gradient between SUT/BST. The peak gradient is achieved by PVTC-SH (8.3 °C) and PVTC-S (6.9 °C). The low MF for PVTC-SL and PVTC-SG has similar value around 7.6 °C. The profile demonstrates the capability of the PVTC to reduce the BST, which eventually decreases the SUT (Fig. 4).

Generated Power from PV Module

The electric generation from PV occurs as unsteady process, as observed for the obtained V-curve in Figure 7a. It needs auxiliary component (charge controller) to maintain the steady voltage output. However, it comes with additional cost and potential power decrement [45]. Thus, the output PV is connected directly to the load to prevent additional power consumption and installation cost. Moreover, the heater is designed as the sole component that neglects voltage variation when generating heat. The plotted V-curve indicates the positive contribution of the addition of suitable PVTC configuration. The minimum voltage for PVTC increases around 7.4% (PVTC-S), 7.3% (PVTC-G), and 6.3% (PVTC-SH). According to [46], temperature increment results in a higher voltage drop. Thus, the obtained higher voltage for the PVTC demonstrates the positive contribution to temperature decrement of the module. However, exceptional result is observed for PVTC-SL and PVTC-SG, which indicates relatively identical value with PV. It is related to poor cooling effect for both systems. Additionally, external factors from weather conditions are also considered for this justification.

The I-curve is plotted in Figure 7b, showing more apparent implications on the impact of cooling effect on

the PV module. The obtained current for PV throughout the evaluation is 1.93 Ampere. It increases by 15.1% for PVTC-SH. Both systems were tested on the same day (Table 2), neglecting the weather factor and proving the positive achievement for implementing suitable PVTC. In addition, increment in the current is also achieved by PVTC-S (6.3%) and PVTC-SG (8.1%). It improves at relatively higher value for the PVTC-SL and PVTC-SG, approximately 10.6% and 12.3%.

The general representation of the conversion rate of the PV is observed according to the power map (Fig. 8). It is clear that PV is unable to achieve the designed maximum power capacity, which is affected by several conditions, such as soiling factor and weather condition [47]. Regardless of

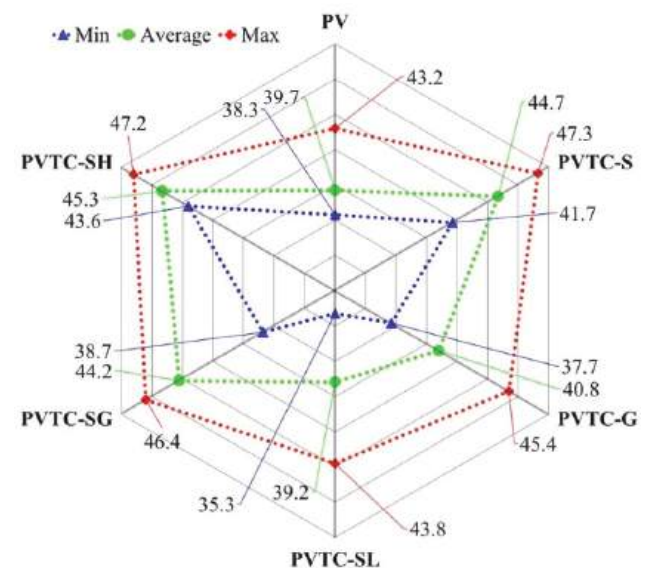


Figure 8. Power curve profile for all tested models.

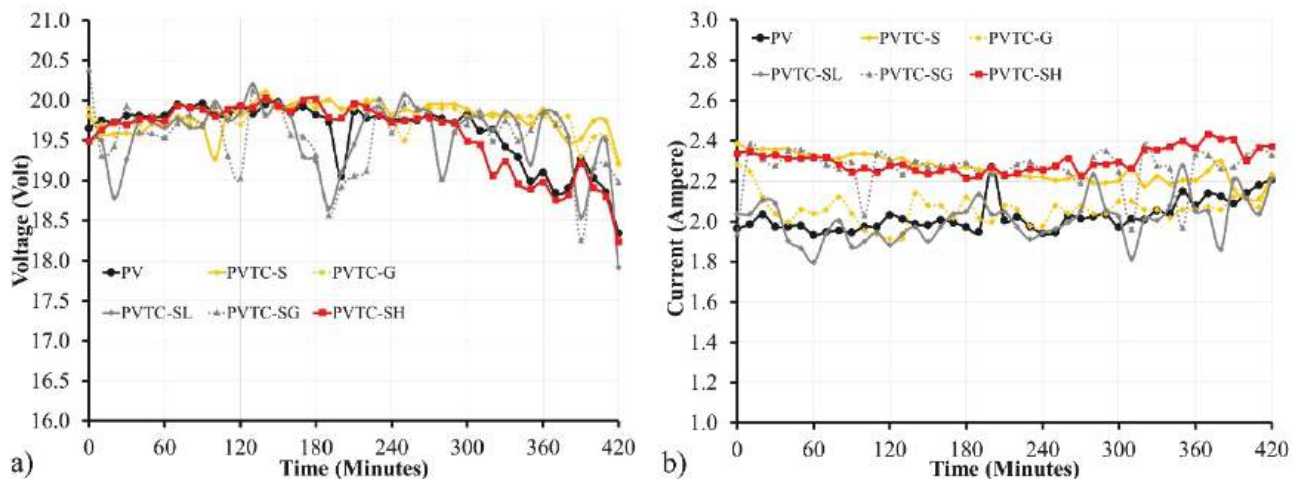


Figure 7. (a) V and (b) I curve for the tested PV and PVTC assembly.

the issue, positive achievement is generally demonstrated by the PVTC system, which consistently indicates a higher peak power. It is the main purpose to maintain the suitable SUT of PV module to ensure the effective power generation of the system [48]. Therefore, the evaluated PVTC in this work shows positive achievement in improving the power output of PV module.

Coolant and Heated Water Temperature

The specific role of employing different PVTC is shown according to the coolant outlet temperature (COT). Figure 9 presents that each model has variative COT profile, implying the specific role of the FM and operating condition based on the designed MF. PVTC-S shows a lower COT compared to PVTC-G. The average COT is 36.9 °C, while it increases about 3 °C for PVTC-G. It shows that PVTC-S has a more suitable heat dissipation behavior than PVTC-G, which supports the higher temperature gradient

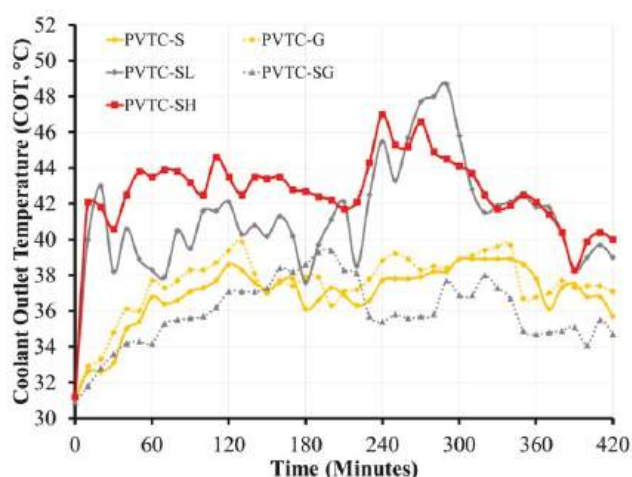


Figure 9. Characteristic of COT for the tested PVTC.

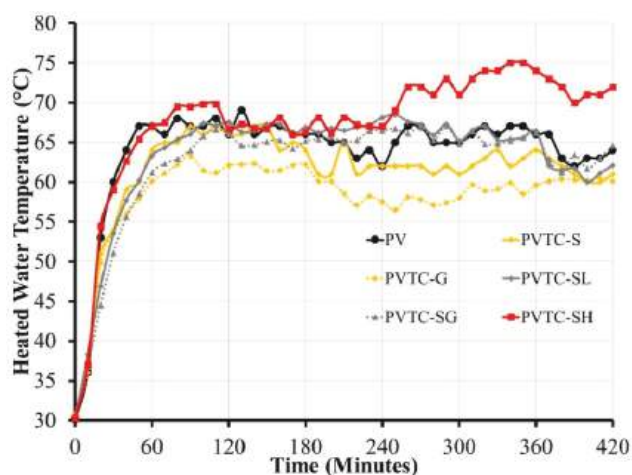


Figure 10. Characteristic of heated water from PV output.

for the PVTC-S (Fig. 6). In contrast, reducing the MF for PVTC-SG demonstrates steady profile, showing certain benefit for utilizing grease/sand mixture as FM for TC. It makes the COT maintain relatively constant value around 35.8 °C.

Utilizing sand at lower MF makes the COT fluctuate, causing severe temperature variation with maximum peak at 48.7 °C. However, the pattern indicates unreliable cooling behavior for the PV modules as this model obtains the lowest power among other PVTC (Fig. 8). Therefore, determination of suitable MF for the coolant is essential to ensure the optimum heat exchange process [49]. It is shown notably for the PVTC-SH using higher MF of the coolant. The COT increases steadily at the beginning before reaching its final peak at midday. In general, PVTC-SH achieves the highest average COT (42.6 °C) than the other model. It confirms the consistent temperature gradient of SUT/BST, as plotted in Figure 6.

The work focuses on the usability of PV output for heating process. The increment in the heated water temperature (Fig. 10) demonstrates the achievement of direct electric conversion for heating the water. It confirms that direct conversion is achievable, showing that the approach is reliable for storing surplus output from PV modules [50] and possibly combining it with renewable thermal system to enhance system efficiency [51]. The profile indicates the power characteristic for each tested model, which varies accordingly due to several factors. Moreover, direct outdoor measurement affects the probability of the variation results and the effective energy generated by PV for direct conversion. It can be combined with the power peak to track the net effective energy for the system (Fig. 8). Despite that, one positive achievement to provide lower temperature cell eventually leads to higher energy intake. It occurs for the PVTC-SH with maximum temperature 72 °C for the heated water.

The two water systems demonstrate the accumulated energy from the TC and PV. It confirms the key aspect of this work, which shows that the integrated PVTC is possible

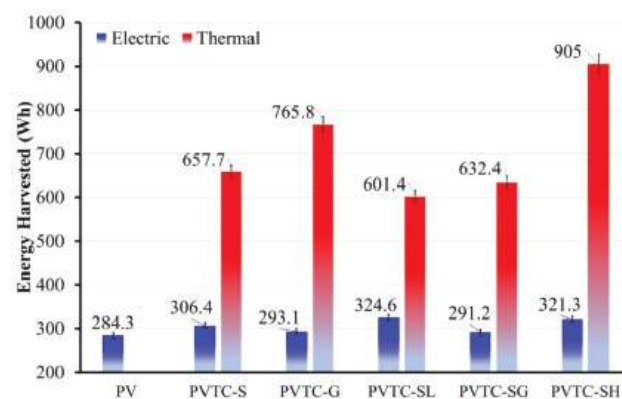


Figure 11. The accumulated electrical and heat energy input for all tested models.

to harvest heat in mutual process. The TC dissipates heat from the modules and stores it partially, while the process reduces the temperature of the PV module, resulting in higher power output. Moreover, the obtained heat from PV module is also applicable for charge operation of PCM [52]. Eventually, the approaches offer flexible storage model for PV system and enlarge the applicability to combine it with electric battery under specific consideration. The collected heat from the coolant adds more input energy, making the two processes produce higher energy density for one assembly PVTC system.

Accumulation of Energy from PVTC System

The PVTC system gains heat and electric energy simultaneously. The accumulated input is presented in Figure 11. The electrical energy from PV modules is lower than its designed maximum capacity, confirming the dynamic condition for on-field measurement, which is significantly affected by the weather conditions. Regardless of the condition, positive tendency is observed for PVTC system, which shows a higher electrical gain than PV. The maximum improvement is 14.2% and 13.1% for PVTC-SL and PVTC-SH. In the case of this condition, using grease as FM is less favorable due to insufficient heat dissipation, making the obtained electrical power relatively low (Fig. 8). It demonstrates suitable cooling behavior PV module promotes a higher power output, which makes the usability of sand as FM considered advantageous based on this finding.

One interesting result is that the system harvests more heat energy than the output electricity from PV module. It is the main challenge for implementing large PV system since the produced waste heat is higher than the generated electric power [53]. From practical perspective, it comes as a serious problem. However, if the system is also intended

for heat generation for water heating, the problem may be taken as additional advantage. According to TC operation, MF influences notably to harvest more heat energy from PV module, as observed for PVTC-SH, PVTC-SG, and PVTC-S. In general, all PVTC positively contribute to collecting the accumulated waste heat from PV module and taken as additional input energy from the assembly.

Figure 12 presents the gained energy from PV module as single component and system (PVTC). The value is obtained from the theoretical PV module output (50 Watts), which operated for 7 hours during the experimental process. PV module only generates electric energy with ratio 80.4%. All PVTC shows a higher value for the generated electricity, confirming the benefit of providing additional cooling mechanism for the PV module. Besides, PVTC also produces additional energy (heat) collected by the coolant. It promotes a positive contribution since the system offers a higher energy input from PV combined with TC, obtaining a higher cumulative energy harvested as a system.

CONCLUSION

The work assesses the PV system operation integrated by active cooling system with thermal collector. The system is operated for specific heating application, accumulating the output from PV and the collected heat from cooling system. The finding reveals that the concept is applicable with two benefits: improving the PV output by reducing the cell temperature and simultaneously harvesting waste heat as input energy for heating system. The cooling system reduces the PV temperature to 40.7 °C and offers maximum coolant output 48.7 °C. The collector is filled with sand and grease, showing significant contribution regarding the cooling ability which corresponds to their physical properties. Also, coolant flowrate plays crucial role in obtaining maximum cooling capacity and temperature output, making additional adjustment can be set according the requirement. PV and thermal collector produce higher cumulative energy around 1.23 kWh, while PV as single component only generates 0.28 kWh. Therefore, this point shows a notable energy output from the combined system.

The system (PV and thermal collector) offers a reliable energy output specifically designed for water heating. It allows the system to reach energy output 350.4% higher than the designed power output of the PV module in this work. However, this preliminary study is designed as an introductory for combined PV-thermal collector water heating. Thus, further work is required to assess in detail about reliability of the system for long term operation, including the possibility of introducing different filler material. Another crucial aspect is net energy balance for operating the pump and detailed techno-economy consideration. The finding from this work is suitable for taking as relevant baseline for further development of the PV-thermal collector for heating purpose, hoping for more advanced technology to maximize solar energy as a clean source for residential heating.

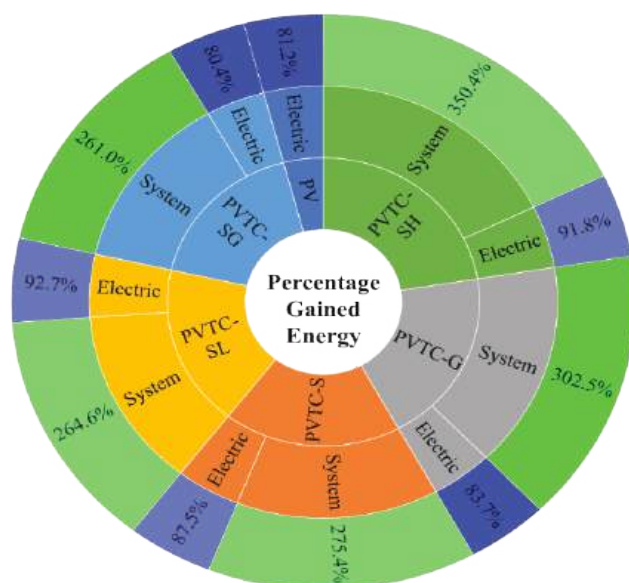


Figure 12. The Percentage of gained energy for all models.

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.

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Research Article

Comparative performance of different solar still configurations

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ABSTRACT

This study evaluates the performance of three solar still configurations: the conventional solar still, the pyramid solar still with fins, and the pyramid solar still with fins covered in black cotton cloth. The variation in absorber plate, glass plate, water temperature, and the quantum of water collected in the attempted configurations was determined at identical conditions. Of the three solar still configurations, the pyramid shaped solar still having fins covered with a black cloth exhibited better efficiency and produced 3.7 liters of potable water. The pyramid shaped solar still having fins closely follows with a water production of three liters. The superior performance is due to the better heat absorption and heat transfer following the enhanced area and the capillary effect of the black cloth. In addition, a deep neural network model with better accuracy, having an R^2 value of 0.96, was created in the Python environment to predict the glass, absorber, and water temperatures along with the water collected. The novel integration of experimental and artificial intelligence in the optimization of solar still provides room for potential seawater-to-potable-water conversion and is suitable for remote or hilly areas.

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INTRODUCTION

Researchers around the world are increasing their focus on developing sustainable and renewable energy, and thereby transforming the world's energy landscape. Of the various renewable energy solutions, solar energy has gained significant attention due to its capability to address the growing energy demand along with mitigating the environmental impact [1]. Though solar energy is employed for numerous applications, solar desalination plays a vital

role, as it provides a viable solution for addressing the acute water shortages in arid regions having limited or no freshwater options [2]. Solar stills, a subset of solar desalination, demonstrated significant success in harnessing solar energy for water purification [3].

Basin type solar stills evaporate the saline water by harnessing the solar energy, followed by the condensation of vapor to yield freshwater [4]. Earlier researchers attempted different modifications, materials, and varied process

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parameters to augment the performance of the solar stills. Ensafisoroor et al. introduced sponges on the basin solar still and successfully increased the water production [5]. Likewise, Hansen et al. used flat, grooved, and fin-shaped absorber plates in an inclined solar still to recover waste heat, resulting in higher water productivity [6]. Goshayeshi and Safaei studied the effect of varying the glass cover's inclination angle (25° , 27.5° , 30° , 32.5° , and 35°) in stepped solar stills and recommended an optimal angle of 32.5° for improved performance [7]. In a related effort, Omara et al. developed a pyramid-shaped solar still with convex cylinders and nanocomposite-based dish absorber plates to achieve greater efficiency [8]. Ahamed et al. attempted phase change material encapsulated fins and thereby successfully enhanced the thermal performance of the basin solar still [9]. Meanwhile, Kabeel et al. employed v-corrugated absorber plates in a tubular solar still and reported improved performance. In addition, they also developed a mathematical model to replicate the operation of the solar still as well [10]. On the other hand, Hameed et al. performed an experimental study on a single slope solar still and correlated the results with the numerical simulation performed using COMSOL. According to their results, both numerical and experimental outcomes are in good agreement with each other [11]. In 2021, Jobrane et al. summarized the research carried out by other researchers on wick-type solar stills, involving variations in geometrical designs, wick arrangements, and integrated heating systems [12].

Few researchers introduced machine learning on predicting the performance of solar stills. Wang et al. successfully applied the Bayesian optimization algorithm to predict the performance of tubular solar stills [13]. Moustafa et al. utilized an artificial neural network (ANN) to forecast the thermal efficiency and water production of solar stills [14]. Similarly, Bahiraei et al. used a hybrid ANFIS model and a PSO-enhanced neural network to predict energy efficiency with improved accuracy [15].

It is inferred that the published literature focused on varying the absorbing surface, condensation enhancement techniques, and geometric modifications to augment the overall performance of a solar still. Likewise, different types of absorber plates and transparent covers are attempted to improve the heat absorption and to curtail the heat losses [16]. In addition, the effect of climatic conditions such as solar radiation [17, 18], ambient temperature [19], and wind speed [20] on the performance of basin solar stills is reported. The present study proposes the integration of fins and black cotton cloth in a pyramid solar still to attain an improved heat transfer and absorption, leading to higher water yield. The use of the machine learning technique (DNN) for predicting the performance of the solar still provides scope for future improvements in system design and optimization with minimum experimentation.

MATERIALS AND METHODS

In this study, three different types of solar stills viz., a conventional solar stills (CSS), a pyramid solar still with fins (PSSF), and a pyramid solar still with fins covered with black cloth (PSSFB), were fabricated using 18-gauge alloy sheets, having a uniform basin area of 1 m^2 . These were designed and uniformly tested at the Department of Mechanical Engineering, Annamalai University, Tamilnadu, India. The dimensions of the stills were 110 cm in length and 110 cm in breadth.

The CSS serves as the reference, whose basin is coated black, covered with a transparent 4 mm thick glass, having transitivity of 88%, and inclined at 11° (based on the location) to maximize the solar exposure. Water is filled to a depth of 1 cm for effective operation. The edges of the transparent glass cover are fixed with a rubber gasket to curtail the losses. The pyramid solar still (Fig. 1: PSSF) retains the basic components of CSS apart from having the pyramid shaped glass cover, fabricated using four glasses having an equal slope. The covers were attached and sealed



Figure 1. Pyramid shaped solar still with fins attached.



Figure 2. Pyramid shaped solar still with fins covered by black cloth.

using silicone sealant. The metallic fins in the pyramid shaped absorber plate are also coated with black paint and positioned at 90°. In the third setup (PSSFB), a layer of black cloth was employed to cover the fins (Fig. 2).

The condensate was collected in a trough located at the base of the glass cover. Thermocouples were positioned at various locations of the solar stills to measure the temperatures of the absorber and glass plates and water. Temperature data were logged using a twelve-channel temperature indicator. A digital thermometer was used to measure the ambient temperature, while the Solar HT kit with a PV204 Solarimeter with an accuracy of $\pm 1 \text{ W/m}^2$ was utilized to record the solar intensity. The distilled water was collected in a transparent beaker. Experiments were conducted between 09:00 a.m. and 4:00 p.m. over 25 different days from March to May 2024. The variation in glass, absorber, water, and ambient temperatures; solar radiation; and water yield were recorded at an equal interval of 30 minutes.

Subsequent to the experimentation, the obtained data were utilized to develop an artificial intelligence model

employing a deep neural network, which is described below. Of the available data, 70% was used for training the model, and the remaining 30% was equally used for validation and testing on a Windows computer.

Deep Neural Network

A deep neural network is a complex model based on machine learning techniques that comprises several layers interconnected by neurons (Fig. 3). It is employed to learn complicated patterns within large datasets automatically [21]. There are several neurons within each layer of a DNN that transform the input data and refine features in order to make an accurate prediction. Moreover, the model is equipped with advanced optimization techniques and a host of activation functions, hence very effective in predictive analytics [22]. The depth and architecture are critical in DNN, as this generalizes from training data onto unseen examples and thus drives actual performance in deployment.

The accuracy of predicting the responses (temperatures and water collection), with respect to training and testing data, is determined with the aid of three performance

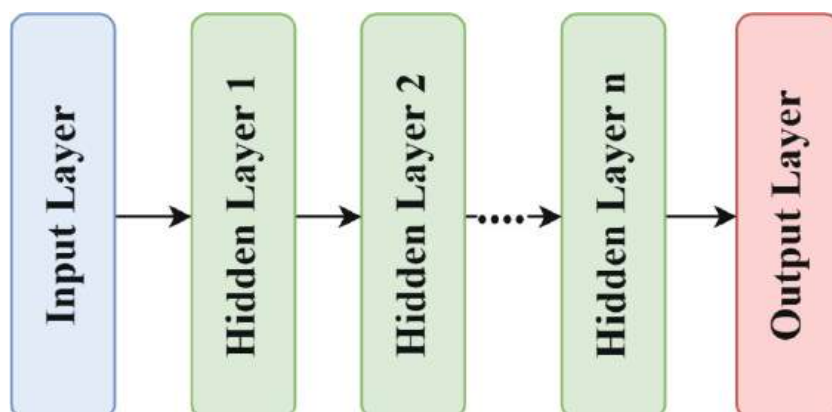


Figure 3. DNN model structure.

metrics: a) Mean Absolute Error (MAE), b) the coefficient of determination (R^2), and c) Mean Absolute Percentage Error (MAPE). MAE represents the average magnitude of errors between predicted and actual values. R^2 indicates the proportion of variance in the actual data explained by the model. MAPE expresses the error as a percentage by averaging the absolute percentage differences between predicted and actual values. These metrics quantify the deviation between predicted and experimental results. The attempted performance measures are calculated by: [23].

$$MAE = \frac{\sum_{j=1}^n (Y_k - y_k)}{n} \quad (1)$$

$$MAPE = \frac{\sum_{j=1}^n (Y_k - y_k)}{Y_K} \times 100 \quad (2)$$

$$R^2 = 1 - \frac{\sum_{k=1}^n (Y_k - y_k)^2}{\sum (Y_k - Y_{k,mean})^2} \quad (3)$$

RESULTS AND DISCUSSION

Variation in Solar Intensity Over Time

The solar intensity (power received from the sun per unit area) is a critical factor influencing the evaporation and condensation process and thereby dictating the overall efficiency and performance of solar stills. Figure 4 illustrates the fluctuations in solar radiation over the daytime. Further, it is observed that a uniform pattern in solar intensity is observed. The solar intensity varies throughout the day due to the rotation of the earth and the position of the sun in the sky [24].

In the early morning hours, the sunlight reaches the earth at a shallow angle, and hence the intensity of solar radiation is lower. Consequently, the solar still absorbs minimal heat,

and the evaporation of water is lesser. As the day progresses, the sun rises and reaches higher angle of incidence, leading to a higher concentrated solar intensity. The solar intensity increases, reaching a maximum (829.6 W/m^2) around noon. The occurrence of maximum solar intensity around noon is similar to the studies of Osigbeme et al [25]. During this period the efficiency of the solar still is higher following the availability of maximum thermal energy for evaporation.

Post noon, the sun starts to descend, and hence the solar intensity declines. The decline in solar intensity gradually decreases the rate of evaporation in the afternoon, and hence the performance of the solar still reduces significantly. By late afternoon and into the evening, the solar intensity greatly reduces, and consequently, the evaporation rate. The fluctuation in solar intensity emphasizes the significance of modifications in the solar still design to harness solar energy effectively.

Variation in Temperatures

The effect of design modifications on the average absorber plate, glass plate, and water temperatures obtained across the three solar still configurations are illustrated in Figs. 5-7. The temperatures in all three solar still configurations rise from 10:00 hours and reach the maximum between 12:00 and 13:00 hours, and then a gradual decrease is witnessed in the afternoon. Higher temperatures during the midday impart more thermal energy to water, accelerating the evaporation of water [26].

In the conventional solar still (CSS), the maximum temperature of the absorber plate reaches around 68 °C when the solar radiation is peaking (Fig. 5). The maximum temperature supports the evaporation process, as the available heat is absorbed by the plate and the same is transferred to the water medium. Subsequently, the water temperature (Fig. 6) in the CSS lags behind the absorber plate temperature (66 °C). This phenomenon is consistent with the studies of Peng and Sharshir [27]. The glass plate, otherwise termed the condensation surface, maintains a lower

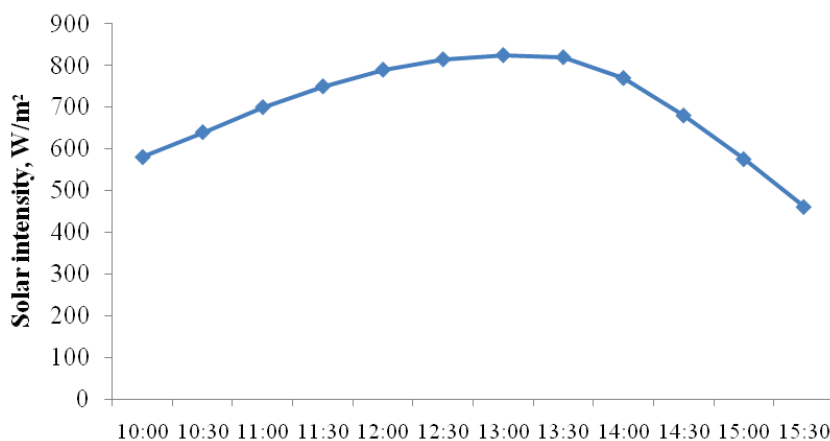


Figure 4. Variation in solar intensity over time.

temperature (Fig. 7), around 55–57 °C, due to its exposure to the ambient air and the evaporative cooling effect [28]. The temperature gradient between the water and the glass plate is crucial for maximizing the condensation rate, thus improving the water production.

However, the introduction of fins in a pyramid shaped solar still (PSSF) improves the overall heat transfer area, promoting better heat distribution to result in higher thermal performances than the CSS. The fins, attached to the absorber plate, facilitate more efficient heat distribution throughout the water [29]. In addition, the pyramid shape promotes concentration of solar radiation to amplify the absorber plate temperature. This modification raises the absorber plate temperature to 73–77 °C (Fig. 5). Consequently, the water temperature also rises, reaching 74 °C (Fig. 6), as the fins allow more heat to be absorbed and transferred. The glass plate temperature in this design reaches 60 °C (Fig. 7), but the increased evaporation rate leads to a higher overall productivity (detailed in the next section).

The third solar still configuration, pyramid shaped solar still with fins covered by black cloth, displays the maximum

thermal performance. The usage of black cloth increases the absorptivity, minimizes the reflective losses, and maximizes the heat retention [30]. As a result, the absorber plate reaches a temperature as high as 75–78 °C (Fig. 5), higher than the other two attempted solar still configurations. Consequently, the water temperature increases to 75 °C (Fig. 6). The difference in temperature between the water and the glass plate, reaching nearly 64 °C (Fig. 7), significantly improves the evaporation and condensation rates. The black cloth-covered fins make sure that the heat is distributed uniformly and retained for a longer duration, leading to a higher temperature in the water. Singh et al [31] opined that enhancing the surface area enhances the absorber plate temperature, consistent with the present study. It is concluded that the variations in the absorber plate, water, and glass plate temperatures among the three configurations illustrate the profound influence of design modifications on solar still performance. Though the conventional pyramid shaped still is effective, the addition of fins and the use of black cloth over the fins provide significant improvements in attaining higher temperatures, leading to increased potable water production (discussed below).

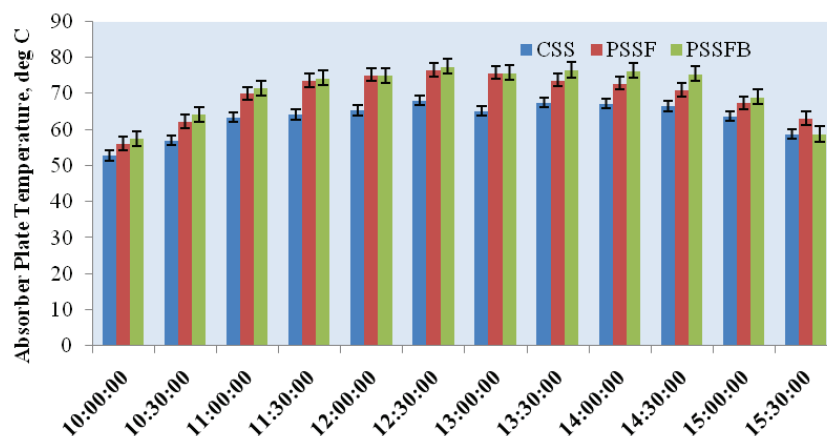


Figure 5. Variation in absorber plate temperature over time.

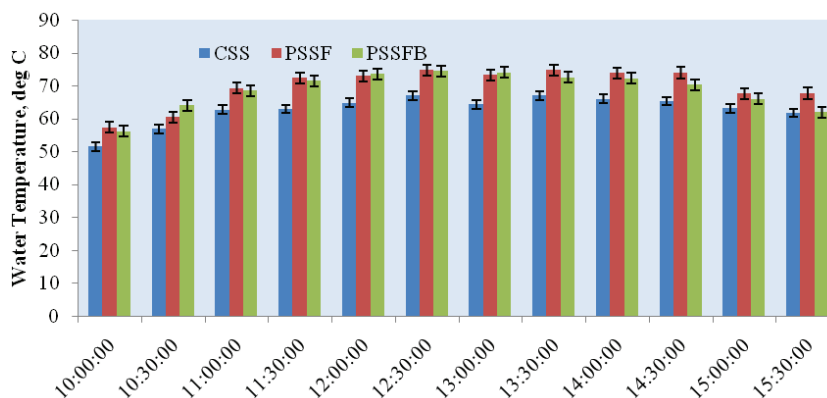


Figure 6. Variation in water temperature over time.

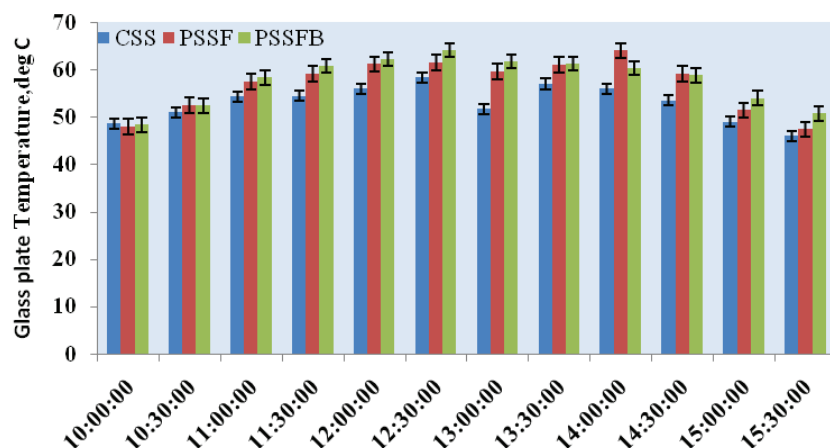


Figure 7. Variation in glass plate temperature over time.

Water Collected

The conventional solar still (CSS) traps the solar energy through a transparent cover, which heats the water prevailing inside. Subsequently, the water evaporates and condenses on the cooler surface of the cover and is collected as distilled water. The CSS produces around 2.3 liters of water per day under optimal conditions (Fig. 8). However, the pyramid shaped solar still with fins (PSSF) yields 3 liters of water, 30% more than the CSS. The increased water yield is due to the higher surface area exposure to sunlight created by the geometrical modification i.e. the pyramid shaped absorber plate, as reported by Hammoodi et al [32]. The surface area and water evaporation rates of the solar still are further increased by the attachment of fins. The fins increase the temperature of the water and the inner surfaces of the solar still, yielding more water. The presence of fins aids in even distribution of heat across the solar still, suppressing the temperature gradients within the still. The fins guide the water towards the collection trough as well.

The maximum water collection (3.7 liters), which is 23 % more than the PSSF is obtained for the pyramid-shaped solar still with fins covered with black cloth (PSSFB) configuration. The 23% increase is attributed to the presence of black cloth which harnesses more solar radiation to obtain superior evaporation and collection efficiency. In addition, the capillary effect of drawing water upwards and spreading it evenly over a larger surface area supports the phenomenon. As the water vapor rises, it condenses on the transparent glasses of the still, resulting in a higher water production. The dual effect of improved thermal absorption and efficient water distribution makes the PSSFB an effective modification for achieving higher thermal performance and more water collection.

Energy Balance

In a solar still, energy balance accounts for heat transfer between the absorber, glass, and water through radiation, convection, and conduction processes. These mechanisms

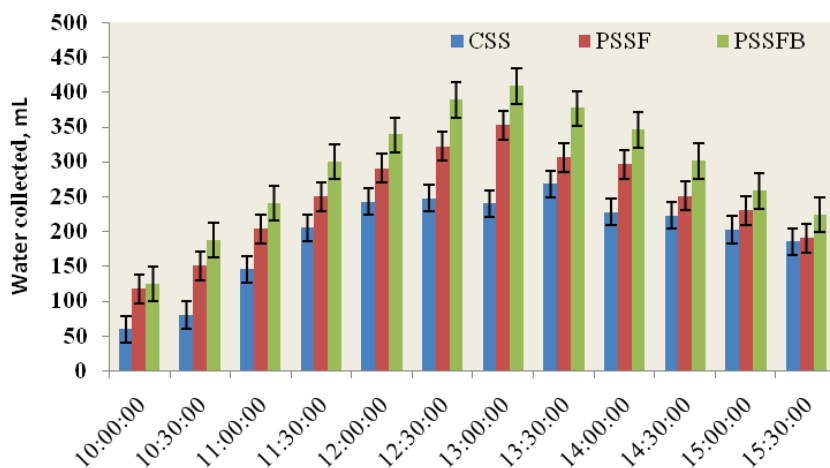


Figure 8. Water collected during day time.

regulate the thermal dynamics essential for efficient water evaporation and condensation. The convective heat transfer between water and glass cover [33]

$$q_{c(w-g)} = h_{c(w-g)} A_w (t_w - t_g) \quad (4)$$

$$h_{c(w-g)} = 0.884[(t_w - t_g) + \{(p_w - p_g)t_w / (268.90 \times 10^3 - p_w)\}]^{1/3} \quad (5)$$

$$p_w = e^{\{25.317 - [5144 / (t_w + 273)]\}} \quad (6)$$

$$p_g = e^{\{25.317 - [5144 / (t_g + 273)]\}} \quad (7)$$

Similarly, heat transfer by radiation between water and glass cover [33]

$$q_{r(w-g)} = h_{r(w-g)} A_w (t_w - t_g) \quad (8)$$

$$h_{r(w-g)} = \varepsilon_{effective} \sigma (t_w^2 + t_g^2) (t_w + t_g) \quad (9)$$

$$\varepsilon_{effective} = 1 / [(1/\varepsilon_w) + (1/\varepsilon_g) - 1] \quad (10)$$

The heat transfer between water and glass cover by evaporation [33]

$$q_{e(w-g)} = h_{e(w-g)} A_w (t_w - t_g) \quad (11)$$

$$h_{e(w-g)} = 16.237 \times 10^{-3} h_{c(w-g)} [(p_w - p_g) / (t_w - t_g)] \quad (12)$$

Radiation heat transfer between glass cover and atmosphere [33]

$$q_{r(g-a)} = h_{r(g-a)} A_g (t_g - t_a) \quad (13)$$

$$h_{r(g-a)} = \varepsilon_g \sigma [t_g^4 - t_{sky}^4] / (t_g - t_a) \quad (14)$$

$$t_{sky} = t_a - 6 \quad (15)$$

Convective heat transfer from glass cover to atmosphere [33]

$$q_{c(g-a)} = h_{c(g-a)} A_g (t_g - t_a) \quad (16)$$

$$h_{c(g-a)} = 5.7 + 3.8 V_a \quad (17)$$

Convective mode of heat transfer between basin and water [33]

$$q_{c(b-w)} = h_{c(b-w)} A_w (t_b - t_w) \quad (18)$$

The yield of the solar still or the amount of water condensed on the inner surface of the glass

$$m_{e(w-g)} = [q_{e(w-g)} / h_{fg}] \times 3600 \quad (19)$$

Uncertainty Analysis

The accuracy and precision of measurements are influenced by various factors, including the selection of instruments, calibration, testing conditions, observations, environmental factors, readings, and test design [34]. To ensure the tests are properly conducted, an uncertainty analysis is performed. Table 1 lists the instruments used in this study, along with their model, accuracy, and operating range. The smallest potential error for each instrument is calculated by dividing the smallest measurable reading, the minor meter reading, by the lowest possible output value. This calculation yields the instrument's minimum possible error.

The uncertainties of the measuring instruments employed in the study are determined using the equation provided in Ref. [34].

Based on the uncertainty analysis, the total uncertainty of the measuring instruments in this study is found to be within $\pm 1.582\%$.

PREDICTION BY DNN MODEL

The DNN model was trained with the temperatures and water yield data collected during the experimental days. By varying the number of hidden layers and the quantity of neurons in the hidden layers, numerous models were constructed. The range of hyperparameters attempted and the optimal neurons in each hidden layer are shown in Table 2.

Table 1. Instruments employed along with accuracy and range

S. No	Instrument	Model	Accuracy	Range
1.	Anemometer	amiciSense GM816	$\pm 5\%$	0-30 ms
2.	Thermocouple	K-type	$\pm 1\%$	0-100°C
3.	Measuring beaker	1L	± 10 ml	0-1.000mL
4.	Solarimeter	PV204	± 1 Wm ²	0-5000 Wm ²

Table 2. Hyperparameters tuned

Parameters	Range	Optimal value
No. of dense layers	1 to 6	3
No. of units in layer 1	1 to 1000	835
No. of units in layer 2	1 to 1000	635
No. of units in layer 3	1 to 1000	769

Table 3. Optimizer parameters range

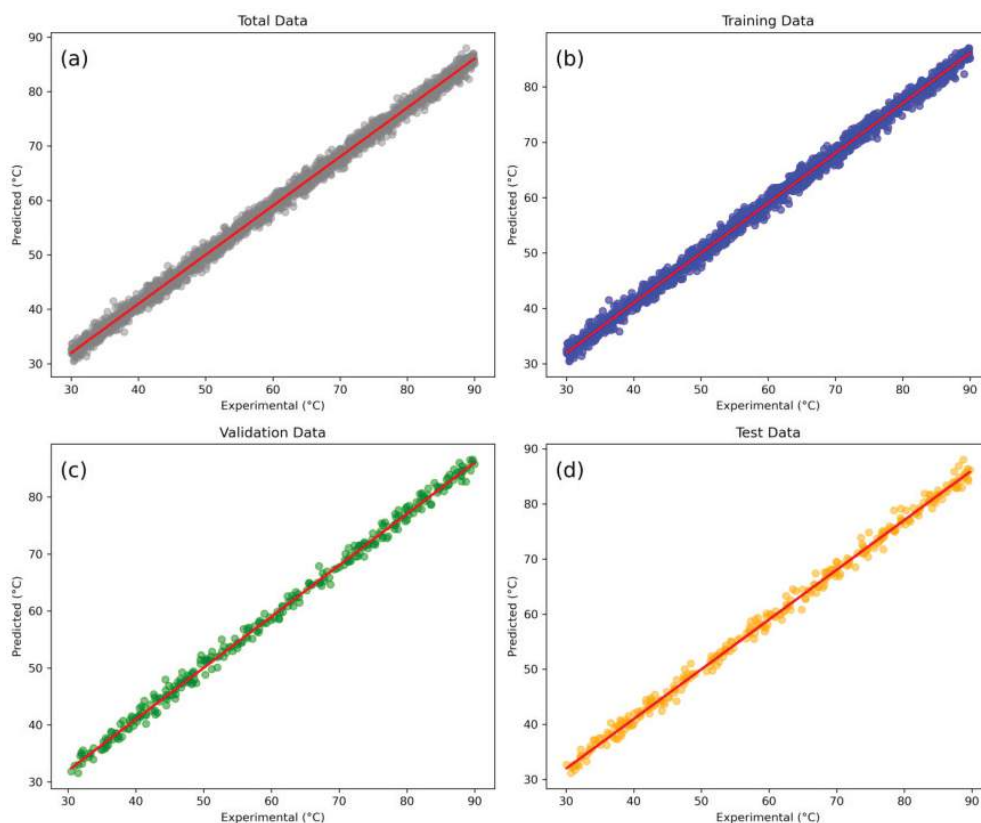
Parameters	Range
Learning rate	1e-3 to 1e-1
Decay	1e-3 to 1e-1

Table 4. Performance metrics

R-Square	MAE	MAPE	MSE
0.9601	1.1986	0.8978	0.0796

The optimum neurons were determined by employing the Adam optimizer, as recommended by Saravanan et al [35]. The learning rates and decays of the Adam optimizer are presented in Table 3.

A scatter plot comparing the actual and predicted temperatures, as well as the amount of water collected, provides valuable insights into the model's accuracy (Figs. 9-12). In the scatter plot for both temperatures and water collected, a larger proportion of experimental data scatter around the diagonal line. The prevalence of data points concentrated in the closer proximity of the diagonal line indicates the existence of a strong correlation between DNN prediction and experimental outcomes. The strong correlation of predicted values by the DNN model indicates higher accuracy [36]. In this model, only a small number of experimental conditions deviate from the median, resulting in a high R^2 value of 0.96, demonstrating a strong goodness of fit. Table 4 presents the performance metrics for the DNN model's accuracy. With a Mean Absolute Error (MAE) of 1.1986, a Mean Absolute Percentage Error (MAPE) of 0.8978, a Mean Square Error (MSE) of 0.0796, and a high R^2 value of 0.9601, the DNN model shows a strong alignment with the experimental results. Therefore, the deep neural network model is recommended for predicting temperatures and water collection in a solar still.

**Figure 9.** Comparison of absorber plate temperatures (experimental vs predicted).

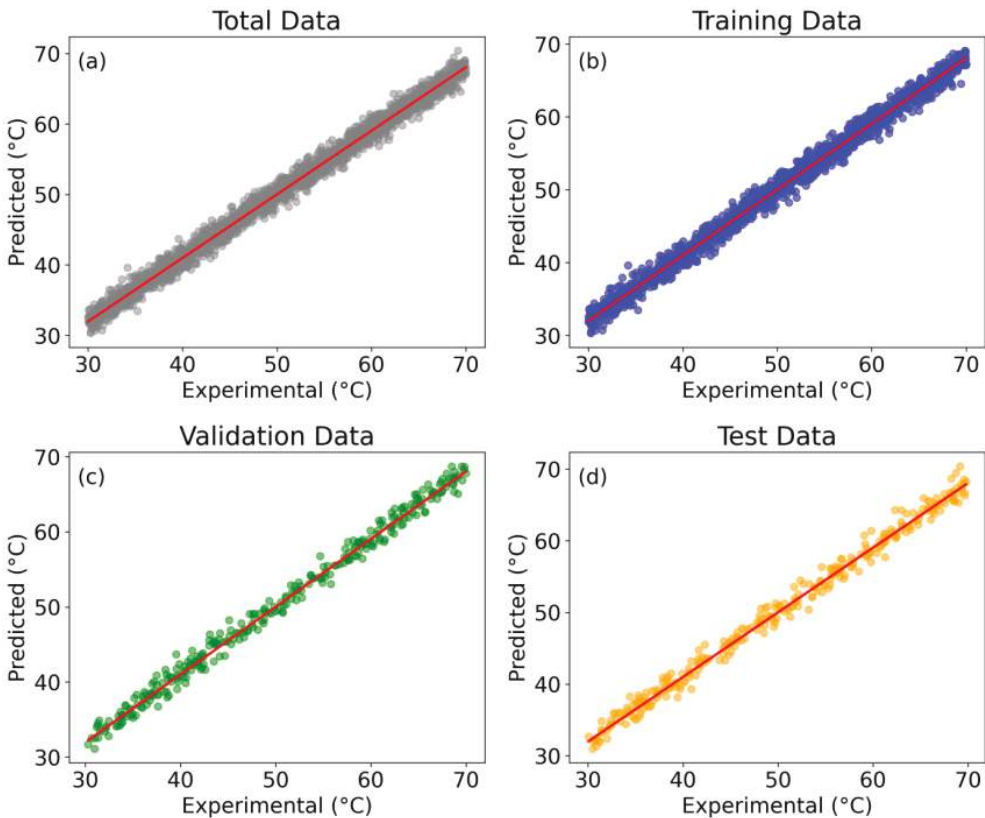


Figure 10. Comparison of glass plate and water temperatures (experimental vs predicted).

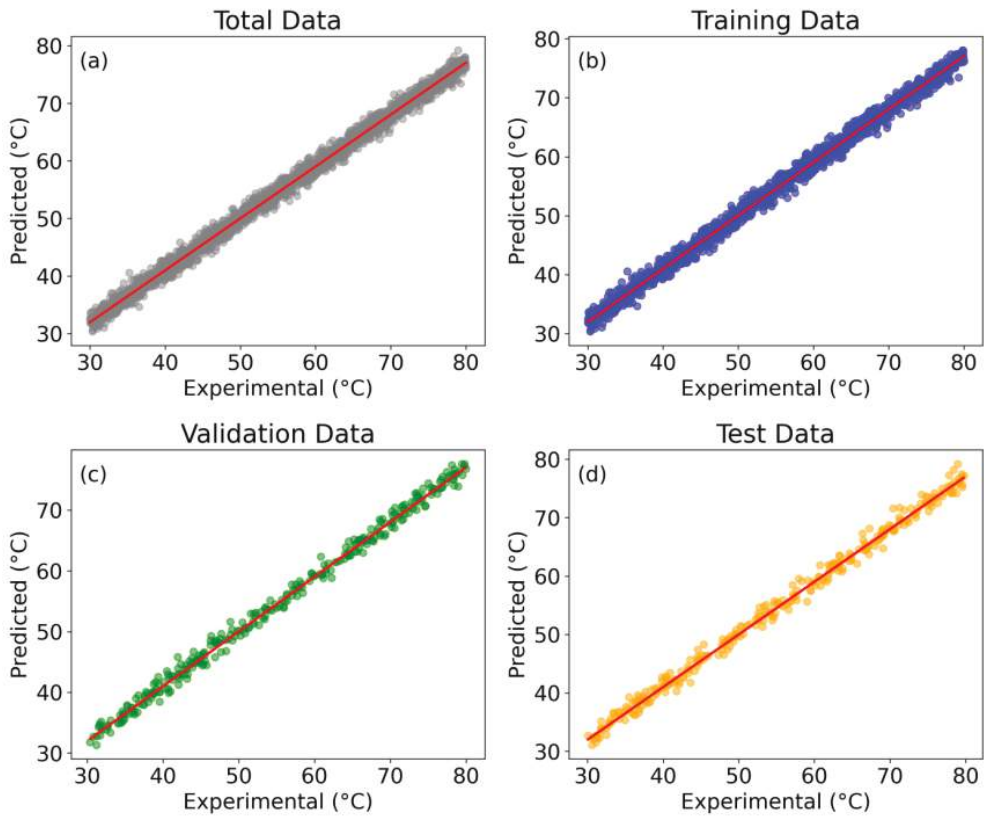


Figure 11. Comparison of water temperatures (experimental vs predicted).

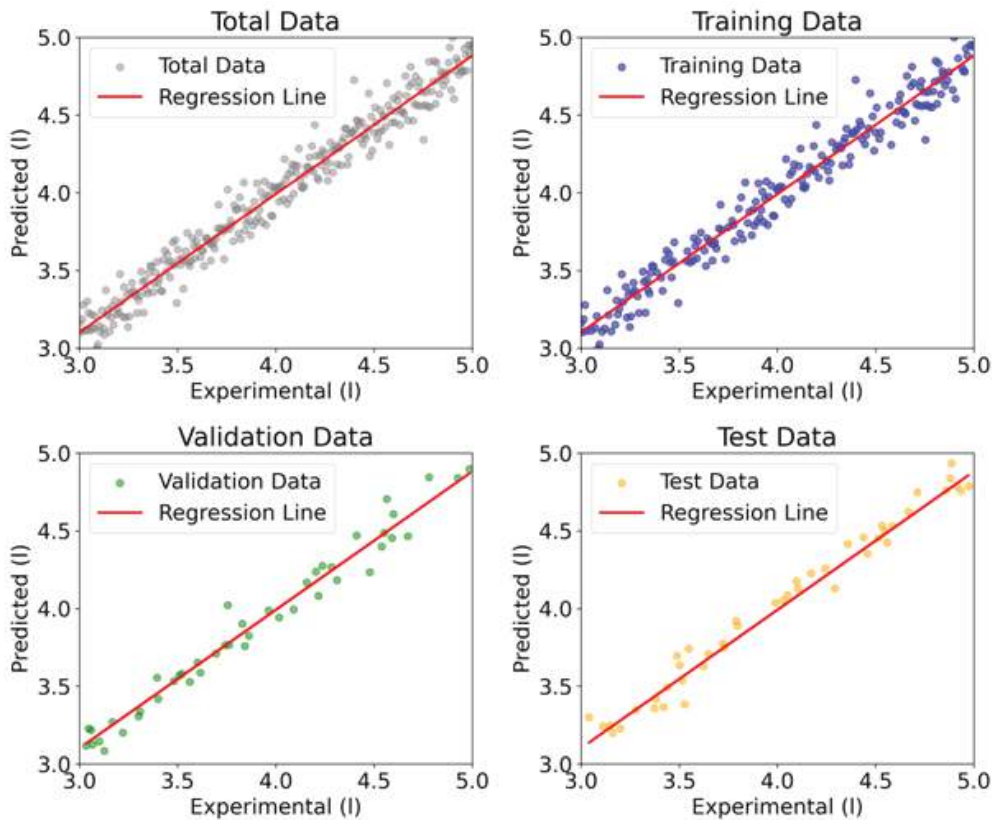


Figure 12. Comparison of water collected experimental vs predicted.

CONCLUSION

The salient conclusions from this novel experimental study on the fabrication and testing of three solar still configurations are as follows:

1. The surface modifications in the pyramid shaped absorber plate with fins improve the absorber plate, water, and glass temperatures.
2. The employment of fins augments the heat transfer area, and covering the fins by a black cloth cover enhances heat retention and improves the evaporation rate.
3. The combined effect of the pyramid shape, fins, and use of black cloth cover results in the highest water collection among the three attempted configurations, highlighting their capability to effectively address acute water shortages.
4. The water collected in the solar still having both fins and the black cloth model is nearly twice that of the conventional solar still. The pyramid shaped solar still with fins attains 33% more efficiency than the conventional solar still.
5. The Deep Neural Network model is capable of predicting the temperature and water collected in a solar still with an accuracy of 96%.

NOMENCLATURE

A_g	Area of glass cover
A_w	Absorber Plate area
CSS	Conventional Solar Still
DNN	Deep Neural Network
$h_{c(b-w)}$	Heat transfer coefficient due to convection (basin-water)
$h_{c(g-a)}$	Convective heat transfer coefficient (glass cover-atmosphere)
$h_{c(w-g)}$	Convective heat transfer coefficient (water-glass cover)
$h_{e(w-g)}$	Evaporative heat transfer coefficient (water-glass cover)
h_{fg}	Latent heat of vaporization
$h_{r(g-a)}$	Radiation heat transfer coefficient (glass cover and atmosphere)
$h_{r(w-g)}$	Radiation heat transfer coefficient (glass cover and water)
MAE	Mean Absolute Error
MAPE	Mean Absolute Percentage Error
$m_{e(w-g)}$	Water vapour condensation rate
p_g	Partial pressures of vapour at the glass surface
PSSF	Pyramid Solar Still with Fins
PSSFB	Pyramid Solar Still with Fins and Black Cloth
p_w	Partial pressures of vapour at the water surface

$q_{c(b-w)}$	Convective mode of heat transfer (basin-water)
$q_{c(g-a)}$	Convective heat transfer (glass cover-atmosphere)
$q_{c(w-g)}$	Convective heat transfer (water-glass cover)
$q_{e(w-g)}$	Evaporative heat transfer (water-glass cover)
$q_{r(g-a)}$	Heat transfer by radiation (glass cover-atmosphere)
$q_{r(w-g)}$	Heat transfer by radiation (water and glass cover)
R^2	Coefficient of determination
t_a	Ambient temperature
t_g	Temperature of glass cover
t_{sky}	Sky temperature
t_w	Temperature of water
Y_k	Actual value
y_k	Predicted value
Y_{k_mean}	Mean actual value
$\epsilon_{effective}$	Effective emittance (glass cover-water surface)
ϵ_g	Upper glass emissivity
ϵ_w	Lower glass emissivity
σ	Stefan–Boltzmann constant

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.

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Research Article

Prediction of dual-axis solar tracking of the PV system for optimal power output using GA, DE, TLBO, Jaya, and Rao-3 algorithms

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ABSTRACT

In this study, novel algorithms approach that forecast the sun's movement more accurately, even in bad weather, and optimize for additional characteristics besides solar tracking. It was investigated through comparative analysis by using standard optimization methods, i.e., the JAYA algorithm, Teaching Learning Based Optimization algorithm, Rao-3 algorithm, Genetic algorithm, and Differential Evolution algorithm for a dual-axis solar tracking of a Photovoltaic system. This dual-axis tracking involves rotation of two-axis, i.e., tilt angles and azimuth angles, to maximize the total available solar radiation at optimized tilt angles and azimuth angles for varying time from sunrise to sunset. The reason for obtaining optimum angles is to align the solar PV panel with the incoming solar radiation to get the maximum output. A non-linear and constrained optimization problem is employed to evaluate the optimum tilt angle and optimum azimuth angle trajectories. As in this case, the objective function is in an explicit form that is not known, the optimization methods are employed to evaluate the objective function, and that method calculates the available total solar radiation tracked and carried out in MATLAB software. The optimization results for the considered algorithms showed that the output electrical energy with the dual-axis tracker is far greater than that of the fixed system. It was found that for summer days, the electrical power generation by different optimization algorithms are as follows: TLBO-16.1713 kWh, JAYA-16.1436 kWh, Rao-3-16.1125 kWh, DE-16.2079 kWh, and GA-16.1969 kWh, and measured fixed system-13.2610 kWh. The obtained results for various optimization algorithms, in comparison to a measured fixed system, demonstrated significant enhancements in power generation.

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Statement of novelty and industrial relevance

Novel algorithms approach that forecast the sun's movement more accurately, even in bad weather, and optimize for additional characteristics besides solar tracking. Dual-axis

trackers that optimize efficiency and cost by combining fixed and single-axis features.

Enhanced Energy Generation: Dual-axis trackers boost solar system energy capture by 30-45% over fixed tilt

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systems and 10-20% over single-axis trackers. They maximise solar capture by keeping the panels perpendicular to the sun all day and year.

Land Efficiency: Higher panel energy output allows for more electricity generation on less land. This helps in locations with high land costs or limited space.

Adaptability: They can be employed in regions where the sun's yearly path presents significant challenges in terms of capturing maximum energy.

Optimal results in Diverse Latitudes: Dual-axis trackers perform best in higher or lower latitudes when the sun angle fluctuates dramatically throughout the year.

Economic Value: Despite the greater upfront expenditures, the enhanced energy yield might result in faster returns on investment. The initial outlay may be higher, but the additional energy production should eventually pay for itself through economies of scale and technical developments.

INTRODUCTION

Renewable energy sources are gradually replacing fossil fuels and are the need of the hour. These are mainly derived from natural resources, either abundant in nature or constantly replenished. Renewable energy resources come from natural resources like wind, sun, waves, tides, and geothermal heat and are clean and pollution-free. Among the various renewable energy resources, solar energy is the most pristine and abundant in countries like India.

A PV cell is an electronic gadget that directly converts the sun's radiation into electrical power. A PV system's performance depends on the solar cells' temperature, which determines how efficiently they operate [1]. There are three selected parameters that determine the solar cell outputs. The photovoltaics selected parameters such as short circuit current (I_{sc}), open circuit voltage (V_{oc}), and maximum power (P_{max}) are presented in [2,3]. The PV panel's efficiency decreases with the increased operating temperature of the module. According to Koundinya et al. (2017), the finned heat pipe can overcome the operating temperature of PV modules. It was also found that the experimental and computation study results are in good agreement [4]. Significant disadvantages of solar PV systems include high costs compared to fossil fuels, poor efficiency, and intermittent operation. In order to overcome these challenges, capturing more and more energy from the sun with the existing PV modules is a challenging task for researchers and the scientific community across the globe. PV materials, geographical locations, environment temperature, weather conditions, angle of incidence, and module direction influence the PV system's energy output. Tracking techniques decide the best orientation of the PV modules towards the sun. The intelligent tracking technique is the most promising among the several existing categories of tracking systems. It is because of its capacity to forecast the correct path of the sun by using predefined algorithms, as mentioned in

[5,6]. The variation in the solar intensity on the PV system exists because of daily and seasonal movements of the earth across the sun. The best possible orientation of solar panels to the trajectories of the sun can be achieved by considering the rotation as well as the revolution of the earth. With the optional use of solar trackers, collected solar energy can be increased by 10 to 100%, depending upon geographical locations and changing weather conditions. The only issue is that this tracking device consumes power equivalent to 2-3% of the increased power output [7]. An experimental study on the solar tracker using a sun tracking system to maximize the productivity of solar still was conducted by Abdallah et al. (2008). The solar tracking system under consideration was computerized and was used to orient the solar system, still having an effect of 1m^2 with the trajectory of the sun's movement throughout the day. The performance of the solar still with the single-axis tracker increased productivity by around 22%, which was greater than the static system under consideration [8]. An experimental study of dual axis solar tracker using a predefined Differential Evolution algorithm for tracking the sun's path throughout the day was done by Seme and Štumberger (2011). It was found that the maximum solar energy of the photovoltaic system was obtained by maintaining the optimum tilt and azimuth angles evaluated by the optimization technique. The results reported in this research showed that by using novel technologies, the efficiency of energy generation inside a PV system could be increased up to 10-50% [9,10]. In another investigation in a fixed system, the energy generation predicted through GA increased by 15.85% for PV modules [11]. The new method for optimizing solar trackers by applying arbitrary design and geometry of Concentrating Photovoltaic (CPV) and PV tracker system that moves less and requires less mechanical effort is now being used to track the sun's location. The optical system used in the solar tracking concentrator quickly reacts to variations in the sun's location throughout the day, month, and year. The goal of a solar tracking system is to hold onto solar light alignment with the surface of the receiver [12–14]. A fixed system and a continuous dual-axis sun tracking system based on a solar map and light-dependent resistor (LDR) sensors were evaluated in terms of power gain and system power consumption. It was discovered that compared to a continuous two-axis sun tracking system, the energy gain from a hybrid sun tracker is essentially non-existent. The proposed design system was found to have low energy consumption, high precision, and low cost [15]. The closed loop tracker was investigated with LDR sensors as input to the solar system. In contrast, in the case of an open loop system, a controller provides the driving signal to the servo motor based on current data inputs and the operating algorithm of the system. On days with cloudy conditions and days with clear skies, the proposed system may produce 12.8% and 26.9% more electricity than a static PV system. A dual axis with an open-loop solar tracker was also found to be automated and deployable anywhere on the plane

[16,17]. There are various tracking methods, but passive and active systems are the most common. In Passive systems, low-boiling-point gaseous liquids drive the system using the received solar radiation. On the other hand, active systems use gears and motors to manage PV modules. They are further classified into five driving systems: intelligent driving, Sensor driving, open-closed loop driving, micro-processor driving, and other combination driving based on the driving mechanisms used. Intelligent driver systems are one of the most used tracking systems nowadays because they use learning algorithms to forecast the precise location of the sun [18]. In another work, the optimum tilt and azimuth angles for the dual-axis tracker were used to optimize the power received by applying Harmony Search (HS) meta-heuristic algorithm. Based on the Julian date model, the performance was checked on six Chinese cities with diverse temperature zones to find the monthly optimum tilt and azimuth angles. The result was very close to the standard values [19]. A comparative study of the performance of the fixed, single-axis, and dual-axis solar tracking system over the year concluded that the generated electricity was found to be 2.25 MWh, 3.04 MWh, and 3.17 MWh, respectively. The single-axis tracking system produced about 35.30% more energy than the static system, while dual-axis solar tracker produced about 41.07% more than the static system under the same conditions. Fahad et al. (2019) concluded that the dual-axis tracker generated 3.96% more power than the single-axis tracker without considering the cloud. When the cloud effect was considered, the difference was reduced to 3.44% [20]. Due to the increasing demand for sustainable and green energy resources, Zaher et al. (2018) proposed an intelligent approach to optimize the orientation of continuous solar tracking systems on cloudy days. It was based on a 5-megapixel ground-based sky camera named 5481 VSE-C and provided by IDS imaging. It was equipped with a Fujinon fisheye lens and protected by a waterproof enclosure manufactured by automation and a fuzzy inference system. The experimental result for Perpignan city has efficiency gains of up to 9% relative to conventional continuous sun tracking systems under overcast conditions [21]. Saymbetov et al. (2021) developed a dual-axis schedule solar tracker with an adaptive algorithm to capture direct sunlight and maximize electrical power. It consists of a polycrystalline solar panel with a power of 60 W, an electric linear actuator, DC motor SV35-130/HP5 BFN, Atmega 328 microcontroller where a tracking algorithm was installed. The experimental work revealed that the amount of electrical power produced by the dual-axis schedule tracker with an adaptive algorithm in October and November was 58.55 WH and 82.54 WH, respectively [22]. In order to design a low-cost hybrid system to deliver the required thermal energy for heating water and street lighting, a parabolic collector combined with a dual-axis tracking mechanism with an LDR sensor was used to move the parabolic dish with the movement of the sun. The experimental work revealed that the thermal efficiency using

dual-axis and single-axis tracking mechanisms was 32.2% and 23.6%, respectively, with the same operating parameters [23]. This study focuses on the design and development of a dual-axis solar tracker using SIMULINK and SolidWorks. The tracker employs an LDR-based algorithm to monitor the sun's trajectory in real-time, utilizing time and location data. Results show that the tracking PV system produces more current than a static system, and the stand designed for 335-watt panels is strong enough to endure various wind conditions [24]. Another system, managed by a microcontroller, tracks the sun's position on both axes, with IoT monitoring for performance and alerts [25]. Additionally, a novel solar tracking system using a particle filter (PF) algorithm improves energy generation by 20.1% compared to fixed systems after a 60-day trial [26]. A dual-axis tracker with ESP8266 and LDR sensors enhances energy absorption and provides real-time monitoring of system data like temperature and power, with potential upgrades such as AI and remote monitoring [27]. D.A. Flores-Hernández D et al. 2024, this study presents a real-time tracking algorithm for dual-axis solar trackers, resulting in a 53.33% reduction in tracking movements, 60.77% reduction in operation time, and 14.18% decrease in energy consumption, while maintaining energy productivity. The tracker's energy consumption remains between 2% and 3% of the energy gain, ensuring profitability. Experimental findings show a mean tracking error of 4.61° (DMA) compared to 2.04° (CA), with azimuthal errors of 1.43° (DMA) and 0.27° (CA), and elevation errors of 4.39° (DMA) and 1.00° (CA). This method extends the tracker's lifetime by 6.8 times and integrates predictive maintenance through real-time data for improved durability and efficiency [28]. N. Koshkarbay et al. 2024 developed an adaptive control system for dual-axis solar trackers to optimize performance under varying weather conditions, using Clear Sky Index (CSI) thresholds and machine learning models for power output prediction with 99.3% accuracy. This system, integrating CSI-based decisions and forecasting, demonstrated energy generation improvements of 18.3% over horizontal configurations, 14.9% over single-axis trackers, and 10.01% over dual-axis trackers [29]. P. N. Praveen et al. 2024, proposed a dual-axis solar tracking system that combines an artificial neural network with an advanced particle swarm optimization method to improve energy forecasting accuracy. It uses meteorological data from Alice Springs and features innovative hidden layers for day/night data selection and input relevance. This approach enhances both ANN training and prediction performance. Future work will focus on multi-output fuzzified systems and further optimization improvements [30]. S.E. Bousbia Salah et al. 2024, introduced a solar dryer featuring a dual-axis tracking system and a parabolic trough concentrator, demonstrating efficient drying of apricots under desert conditions. The system achieved temperatures of up to 115°C, with an average thermal efficiency of 25.93%, and reduced the moisture content of apricots by 71.76%. Economic analysis

showed a low capital cost and a payback period of 0.43 years, highlighting its potential for widespread use in agricultural drying [31]. K. Kumba et al. 2024, explored the effectiveness of various solar tracking systems in enhancing energy output from photovoltaic panels, emphasizing dual-axis and hybrid trackers. It highlights advancements in tracking technologies, including machine learning and IoT integration, to improve energy capture efficiency. The research identifies key challenges and suggests future improvements, such as hybrid systems, cost-reduction strategies, and policy support, which are crucial for promoting sustainable energy solutions and improving the economic viability of solar energy systems [32].

The objective of this work is to enhance the available energy from the sun by utilizing a two-axis tracking device, which continuously makes the PV module align with the sun's rays. There are two methods for increasing the amount of solar radiation energy accessible. The first strategy is to absorb as much solar radiation as possible using the appropriate solar radiation-absorbing materials. The second method, presented in this work, is to use the optimization methodology for the solar photovoltaic panel over a day to increase radiations that reach the PV module. The optimization technique aims to maximize the PV panel's electrical power by tracking the sun's optimal tilt and azimuth angles. In this work, solar tracker optimization techniques like the Teaching learning-based optimization (TLBO) algorithm, Jaya algorithm, Rao-3 algorithm, Differential algorithm (DE), and Genetic algorithm (GA) are used. This work considers tilt and azimuth angles for a dual-axis solar tracker of a PV system, and the considered optimization algorithms

optimize it. The second aim of this work is to evaluate the optimum tilt-azimuth angle of the PV module using different algorithms. The purpose was to maximize the electrical power through the solar PV panel by tracking the sun's trajectories like azimuth and tilt angles throughout the day.

MATERIALS AND METHODS

A dual-axis solar tracker, as presented in Figure 1, is a mechanism designed to track the sun's trajectory in the sky, enabling it to make simultaneous adjustments in both horizontal (azimuth) and vertical (elevation) directions. This functionality allows the solar panel to maintain an optimal angle relative to the sun's position, thereby optimising its exposure and energy generation during the course of the day. This phenomenon leads to an enhancement in the efficacy of energy harvesting from photovoltaic panels.

TRAJECTORIES DETERMINED BY THE VARIOUS ALGORITHMS

Teaching Learning-Based Optimization (TLBO) Algorithm

Rao and Patel (2012) proposed the "Teaching Learning-Based Optimization" algorithm based on learning and teaching processes. The population size, number of generations, size, and other common regulating factors are required for the TLBO algorithm, whereas other algorithms require algorithm-specific control parameters.

There are two fundamental modes of algorithms: learners learning through teachers and learners interacting with

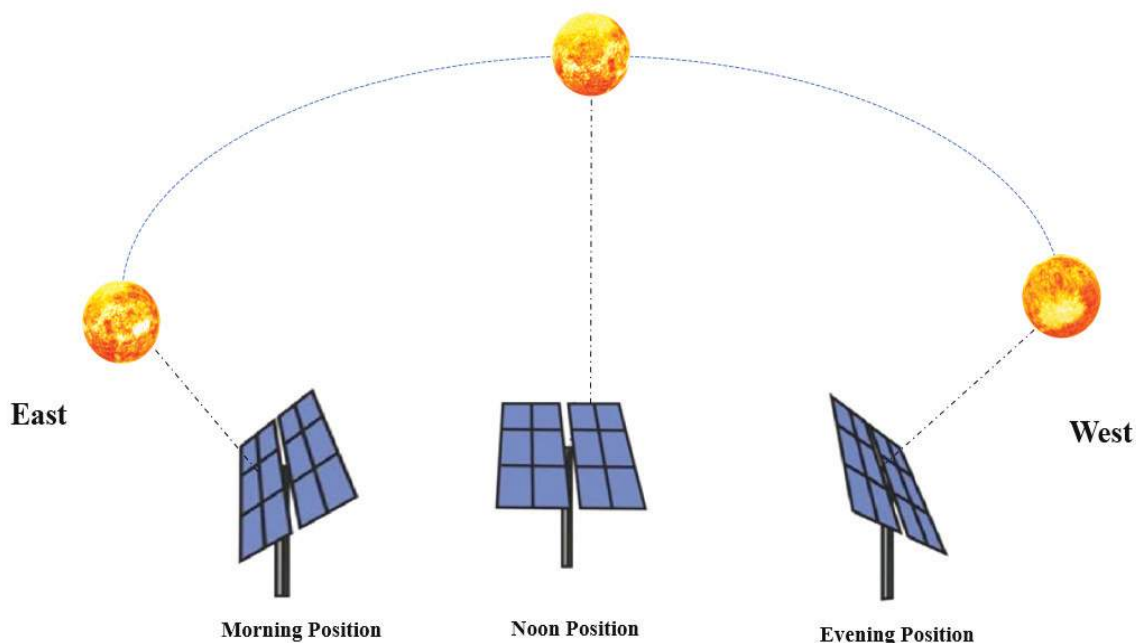


Figure 1. Schematic of dual-axis solar tracker.

other learners. The TLBO method considers a group of learners to be a population size, and different subjects presented to the group of learners to be different design factors for the given optimization problem, with the learners' outputs being the fittest value of the optimization problem. A teacher is regarded as the ideal solution among the general population. The design variable is a parameter included in the objective function of the given problem, and the best solution provides the best value for the objective function.

The working principle of the TLBO algorithm is given in Figure 2, [33].

JAYA Algorithm and Rao-3 Algorithm

The JAYA method is a new and easy optimization tool that addresses restricted and unconstrained difficulties, and its flowchart is represented in Figure 3 [34]. It is based on the best solution moving forward and the worst solution avoiding the population size N . The initial population for each design variable is created randomly within the lower

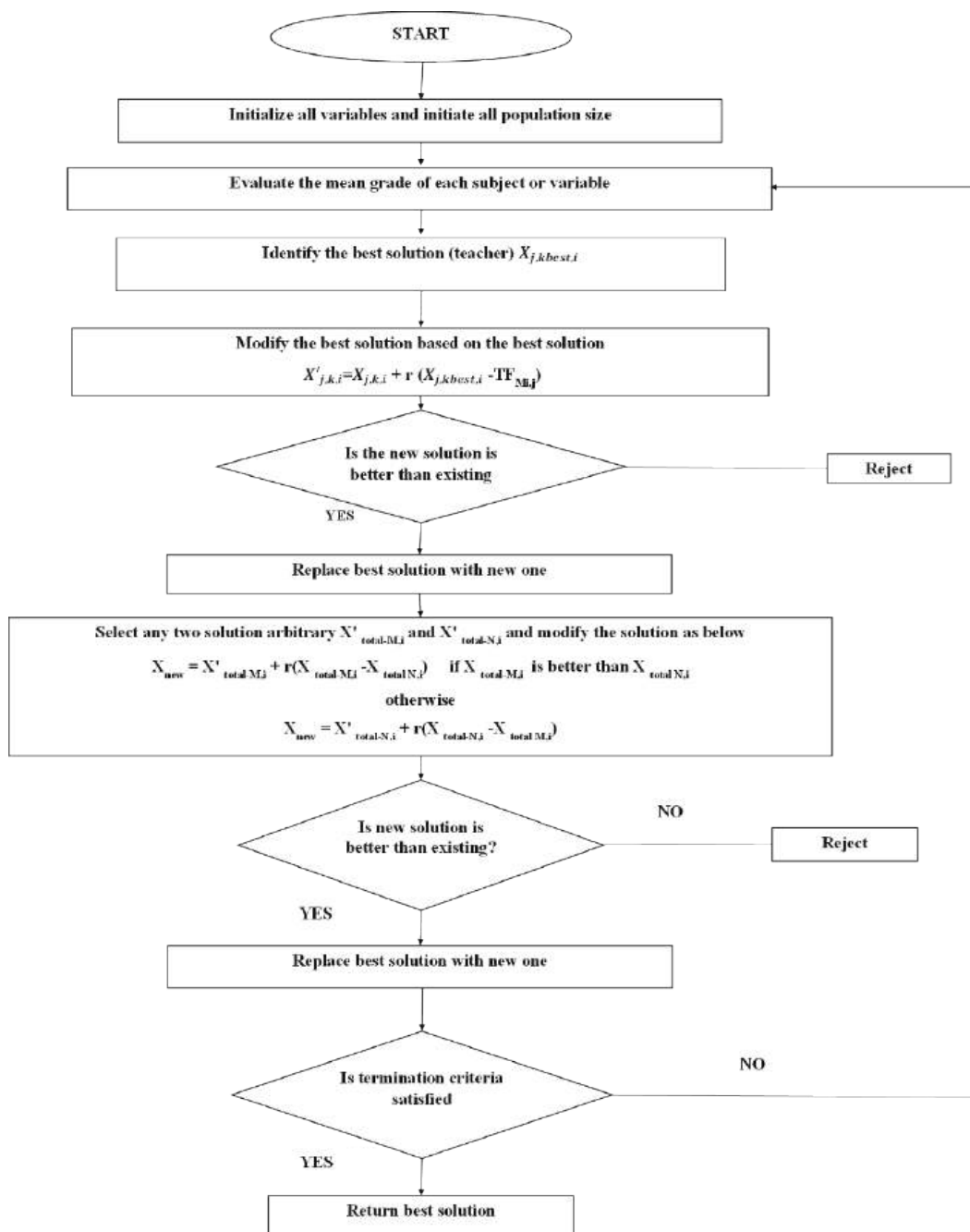


Figure 2. Flowchart of TLBO algorithm.

and upper bounds. Then, based on the idea of the best and worst candidate solutions within the population size, N , the values of the randomly generated variables are modified in each iteration. The main equation used in Jaya optimization algorithm is as follows by equation (1):

$$X'_{j,k,i} = X_{j,k,i} + r_{1,j,i}(X_{j,best,i} - |X_{j,k,i}|) - r_{2,j,i}(X_{j,worst,i} - |X_{j,k,i}|) \quad (1)$$

Where r_1 and r_2 are random numbers in the range $[0,1]$, the function values at the end of each iteration become the next iteration's input value.

Rao and Pawar (2020) proposed the Rao-3 algorithm 2020 based on Quasi-oppositional. For the Rao-3 Algorithm, the modified equations used in optimization are as follows by equation (2) and its flowchart as shown in Figure 3, [35]:

$$X'_{j,k,i} = X_{j,k,i} + r_{1,j,i}(X_{j,best,i} - |X_{j,worst,i}|) + r_{2,j,i}(|X_{j,k,i} \text{ or } X_{rand,j,k,i}| - (X_{rand,j,k,i} \text{ or } X_{j,k,i})) \quad (2)$$

Differential Evolutionary (DE)

The differential evolution (DE) algorithm is a straightforward, effective, and efficient method for tackling global optimization problems. The effectiveness of the differential evolution algorithm is highly dependent on its mutation strategy, which is discussed in [10,36]. The flowchart of the DE algorithm is shown in Figure 4.

In the initialization, the initial population includes the 'N' solutions (i.e., size of the population member or size), and each population size contains a 'D' variable, where D is the design variable of search space. Therefore, the solution of the population member at generation 'G' is given as equation (3):

$$X_i^G = (X_{i,1}, X_{i,2}, \dots, X_{i,D}), \quad \text{where, } i = 1, 2, 3, \dots, N \quad (3)$$

The member's initial population should be spread across the entire search space. The initialization approach for member's solution that is most usually used as equation (4):

$$X_{i,j} = X_j^{min} + rand(0,1) * (X_j^{max} - X_j^{min}) \quad (4)$$

Where X_{jmin} and X_{jmax} are the lower and upper bounds of the j_{th} dimension of the variable size, respectively, and $rand(0,1)$ is a fundamental value between "0" and "1".

DE mutation strategy

DE/rand/1

The DE strategy used in the mutation, which is given by the equation (5):

$$V_i^G = X_{r,1}^G + F(X_{r,2}^G - X_{r,3}^G) \quad (5)$$

Where r_1 , r_2 , and r_3 represent the distinct integer randomly generated within the range of $[1, N]$, and F is the scaling factor used to determine the mutation scale. The scaling factor's range is restricted to $(0,1)$.

Crossover operator

After mutation, the trial vector $U_i^G = (U_{i,1}^G, U_{i,2}^G, \dots, U_{i,D}^G)$ is created for each population, which is obtained by the equation (6):

$$U_{i,j}^G = \begin{cases} V_{i,j}^G & \text{if } rand_i(j(0,1)) \leq CR \text{ or } j == jrand \\ X_{i,j}^G & \text{otherwise} \end{cases} \quad (6)$$

Selection operator

Based on their fitness value, the selection operator selects whether the target vector survives and moves on to the next generation. The decision vector for the maximizing issue is as follows by the equation (7):

$$X_i^{G+1} = \begin{cases} U_i^G & \text{if } (fit(U_i^G) \leq fit(X_i^G)) \\ X_i^G & \text{otherwise} \end{cases} \quad (7)$$

Genetic Algorithm (GA)

Holland, 1984, developed an algorithm known as a genetic algorithm that is based on Charles Darwin's theory of natural evolution. The natural selection of members of population size is the first step in the Genetic Algorithm (GA), a process of evolution. They give birth to children who carry their parents' genes, and these children's virtues

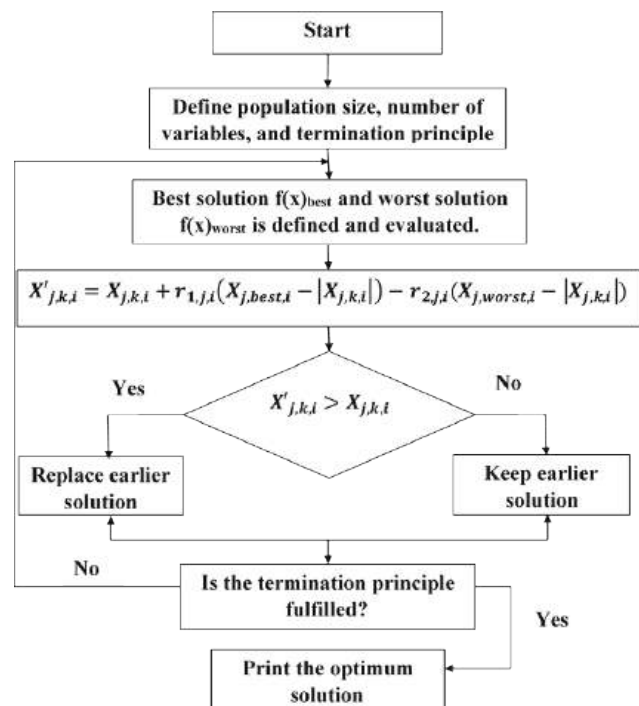


Figure 3. Flowchart of JAYA and Rao-3 algorithm.

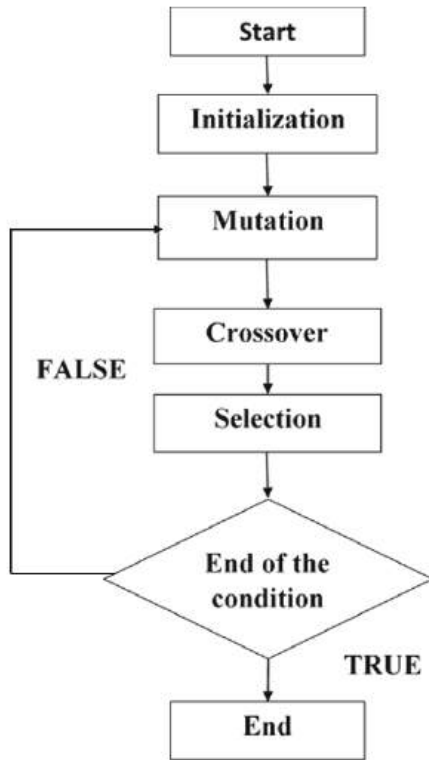


Figure 4. DE algorithm flow chart.

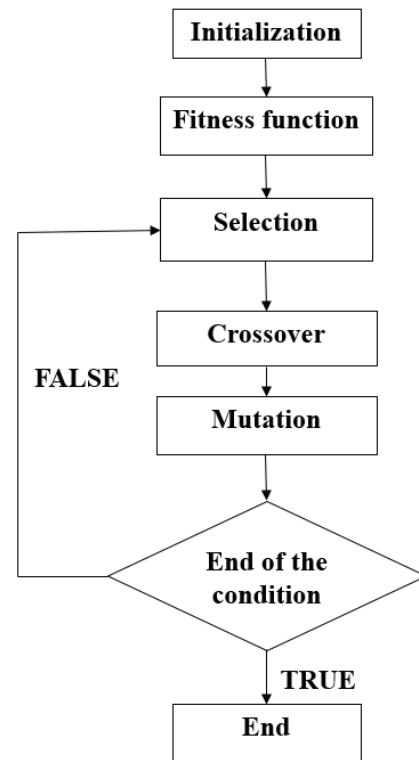


Figure 5. Flowchart of GA algorithm.

will be transmitted to the following generation. If the parents' qualities have a higher fitness value, their kids will outperform the parents' genes and have a better chance of survival. This algorithm will continue to iterate until the fittest individuals are identified. A genetic algorithm has five well-thought-out phases, as shown in Figure 5 [37].

The above-proposed algorithms are used to determine the total solar radiation at optimal tilt and azimuth angles. All of these methods used the same set of optimization parameters to find the best optimal solution. Our methods were designed to function together with a population size of 172 individuals, an iteration count of 1000, and two optimization variables, tilt angle and azimuth angle. In case of the Genetic algorithm, to encourage genetic variety in the population, we kept the crossover probability at 0.8 and the mutation scale factor at 0.5, facilitating the exploration of novel solution spaces.

Mathematical Model

By the mathematical model of the solar orbit and position and "Radiation maximization" demand, the objective function is:

$$\text{Max}(I_c(\beta, \gamma)) \quad (8)$$

Equation (1) represents the maximization of solar radiation (I_c) at an optimized azimuth angle (γ) and tilt angle (β).

Constraints:

(i) Azimuth angle:

$$\gamma_{min} \leq \gamma \leq \gamma_{max} \quad (9)$$

Where γ_{min} & γ_{max} represents limits between the lower and upper limit values. In the optimization problem, the azimuth angle lies between $[-110, 110]$.

(ii) Tilt angle:

$$\beta_{min} \leq \beta \leq \beta_{max} \quad (10)$$

Where β_{min} & β_{max} represent limits between the lower and upper limit values. The tilt angle optimization problem lies between $[0^\circ, 90^\circ]$.

The calculation of electrical power generated within the PV system can be determined by integrating the instantaneous value of solar radiation received by the surface of the PV panel with specified time intervals. The system efficiency is influenced by different factors like tracking algorithms, solar radiation receiving effective surface area of the PV panel, changing time with the sun's movement throughout the day, and the DC/DC converter and inverter efficiency.

The solar radiation time-dependent for reaching the earth's surface depends on the extra-terrestrial sunlight-based radiation estimation, which is reduced by the quality of air present in the atmosphere. Lowering the values of the sun's radiation reaching the earth's surface are

depicted as components of the sunbeam length through the atmosphere as clearness indices. On clear days, deliberate radiation from the sun on the surface of the earth and predicted extra-terrestrial solar beam develop their qualities over time.

The total radiation or instantaneous radiation (I_h) is the sum of the direct or beam radiation (I_{bh}) and the diffuse radiation on the horizontal surface (I_{dh}) as expressed in equation (11) [38].

$$I_h = I_{bh} + I_{dh} \quad (11)$$

The direct radiation on the horizontal surface (I_{bh}) and diffuse radiation on the horizontal surface (I_{dh}) depends on the Geographical location, day of the year, time, and weather situations. The lowering of solar radiation is increasing in tandem with the lengthening of the sun's ray pathways inside the atmosphere. The impact of the atmosphere on the scattering and absorption of radiation fluctuates over time due to changes in atmospheric conditions and air mass. Defining a standard "clear" sky and calculating the amount of radiation received on a horizontal surface during each hour and day under these standard conditions is beneficial. The estimation of solar beam radiation passes through clear atmosphere on a horizontal surface by considering the zenith angle (θ_z) and altitude for standard atmospheric circumstances. Therefore, the reduction of solar radiation is defined by the transmittance coefficient for beam (I_{bh}) and diffuse radiation (I_{dh}) for clear days, which are denoted by τ_b and τ_d , respectively, and can be calculated by the following equation [13,38]:

$$\tau_b = \frac{I_{bh}}{I_t} \quad (12)$$

And the estimation of the clear-sky diffuse radiation (I_{dh}) on horizontal surfaces are as follows:

$$\tau_d = \frac{I_d}{I_t} \quad (13)$$

Where transmittance coefficient for beam radiation can be calculated by the equation (14):

$$\tau_b = a_0 + a_1 \exp\left(\frac{-k}{\cos \theta_z}\right) \quad (14)$$

The values of the constants a_0 , a_1 , and k for the standard atmosphere with a visibility of 23 km are determined based on the values of a_0^* , a_1^* , and k^* , which are provided at altitudes below 2.5 km.

$$a_0^* = 0.4237 - 0.00821(6 - A)^2 \quad (15)$$

$$a_1^* = 0.5055 + 0.00595(6.5 - A)^2 \quad (16)$$

$$k^* = 0.2711 + 0.01858(2.5 - A)^2 \quad (17)$$

Where A is the altitude from the sea level in km, the constants a_0 , a_1 , and k can be calculated by applying the correction factors: $r_0 = \frac{a_0}{a_0^*}$, $r_1 = \frac{a_1}{a_1^*}$, and $r_k = \frac{k}{k^*}$. The values of correction factor are shown in table 1 for four different climatic types. Similarly, the transmittance coefficient for diffuse radiation on a horizontal surface during clear days can be calculated using the following equation (18):

$$\tau_d = 0.271 - 0.294\tau_b \quad (18)$$

Table 1. Correction factor for four different climates [Duffie et al. [38], with permission from John Wiley and sons].

Type	r_0	r_1	r_k
Tropical	0.95	0.98	1.02
Mid-latitude summer	0.97	0.99	1.02
Sub-arctic summer	0.99	0.99	1.01
Mid-latitude winter	1.03	1.01	1.00

The average extra-terrestrial solar radiation is denoted by I_p , which is obtained by equation (19), [17,38]:

$$I_t = \frac{1}{\Delta t} \int_t^{t+\Delta t} I_0 e(t) \sin \alpha(t) dt \quad (19)$$

Where, I_0 is the solar constant which is equal to 1367 w/m², and the eccentricity factor is given as equation (20), [38]:

$$e(t) = 1 + 0.034 \cos \frac{2\pi n}{365} \quad (20)$$

Where n represents the n th day of the year, counted from the first January. The total solar radiation represented by ' I_c ' in equation (14) is the function of the solar radiation (I_h), diffuse radiation (I_{dh}), and beam radiation (I_{bh}) on the surface of the PV panel with optimal tilt and azimuth angles for a different time is obtained as equation (21), [17,38]:

$$I_c = I_{bh} \frac{\cos i}{\sin \alpha} + I_{dh} \frac{(1 + \cos \beta)}{2} + \rho I_h \frac{(1 - \cos \beta)}{2} \quad (21)$$

Where ' ρ ' represents the ground reflectance factor and the value is 0.7, taken from Seme & Štumberger, 2011 and Seme et al., 2017, and ' i ' represents the incidence angle of beam radiation from the sun to the solar PV surface. The relation between solar incidence angle and the other sun angles is obtained as equation (22), [38]:

$$\cos i = \cos \alpha \cos(\gamma_s - \gamma_w) \sin \beta + \sin \alpha \cos \beta \quad (22)$$

Where ' γ_s , γ_w ' represents solar azimuth and wall azimuth angle, respectively, and ' α ' represents the altitude angle of

the sun, which is obtained as equation (23) and zenith angle is inverse of $\sin \alpha$ [38]:

$$\sin \alpha = \sin L \sin \delta_s + \cos L \cos \delta_s \cos \omega_s \quad (23)$$

And,

$$\omega_s = 15^\circ(12 - T) \quad (24)$$

Where in above equations (23) & (24), L , δ_s , ω_s , & T represent the latitude angle for a particular geographical location, solar declination angle, solar hour angle, and solar time, respectively.

The integration of the electric energy for a given time interval $[t_1, t_2]$ gives the produced electrical energy by the following equation is given as equation (25), [38]:

$$E_{pv} = \eta_{pv} A_{pv} \int_{t_1}^{t_2} I_C(t) dt \quad (25)$$

Where, in equation (25), E_{pv} , η_{pv} , A_{pv} & I_C represent the PV system that has produced electrical energy. The total efficiency consists of the PV panel, dc/dc converter and inverter, active surface area of the panel, and total solar radiation during a predefined time interval $[t_1, t_2]$, respectively.

RESULTS AND DISCUSSION

This section deals with the results obtained during the optimization of dual axis solar tracker for a PV module in order to achieve maximum solar radiation at optimum tilt-azimuth angles and further maximize the electrical energy by using optimization algorithms like JAYA algorithm, Teaching learning-based optimization (TLBO) algorithm, Rao-3 algorithm, Differential evolutionary algorithm (DE) and Genetic algorithm (GA). The codes of algorithms were run in MATLAB software to determine the

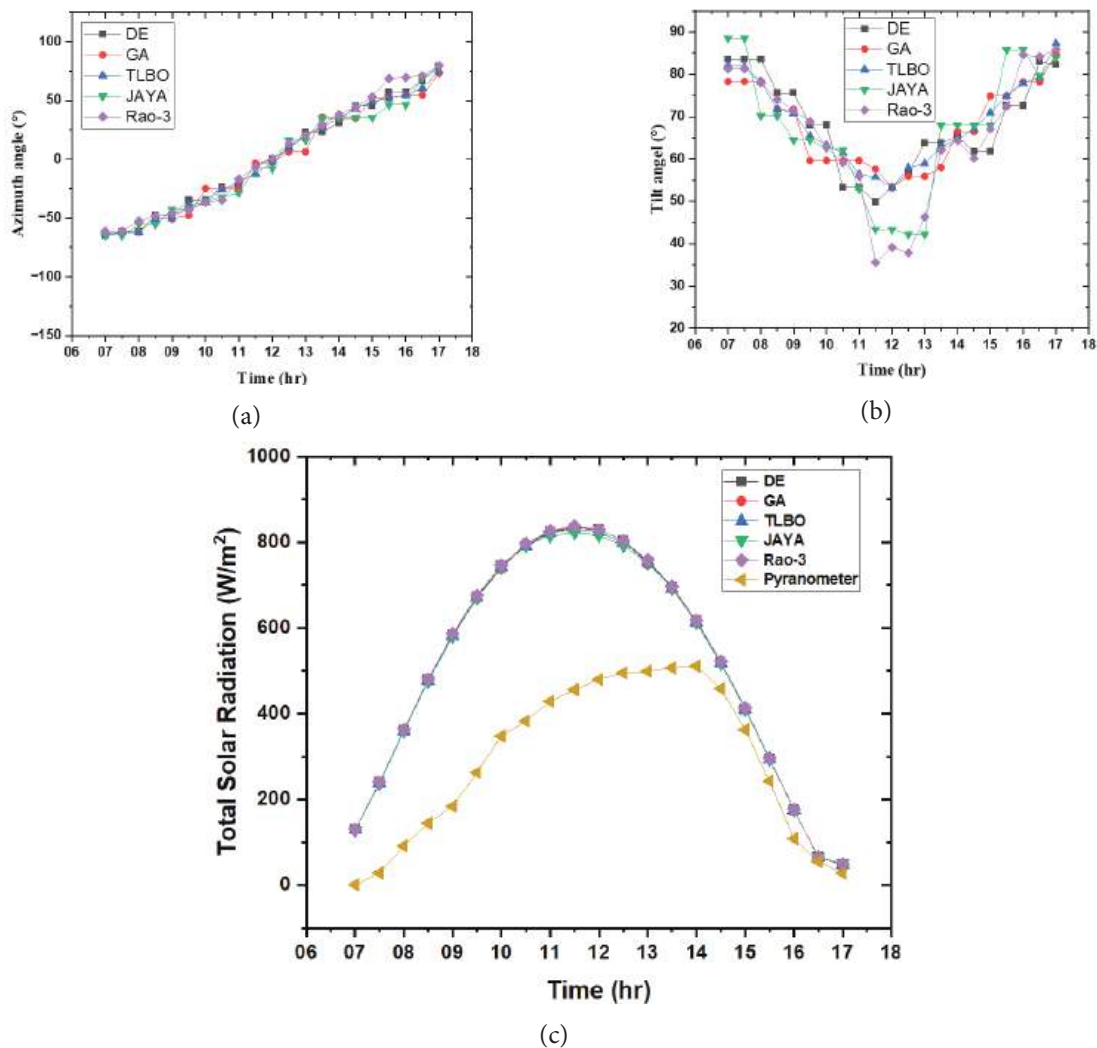


Figure 6. Results for winter day, $n^{\text{th}}=350$, at Surat city: (a) Optimum Azimuth angle β v/s time, (b) Optimum tilt angle β v/s time, and (c) Total solar radiation determined by proposed algorithms and Pyranometer.

optimum azimuth and tilt angles to obtain the maximum solar radiation tracked.

The trajectory of the solar tracker depends upon the mathematical modeling, geographical location (i.e., Latitude and longitude), solar time, and date for the particular location. In this work, the data were taken for the location of Surat city, India, which comes under tropical climate type: Latitude=21.1702° N, Longitude=72.8311° E. The lower and upper limits of solar tilt angles are $[0^\circ, 90^\circ]$, and for solar azimuth angles are $[-110^\circ, 110^\circ]$ over a day from 7 AM to 5 PM. By implementing the parameters mentioned above into the optimization algorithms for Winter Days, $n^{\text{th}}=350$, Summer Days, $n^{\text{th}}=136$, and spring Days, $n^{\text{th}}=58$ to determine the total solar radiation at optimum tilt angle and solar azimuth angle for a dual-axis solar tracker. The optimization results show the total solar radiation variation for optimum tilt and azimuth angles throughout the day. The total solar radiation exhibits a progressive increase until reaching its peak at noon, followed by a fall from 7

AM to 5 PM when the optimal tilt and azimuth angles are employed. The study examines the variability of total solar radiation and determines the optimal tilt-azimuth angles for different algorithms.

Figure 6 (a) represents the result of the optimum azimuth angle on winter days ($n^{\text{th}} = 350$ days of the year), which vary between -110° to 110° with the different times from sunrise to sunset. The azimuth angle is negative azimuth angle before noon, zero at noon, and positive azimuth angle in the afternoon. Figure 6 (b) represents the result of the optimum tilt angle, which is determined by the above-proposed algorithm for the winter days ($n^{\text{th}}=350$ days of the year). The tilt angle varies between $[0^\circ, 90^\circ]$, and the tilt angle minimum at the noon position. Figure 6 (c) shows the result of optimum solar radiation for the winter days (i.e., $n^{\text{th}}=350$ days of the year) by the above optimization algorithms. The maximum solar radiation for the dual-axis tracking system was obtained at different optimum tilt angles and azimuth angles. In addition, the amount of

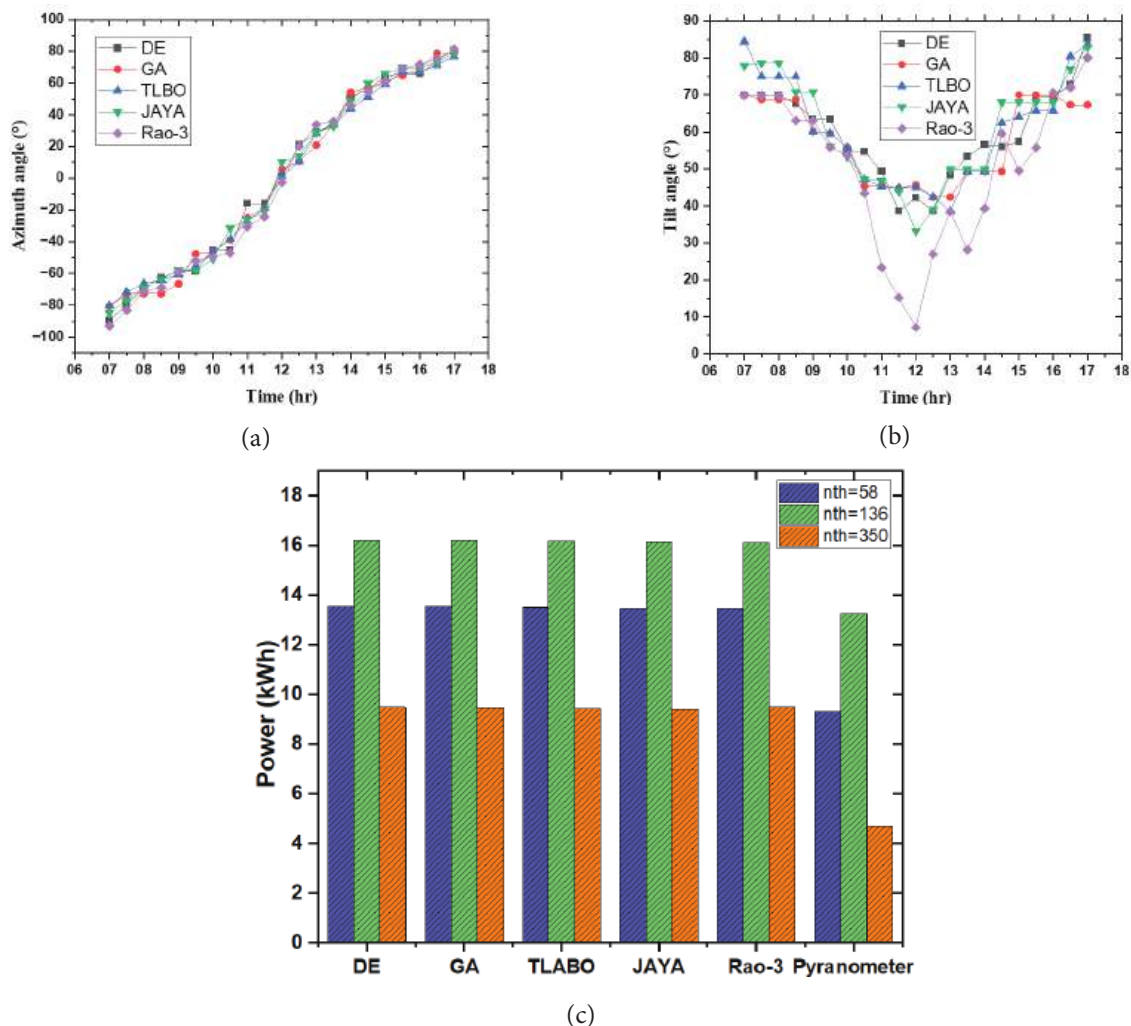


Figure 7. Comparative results of electrical power for spring day (58th), summer days (136th), and winter day (350th) of Surat city (coordinates: 21.1702° N, 72.8311° E), determined by various algorithms and measured.

electrical energy that is contained within the PV panel was evaluated and depicted in Figure 7. This was determined using calculations based on a polycrystalline PV panel with an active area of 15.32 m^2 and a total efficiency of 12 %.

Figure 8 (a) represents the optimized value of azimuth angle throughout the day on a typical summer day in Surat City. It can be seen that the results obtained for DE, GA, TLBO, JAYA, and Rao-3 algorithms are almost same. The variation in tilt angles are given in Figure 8 (b), the curves are forming a valley shaped structure with the peakpeak value given by different algorithm. Figure 8 (c), as shown above, represents the maximum tracked total solar radiation for the dual axis tracker obtained at optimum azimuth and tilt angle, as shown in Figure 7 (a) and Figure 7 (b), respectively, determined by the use of optimization algorithms from 7 AM to 5 PM. At the optimized angles, the variation of solar radiation throughout the day is shown in Figure 7 (c). Further, the electrical energy in the PV panel was evaluated, as shown in Figure 9. This was calculated for a polycrystalline PV panel with an active area and total efficiency of 15.32 m^2 and 12%, respectively.

Figure 9 (c) represents the result for the maximum solar radiation obtained at optimum tilt and azimuth angle as shown in Figure 9 (a) and (b), respectively, at Surat city coordinates 21.1702° N , 72.8311° E for the 136th day of the year. The results of total solar radiation obtained by the proposed method compared with the result obtained by measured data by pyranometer, which is more than the measured data. Further calculations for the PV panel's electrical output, as depicted in Figure 7, were made using a polycrystalline PV panel with an active surface area of 15.32 m^2 and a total efficiency of $\eta = 0.12$ (including the PV panel, dc/dc converters, and inverter).

The electrical energy calculation was based on total solar radiation for a 2.5 kW photovoltaic (polycrystalline silicon PV) system with an active surface area of 15.32 m^2 . The total measured efficiency, which contains the efficiency for solar photovoltaic (PV) panels, dc/dc converters, and MPPT inverters, is $\eta = 0.12$ [17]. The results show that the dual-axis solar tracker gives additional output power than the measured.

The Electrical power for the tracked system at Surat city, India, for spring days, summer days, and winter days for different algorithms are as under: For spring days, the electrical power generation by different optimization algorithms are as follows: TLBO-13.4953 kWh, JAYA-13.4664 kWh, Rao-3-13.4805 kWh, DE-13.5539 kWh, and GA-13.5437 kWh. These results demonstrate notable enhancements compared to the electrical output of the measured fixed system, which stands at 9.3150 kWh. For summer days, the electrical power generation by different optimization algorithms are as follows: TLBO-16.1713 kWh, JAYA-16.1436 kWh, Rao-3-16.1125 kWh, DE-16.2079 kWh, and GA-16.1969 kWh. These results demonstrate notable enhancements compared to the electrical output of the measured fixed system, which stands at 13.2610 kWh. For winter days, the electrical power generation by different optimization algorithms are as follows: TLBO-9.4490 kWh, JAYA-9.4071 kWh, Rao-3-9.5174 kWh, DE-9.4881 kWh, and GA-9.4846 kWh. These results demonstrate notable enhancements compared to the electrical output of the measured fixed system, which stands at 4.3852 kWh.

The optimization algorithms are compared with that of Seme et al. 2011 [9, 10]. The results are obtained for Slovenia, Maribor ($46^\circ 33' \text{ N}$, and $15^\circ 39' \text{ E}$), as shown in Figure 10, and was found to be in good agreement. It is now

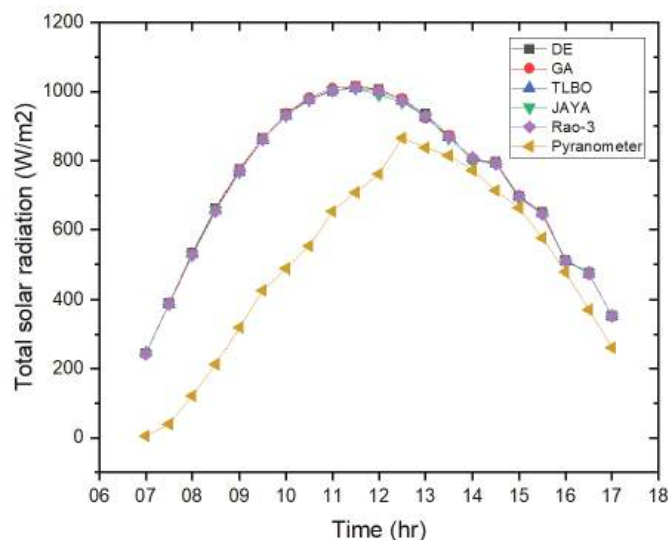


Figure 8. Results for spring Day, $n^{\text{th}}=58$, at Surat city: (a) Optimum Azimuth angle β v/s time, (a) Optimum tilt angle β v/s time, and (c) Total solar radiation determined by proposed algorithms and Pyranometer.

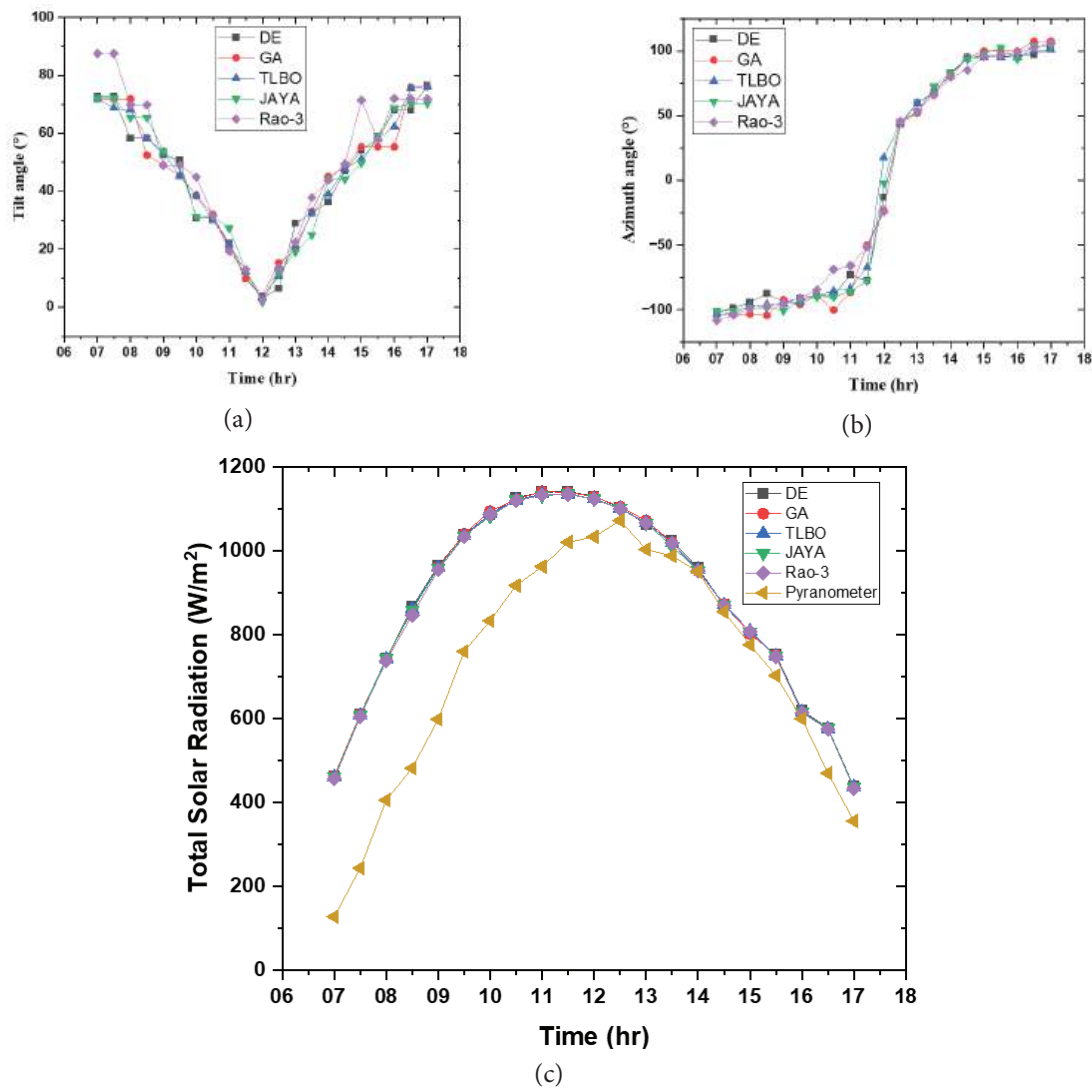


Figure 9. Results for summer day, $n^{th}=136$, at Surat city: (a) Optimum tilt angle β v/s time, (b) Optimum Azimuth angle β v/s time, (c) Total solar radiation determined by proposed algorithms and Pyranometer.

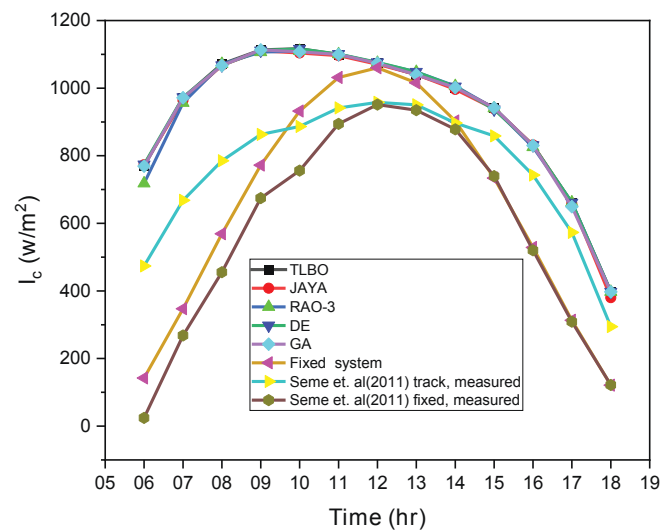


Figure 10. comparative results for summer days, $n^{th}=199$: The solar radiation for the city Maribor, Slovenia (coordinates: 46° 33' N & 15° 39' E), track measured and fixed measured with the calculated tracked and fixed system (at a constant angle, $\beta=24^\circ$) determined by various algorithms.

shown that the dual-axis solar tracker gives additional output power than the fixed system.

CONCLUSION

The present study was to optimise solar radiation absorption by photovoltaic (PV) modules in order to increase the amount of energy derived from sunlight reaching their surface. A non-linear and bounded optimisation problem is utilised to assess the optimal paths of tilt angle and sun azimuth angle. Following are the conclusions that may be drawn from this work:

- The objective function is in the form of an explicit function that is not known. Optimization method like the TLBO algorithm, JAYA algorithm, Rao-3 algorithm, DE algorithm, and GA algorithm have been utilised for the purpose of assessing the objective function. The electrical power increment achieved by various proposed algorithms, such as TLBO Algorithm, JAYA Algorithm, Rao-3 Algorithm, Differential Evolution Algorithm, and Genetic Algorithm, in comparison to a measured fixed system, demonstrated significant enhancements in power generation.
- The optimization results for different algorithms show that the output electrical power with the dual-axis tracker is more than that of the fixed system.
- The Electrical power for the tracked system at Surat city, India, for spring days, summer days, and winter days for different algorithms are as follows. For spring days, the electrical power generation by different optimization algorithms are TLBO-13.4953 kWh, JAYA-13.4664 kWh, Rao-3-13.4805 kWh, DE-13.5539 kWh, GA-13.5437 kWh, and measured fixed system-9.3150 kWh.
- For summer days, the electrical power generation by different optimization algorithms are as follows: TLBO-16.1713 kWh, JAYA-16.1436 kWh, Rao-3-16.1125 kWh, DE-16.2079 kWh, and GA-16.1969 kWh, and measured fixed system-13.2610 kWh.
- For winter days, the electrical power generation by different optimization algorithms are as follows: TLBO-9.4490 kWh, JAYA-9.4071 kWh, Rao-3-9.5174 kWh, DE-9.4881 kWh, and GA-9.4846 kWh, and measured fixed system-4.3852 kWh.
- The results obtained through the algorithms were compared with the experimental results from the literature for Maribor, Slovenia, and found to be in good agreement.
- The optimization algorithm based dual axis tracker greatly improves the solar energy trapped for industrial applications like drying, hot water generation, building heating, and power generation.
- The integration of Internet of Things (IoT) with a dual-axis solar tracker further helps in large solar farms so that they can be remotely monitored, diagnosed, and operated.

NOMENCLATURE

β	Tilt angle
γ	Azimuth angle
α	Altitude angle
ρ	Ground reflectance factor
i	Incidence angle
δ_s	Solar declination angle
ω_s	Solar hour angle
$[t_1, t_2]$	Time interval
N	Population size
I	Solar radiation
l	Sunbeam path length
K	Clearness indices
I_0	Solar constant (1367 w/m ²)
L	Latitude angle
T	Solar time
E_{pv}	Electrical energy by PV system
η_{pv}	Efficiency of PV panel
A_{pv}	Surface area of PV panel

Subscript

Max	Maximum
Meas	Measured
Min	Minimum
n	n th day of the year

Abbreviation

TLBO	Teaching Learning Based Optimization
GA	Genetic Algorithm
DE	Differential Evolution
PV	Photovoltaic

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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.

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Research Article

Relative assessment of single-tube and multi-tube inverse diffusion flame for thermal and emission characteristics

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CO and NOx Emission; Flame Length; Flame Shape and Appearance; Inverse Diffusion Flame; Multi-tube IDF

ABSTRACT

The thermal and emission characteristics of multi-tube inverse diffusion flame is investigated and compared with single-tube inverse diffusion flame. Liquefied petroleum gas is used as a fuel. The study focuses on flame length, CO and NOx emissions and temperature profiles under varying equivalence ratios and air velocities. Multi-tube configurations enhance air-fuel mixing by increasing the contact surface area. This results in shorter and more compact flames that results in low NOx emissions compared to single-tube inverse diffusion flames. The flame length of the multi-tube inverse diffusion flame reduces by an average of 12.7% compared to the single-tube configuration. The temperature profile of the multi-tube inverse diffusion flame reveals distinct flame zones at higher air velocities, while at lower air velocities, the flames merge into a homogeneous high-temperature region in the post-combustion zone. CO emissions reduce significantly with the multi-tube inverse diffusion flame, which emits 18 PPM less CO than the single-tube inverse diffusion flame at low equivalence ratios. Conversely, NOx emissions exhibit an opposite trend, with higher levels observed at elevated equivalence ratios and at reduced air velocities. The multi-tube inverse diffusion flame consistently shows lower NOx emissions, emitting 4 PPM less at higher equivalence ratios and 3 PPM less across all velocity variations compared to the single-tube configuration. A novel multi-tube inverse diffusion flame burner design, featuring air and fuel ports divided into multiple smaller, circumferentially arranged segments improves air-fuel mixing, stabilizes the flame, reduces emissions, and provides valuable insights for optimizing burner performance for optimum emissions and flame length.

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INTRODUCTION

Premixed and non-premixed flames are the two primary categories into which flames can be separated. Compared

to premixed flame combustion, diffusion flame combustion offers greater stability, safety, and a wider working range, making it a very practical option for many combustion systems [1]. One unique kind of diffusion flame is the

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inverse diffusion flame (IDF). The IDF emits less soot than the normal diffusion flame (NDF) [2]. IDF benefits from a high velocity air jet that assures fuel entrainment and improves fuel-air mixing. IDF enjoys wider range of stability, greater operating safety, and less emissions compare to NDF [3]. IDF's dual-flame structure is comparable to partially premixed flames [4–6]. IDF's thermal and emissions characteristics can be improved by achieving optimum air-fuel mixing. Depending on the extent of the air-fuel mixing needed, either active or passive mixing treatments are used. Ng *et al.* [7] use outside force air (active treatment) to create and control air-fuel mixing. The passive mixing approach doesn't require an outside force. Geometry adjustments (modification in air and fuel jet arrangements) are used to create the passive mixing systems. It is worth noting that the passive approach is more reliable and does not add complexity to the existing burner system. Smaller tubes help in enhancing turbulence, and more tubes contribute to expanding the air-to-fuel area of interaction. Increased contact area between the air and fuel, resulting in enhanced mixing and more effective burning. IDF with co-annular (CoA) burner layouts which has an annular fuel port surrounds turbulent centre air-jet is one way to increase surface area available for air fuel mixing.

The momentum difference between fuel and air jet is another factor influencing the entrainment/mixing between the outer fuel jet and core air jet of IDF. Increasing the velocity difference between the fuel and air jets results in partially premixed flame and improves air-fuel mixing. Higher mixing between the fuel and air jet results in the formation of a compact blue flame with a lower flame height. Appropriate control of the air-fuel momentum ratio, and change in burner geometry helps to optimise the IDF flame characteristics like visible flame shape, flame luminosity, temperature distribution, and emissions [4,8–11]. A higher heat transfer rate and improved stability can be achieved by the highly turbulent flame and enhanced mixing created by the higher air jet momentum entraining the fuel jet inward. Simultaneously, the surrounding ambient air in the vicinity of the fuel jets can be forced into the combustion zone to ensure cleaner and more efficient burning [12]. In the study of turbulent methane inverse diffusion flames (IDF) in coaxial burners, Sobiesiak and Wenzell [4] observed partial premixing that evolved into a well-mixed reaction zone. It was concluded that the flow parameters and nozzle shape affect amount of partial premixing which occurs on the flame centreline.

Reduced levels of soot and NO_x and high levels of CO emissions are desired characterise the IDF [13–16]. Inverse diffusion flames prevent excessive surface development of soot particles because of the shorter flame and the associated shortened residence durations in the window of the stoichiometric strength in flame envelope. Santoro *et al.* [17] reported that the rate of soot formation increases in the annular region of the flame as residence time increases. The effects of flow characteristics on soot formation and oxidation in non-premixed coflowing hydrocarbon–air flames

were experimentally investigated by Lin and Faeth [18]. The soot particles cross the flame sheet earlier when the air velocity is higher than the fuel velocity due to entrainment patterns. The fuel-rich zone has a shorter residence time and less particle surface growth. Due to their small size and short surface time for growth in the fuel-rich zone, these particles can completely oxidise after crossing the flame sheet. Thus, flames with higher fuel-to-air velocity ratios generate more soot. Kaplan and Kailasanath [13] studied the flow-field influence on soot production in methane-air IDFs and discovered that the surface growth rate of IDFs is very slow along the particle path line, and soot is produced significantly less than that in NDFs.

IDF is an excellent choice for air-staged burners as it is possible to achieve extremely low levels of NO_x emissions [15]. Turbulence plays key role in deciding optimum level of NO_x emissions. Overall, greater mixing lowers the NO_x emission index, where the lower peak temperature overcomes the longer residence time. Mishra [19] observed that when the Reynolds number increases, the NO_x level drops. Partridge *et al.* [15] used IDFs on staged-air combustors to obtain ultra-low NO_x emissions. Primary findings were that the highest NO_x emission index occurred at an equivalent ratio of one [16].

One way to improve performance of IDF is to use circumferentially arranged fuel ports (CAP) with coaxial air (CoA). This approach improves the circumferential fuel jets, central air jet and ambient air contact area. Investigations were conducted to capture thermal and emission characteristics of the flame under various equivalence ratios and air-jet Reynolds numbers [7,10,20–28]. Sze *et al.* [10] investigated turbulent LPG-air IDFs stabilized on CoA and CAP, with an array of fuel jets surrounding the centre air jet. Better air-fuel mixing had been reported in CAP burners compared to coaxial IDFs. This facilitate reduced NO_x emissions via NO-reburning. The temperature distribution patterns of both flames were comparable, but the CAP flame was shorter with higher temperatures, greater flame stability, higher fuel-air contact ratio, and quicker rates of oxygen depletion, indicating better fuel-air mixing, and more efficient combustion at low Reynolds numbers. CAP IDFs described for better air-fuel momentum flux ratio (MFR), enhanced flame stability, higher temperature, and low emissions [23,24].

In CAP burner ratio of air jet diameter (d_a) to fuel jet diameter (d_f) determines burner performance. Dong *et al.* [29] investigated the impact of the diameter ratio (d_a/d_f) on the burner's thermal and emission characteristics using the CAP burner. A smaller diameter (d_a) leads to a blue colour flame with increased maximum flame temperature and improved thermal characteristics. Higher flame stability is observed over broader range of air jet Reynolds numbers and larger band of overall equivalence ratio. Smaller diameter (d_a) results in efficient combustion generating higher CO, HC and emitting less NO_x. Dong *et al.* [21] highlighted that NO_x and soot emissions are reduced and high thermal

efficiency is offered by CAP burners. Barakat *et al.* [28] examined two CAP burner geometry. One with a centrally positioned air-port and the other with circumferentially distributed air-ports. It is observed that circumferentially distributed air-ports offers less emissions. While working on open and impinging inverse diffusion flames, Choy *et al.* [30] claimed that while smaller air jet diameters lower NOx emissions but noise radiation increases due to high turbulence intensity. Şener *et al.* [31] found that the heating height influence the thermal performance of the burner in impinging flames and the emission levels of unburned CO, NO, and HC. Mawali *et al.* [32] discovered that fuel velocity significantly influences the gradient of concentration inside the mixture. Enhancing the fuel velocity led to an increased gradient concentration inside the mixture. Badiger *et al.* [33] conducted an experimental analysis on the thermal characteristics of a jet impinging on a flat vertical surface with and without swirl. The placement of the impinging plate relative to the inner reaction zone of the inverse diffusion flame affects the heat intensity parameter on the striking plate.

The reported study clearly demonstrates that active and passive methods are helpful in achieving optimum thermal and emission characteristics of IDF. Passive methods like use of small-diameter air-ports and CAP burners facilitate an enlarged contact area for the air and fuel, leading to improved mixing and more efficient combustion. While numerous studies have focused on circumferentially arranged fuel ports, research on circumferentially arranged air ports remains limited. Building on this fact, present study explores the division of both air and fuel ports into multiple smaller ports. The research aims to assess the impact of flow parameters on flame morphology, including temperature, length, and emissions of multi-tube IDF. A

comparative analysis of flame morphology between multi-tube IDF and single-tube IDF is presented to determine the configuration that yields optimum mixing and emissions.

EXPERIMENTAL SETUP AND METHODOLOGY

The schematic diagram of the test facility as shown in Figure 1 is develop and utilize to investigate the effects of flow parameters on single-tube and multi-tube IDF characteristics. The main components of the test-setup are the air blower, air flow meter, mass flow controller, fuel cylinder, SCADA panel, burner, digital camera, flue gas analyser and enclosure shown in Figure 2. Customized software is designed to control and to measure the flow of fuel and air. Additionally, it stores and display real-time data of temperature, VFD frequency, and fuel and air flow rates. VFD is set at specific frequency to control and regulate the air flow rate.

An annular fuel tube with a diameter of 37 mm and an air tube with a diameter of 19 mm are used in the single-tube IDF geometry, as shown in Figure 3(a). For the multi-tube IDF configuration, the diameters of the air and fuel tubes are derived by keeping total cross-sectional area constant between single-tube and multi-tube IDFs. This reduces diameters of the air and fuel tubes as number of tubes increases. In the multi-tube IDF configuration, one tube is centrally positioned, while the other three tubes are arranged concentrically on pitch circle diameter (PCD) of 40 mm. Geometrical arrangement of four tube IDF is described in Figure 3(b). In the multi-tube IDF configuration, the air tube diameter is 9.5 mm, while the fuel tube diameter is 18.5 mm. Figure 4(a) shows a multi-tube IDF burner installed inside the enclosure.

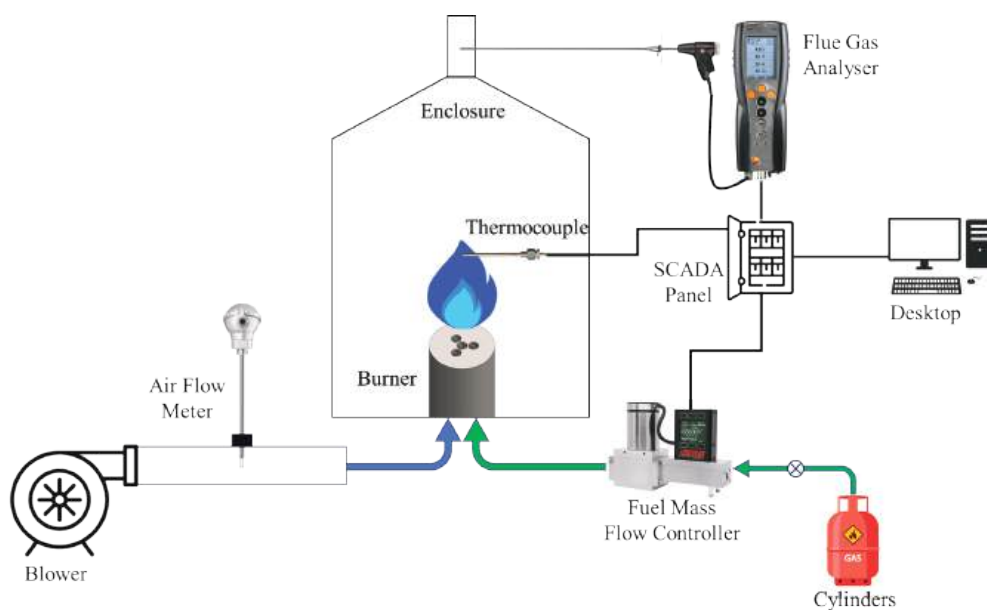


Figure 1. Schematic diagram of the test facility.

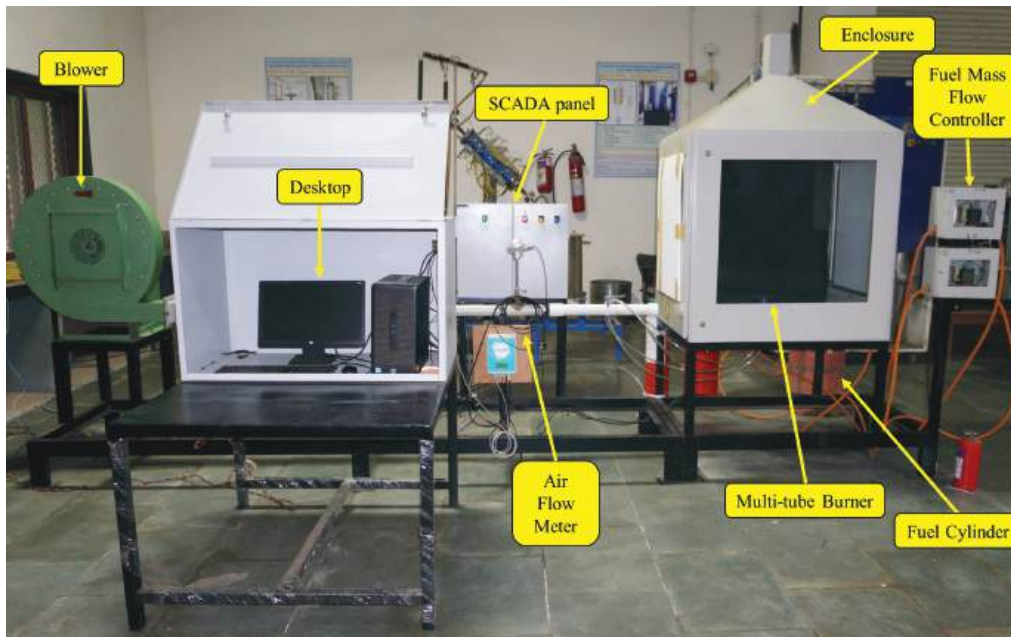


Figure 2. Computerized experimental setup.

Air is supplied via a centrifugal air blower with a 7.5-HP electric motor. In downstream of blower sufficiently long pipe used so that flow becomes fully develop. A calibrated LEOMI make insertion type thermal mass flow meter (LEOMI 586) is used for measuring air flow rate with an accuracy of 1.5% of the reading. A variable-frequency drive (VFD) is used to change the frequency and voltage supplied to the electric motor to set the required air flow rate. LPG (40% propane, 60% butane) is used as a fuel. The mass flow controller (Alicat Scientific: 0–50 SLPM) with an accuracy

of $\pm 0.8\%$ of reading and $\pm 0.2\%$ of full-scale reading is used to regulate and control the fuel flow rate.

B-type thermocouple (bead diameter = 1 mm, $\pm 2^\circ\text{C}$) is employed to measure the flame temperature. The thermocouple is placed on a vertical height gauge, as shown in Figure 4(b). The axial and radial flame temperature distributions are measured by positioning the thermocouple on the cross-sliding table and height gauge. In the axial direction, temperature is measured at 5, 10, 15, 20, 30, 40, 50, 70, 85, and 100 mm from the base of the burner. In the radial direction, temperature is measured at 0, 5, 10, 15, 20, 25, 30,

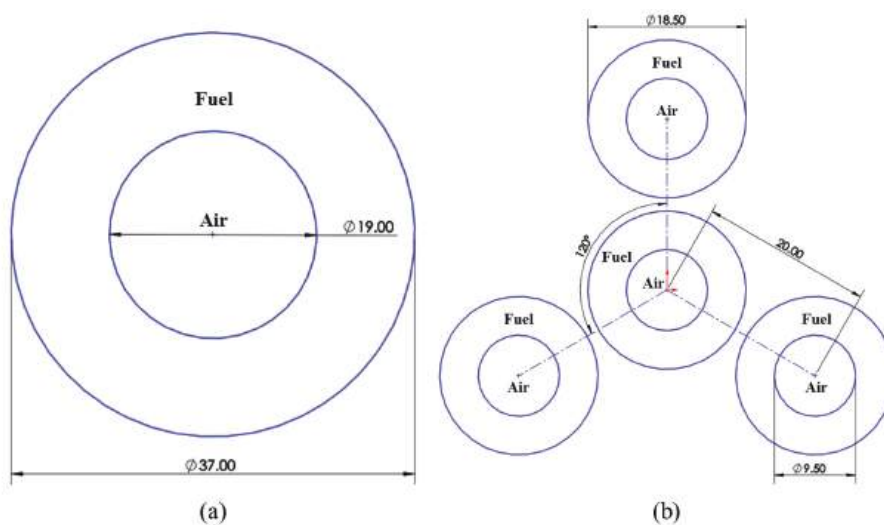


Figure 3. (a) Single-tube and (b) Multi-tube IDF burner.

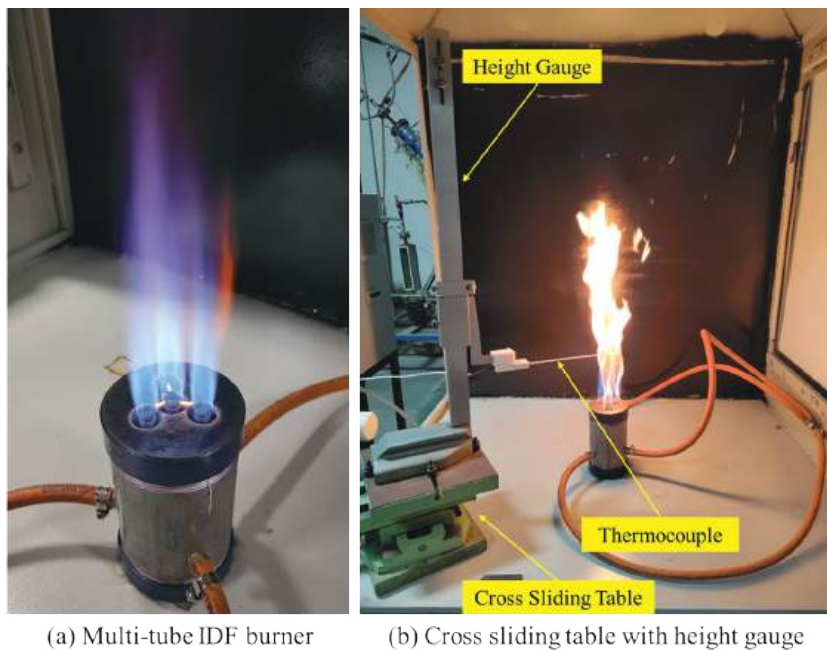


Figure 4. Temperature measurement setup.

35, and 40 mm from the centre of the tube. The thermocouple is placed horizontally so that it remains perpendicular to the flow. The measured temperature is corrected for radiative heat losses [34].

The appearance and length of the flame were analysed using images taken with a digital camera (Sony Alpha ILCE 6000Y) equipped with a 24.3 MP APS-C CMOS sensor and ISO 100–51200 sensitivity. Videos of flames are recorded at 50 frames per second. Each frame is then extracted, and the average image of the flame is produced by combining all of the extracted images. This average image is used to find out the visible flame length.

An enclosure with a cross-section of 1000 mm × 800 mm and a height of 1000 mm is fabricated to avoid interference from the surrounding. To measure the emission, a convergent section is constructed on the top of the enclosure.

A calibrated flue gas analyser (Testo 340-flue gas analyser) is used to measure exhaust gas emissions. A probe with a ceramic filter is inserted into the duct and set normal to the flow direction. CO and NO_x emissions are measured using the analyser.

The studies are conducted for four tubes geometry in different operating conditions. LPG fuel is used in experiments with different equivalence ratios. By changing the fuel flow rate while keeping the air flow rate constant, the equivalence ratio can be changed. In this case, the air velocity remains constant. The air flow rate is kept constant at 125 SLPM, resulting in an air velocity of 7.35 m/s. The parameters for adjusting the equivalence ratios (ϕ) in the experiment are listed in Table 1.

The effect of air velocity alteration on thermal and emission characteristics are also investigated by changing

Table 1. Test conditions for different equivalence ratios (ϕ) (Air flow rate: 125 SLPM, $V_{air} = 7.35$ m/s)

ϕ	LPG flow (SLPM)
0.82	3.60
0.93	4.11
1.05	4.63
1.17	5.14
1.28	5.66
1.40	6.17
1.52	6.68

Table 2. Test conditions for different air velocities (Fuel flow rate: 5.14 SLPM)

V_{air}	Air flow (SLPM)
5.65	96.15
6.12	104.17
6.68	113.64
7.35	125.00
8.16	138.89
9.18	156.25
10.5	178.57

the air flow rate at a constant energy input of 8.63 kW. The fuel flow rate needs to be kept constant in order to maintain a constant energy input to the burner. For this, fuel supply rate is maintained at 5.14 SLPM for constant energy input of 8.63 kW. Table 2 displays different test conditions for varying air velocity at a constant energy input.

RESULTS AND DISCUSSION

The equivalence ratio, and air velocity are varied to study thermal and emission characteristics of single-tube and multi-tube IDF. The main motto of the present work is to observe changes in flame shape and appearance, flame length, and flame emissions as air and fuel mixture strength varies. Variation in mixture strength can be altered in two ways. In the first approach, fuel flow rate is kept constant, and air flow rate is varied by changing air velocity. Data are tabulated in table 2. This is also a case of constant energy input. In the second approach, air flow rate is kept constant and fuel flow rate is changed. Flow data for this approach are presented in table 1. In this approach, energy input to the burner also changes with a change in fuel flow rate. Thus, variation of mixture strength is achieved in either way, but in the case of constant fuel flow rate, energy input is fixed, while in the second case, energy input alters. The shape and appearance of the flame will change depending on the above approaches. The above approaches also have an effect on flame characteristics. In each case, the emissions from the two burners are compared. Temperature contours are plotted on the central plane, passing through the axis of central and circumferential tubes.

Flame Structure

Flame structure of single-tube IDF is studied by Patel and Shah [3]. The IDF consists of two sections as shown in Figure 5. The bottom section is referred to as a flame base,

in which fuel is entrained towards the core of the air jet. The upper section, known as the flame torch, is made up of the inner reaction zone and the post-combustion zone and is where fuel and air mix together and combustion takes place. The flame torch and flame base are connected by the flame neck with the shortest cross-sectional area. In IDF, the degree of fuel entrainment by a high-velocity central air jet controls the fuel's mode of burning [25–27], whether it is premixed or diffusion. A blue zone is always visible after the entrainment zone, indicating that the combustion is premixed.

Multi-tube IDF is having different flame structure as compared to single-tube IDF. In multi-tube IDF entrainment zone is observed at circumferential tube only. Also core reaction zone is divided into two parts. One is for central tube and another is for circumferential tube. Core reaction zone for different tube is separated by yellow sooty zone as shown in Figure 6. This yellow sooty zone is having teardrop shape. This teardrop like soot zone is because of fuel accumulation at this location. After core reaction zone, unburnt fuel gets oxidized in post combustion zone. Post combustion zone is also having two zones. Inner post combustion zone is clearly visible in yellow color but outer tube post combustion zone is not visible with naked eyes.

Effects of Variation in Equivalence Ratio (ϕ)

Variation of the equivalence ratio is done by changing the fuel flow rate at a constant air flow rate. This keeps air momentum constant. Flame shape, appearance, length and emissions are measured and variations are discussed in following sections.

Flame Shape and Appearances

Figure 7 and 8 shows variation in flame shape and appearance for single-tube and multi-tube IDF respectively. This variation indicates that flame length increases with an increase in fuel flow rate for both the IDFs. A blueish zone

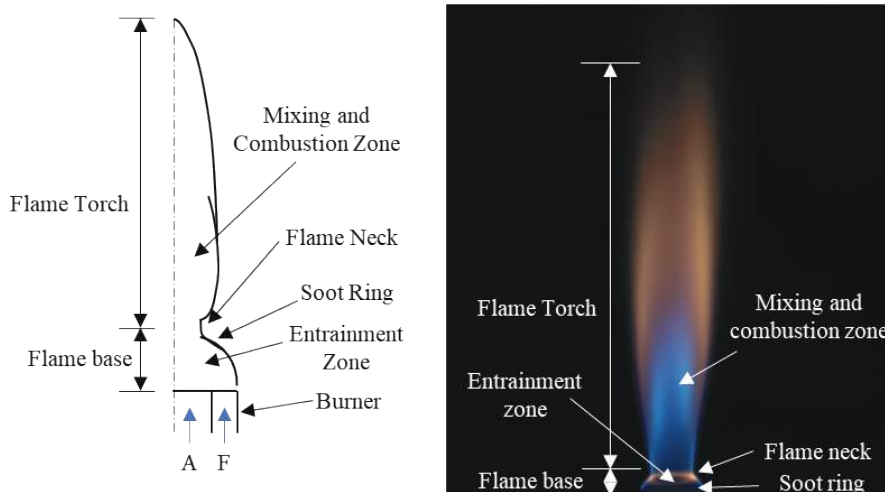


Figure 5. Flame structure of single-tube IDF.

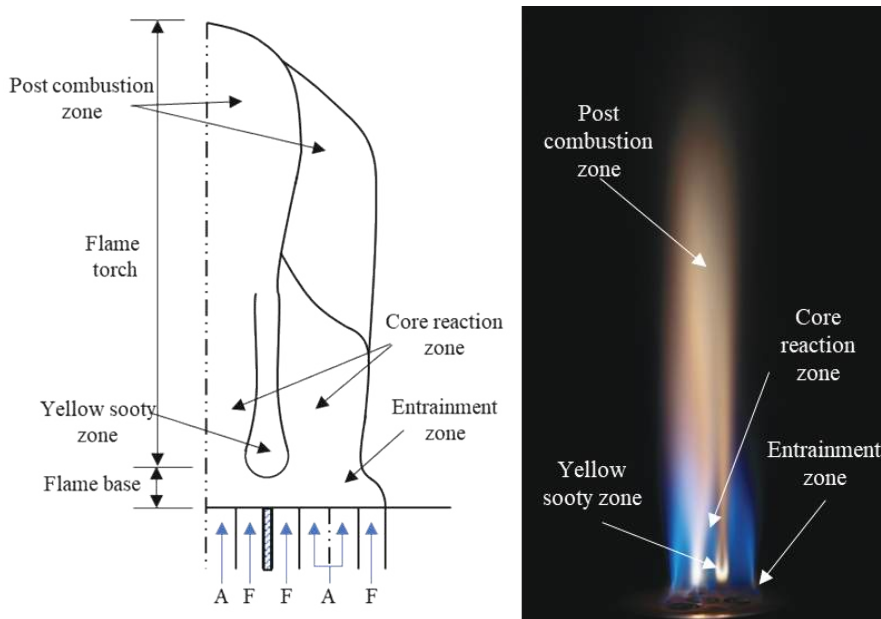


Figure 6. Flame structure of multi-tube IDF.

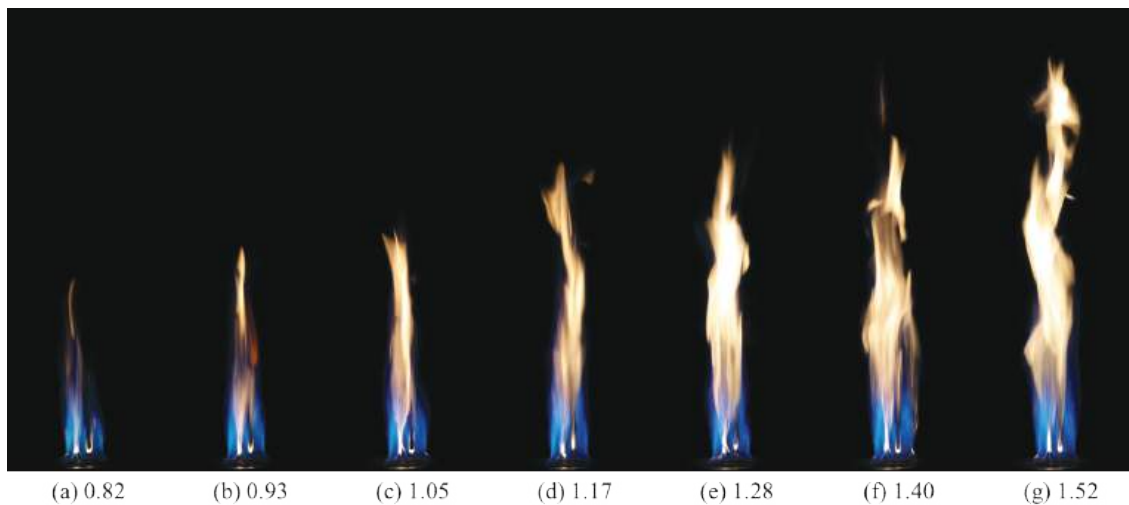


Figure 7. Flame images of multi-tube IDF at different equivalence ratios (ϕ).

appears at the base of the flame, while the flame tip is yellowish in colour. This yellowish colour is because of soot formation in the IDF. From Figure 7 and 8, it is observed that the blue colour region has almost the same blue zone length throughout all equivalence ratios. This might be because of constant air momentum. As the equivalence ratio increases, the length of the yellowish region increases. At a higher fuel flow rate, supplied air is not sufficient, so unburnt fuel takes more time to get sufficient oxygen from surrounding air, which results in a higher flame length and a yellowish-sooty flame region at the top of the flame.

Higher equivalence ratios lead to longer residence times for oxidation, which facilitate the rapid formation of soot particles and increase soot production. Soot particles cross the flame sheet earlier when the air to fuel velocity ratio is higher at a lower equivalence ratio, resulting in a less yellowish-sooty zone [17,18]. In multi-tube IDF, the yellow spot observed near the base of the flame is because of fuel accumulation between the centre fuel tube and the annular fuel tube. For multi-tube IDF at a lower equivalence ratio, there is a separate flame for each tube. But as the equivalence ratio increases, separate flames merge and form a single large flame. Because of the higher air velocity, fuel gets

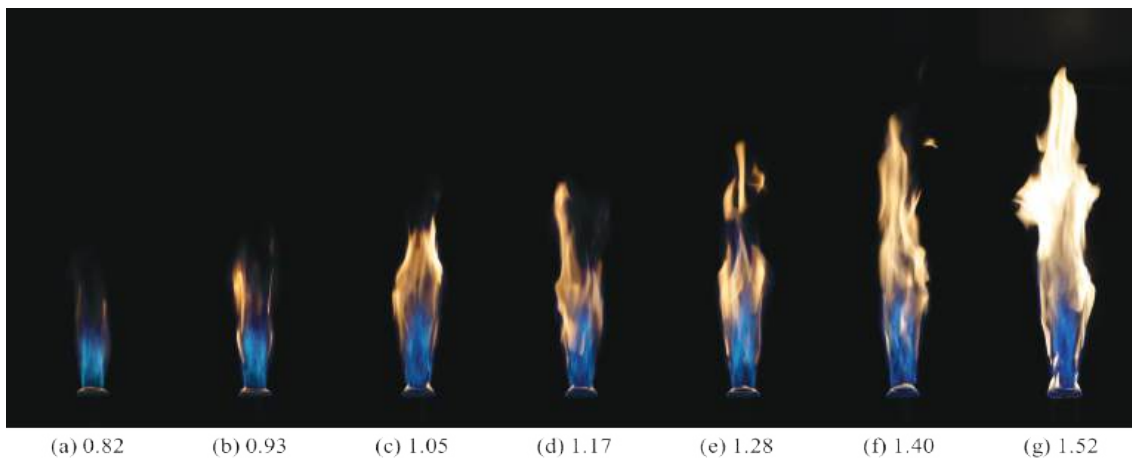


Figure 8. Flame images of single-tube IDF at different equivalence ratios (ϕ).

entrained in the centre of single-tube and multi-tube IDF. This entrainment zone creates a blueish reaction zone with better mixing and enhanced combustion.

In single-tube IDF, a similar trend is observed for the blue and yellow regions of the flame. In single-tube IDF, a yellow soot ring near the entrainment region is also observed [3,35].

Flame Length

Flame length is an important parameter for diffusion flames. Flame length measurements have been used to verify models of flame structure and calculate the residence time of soot particles [36–40]. The flame length indicates the distance it will take to completely consume the fuel in

any burner. Flame length is calculated from the average image of the flame. A 30-second recording is converted into multiple frames, and the average of these frames gives an average image of flame. Flame length is calculated using an open-source tool named ImageJ. The known length is used to set the scale in ImageJ.

The flame length of single-tube and multi-tube IDF is compared at different equivalence ratios. Figure 9 compared the change in flame length with variation in equivalence ratio. As the equivalence ratio increases, the flame length also increases. At a higher equivalence ratio, fuel supply is greater, which takes more time to completely burn the fuel. Because of the higher fuel flow rate, unburnt fuel in the main combustion zone is consumed in the post-combustion zone, which is the main reason behind the higher flame length at higher equivalence ratio. From Figure 9, it is observed that the flame length for single-tube IDF is shorter in almost all cases except for the equivalence ratio of 0.82. At a lower equivalence ratio, single-tube IDF has a shorter flame length.

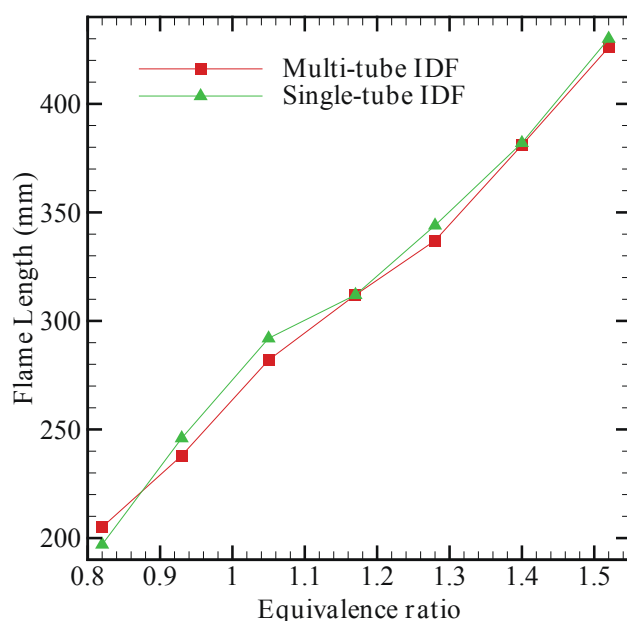


Figure 9. Flame length at different equivalence ratios.

Flame Emissions

Flame emissions are sensitive to mixture strength vary with alteration in equivalence ratio. CO emissions are mainly due to incomplete combustion happening in flames. In the case of the yellow sooting flame, CO gets oxidized, which results in lower CO emissions. Figure 10 shows a similar trend, where high CO emissions are observed at an equivalence ratio of 0.82. It is observed that single-tube shows higher emission at a 0.82 equivalence ratio compared to multi-tube IDF. At a lower equivalence ratio, the multi-tube IDF emits 18 PPM less CO than the single-tube IDF. At equivalence ratio 0.82, a single-tube IDF has higher CO because it has less contact surface area with ambient air. Multi-tube IDF has more ambient air contact, which helps in the oxidation of CO into CO₂. CO emissions are being reduced by increasing the equivalence ratio. At equivalence

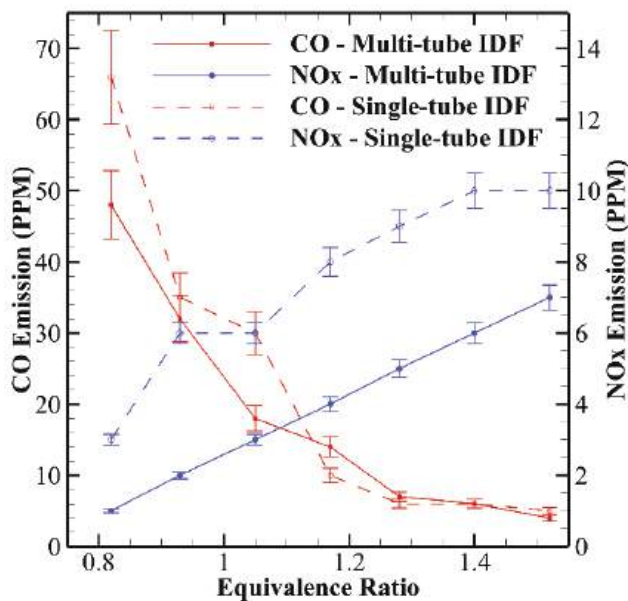


Figure 10. CO and NOx emissions at different equivalence ratios (ϕ).

ratios above 1.28, the variation in CO emissions is negligible and comparable for both single-tube and multi-tube IDF. The difference in CO emissions for multi-tube and single-tube IDF is nearly 18 PPM at a lower equivalence ratio. At the other end, CO emissions are comparable at a higher equivalence ratio.

NOx is generated at higher temperatures due to the longer residence time for N_2 in high temperature zones. As shown in Figure 10, NOx emissions increase with an increase in the equivalence ratio. For multi-tube IDF, NOx emissions show a linear trend. The difference in NOx

emissions between single and multi-tube IDF increases with increases in the equivalence ratio. From Figure 10, it is clearly seen that at a higher equivalence ratio, NOx emission is higher for single-tube IDF. In comparison to single-tube IDF, multi-tube IDF reduces NOx emission by 4 PPM at higher equivalence ratio. As the equivalence ratio increases, flame length increases, and high temperature zone length also increases. N_2 remains in a high-temperature zone for a longer period of time. So that NOx emission is increased due to the higher residence time of N_2 . NOx emission for multi-tube burner is lower compared to single-tube burner. This is mainly because of the shorter flame length in the case of a multi-tube burner. Higher surface area and high turbulence in a multi-tube reduces over all flame temperature. This reduces NOx emissions in multi-tube compared to single-tube IDF.

Effects of Variation in Air Velocity

Effect of change in air velocity at constant fuel flow rate is considered to investigate effect of air momentum on flame shape, appearance, flame length and emissions.

Flame Shape and Appearances

From Figure 11 and 12, it can be easily observed that reducing air momentum increases flame length. flame colour is completely blueish at higher air momentum. At lower air momentum, the blueish region is shorter. As air velocity increases, fuel and air layers with different velocities will experience higher shear strain, and this difference in velocity leads to enhanced mixing [41]. This enhance mixing results in faster fuel burning with blueish region of flame. The yellowish region has maximum flame luminosity at lower air velocities. In higher momentum, in case of multi-tube IDF a separate flame for each tube is clearly visible. This distinction vanishes as air velocity reduces. At low air velocity and lower momentum scenario, a yellowish

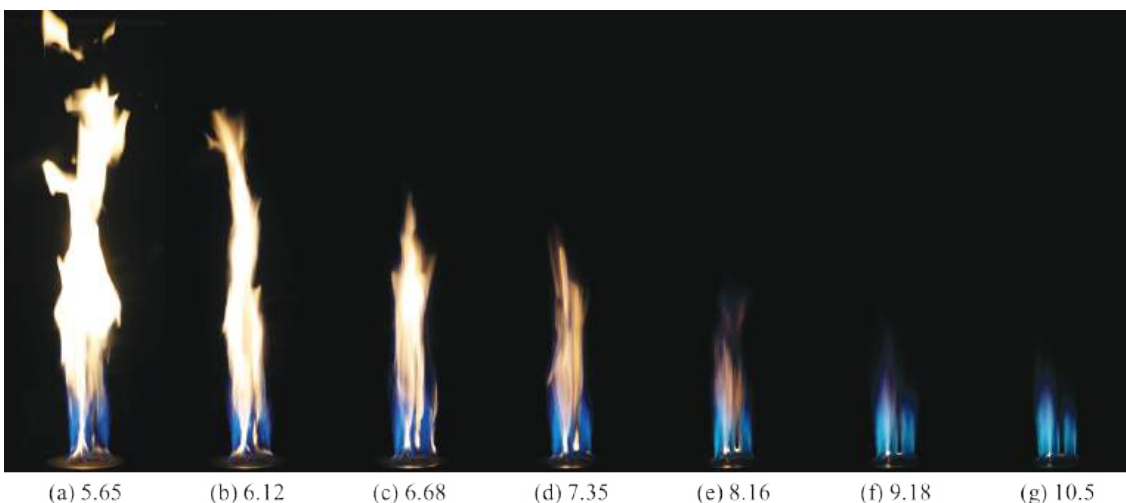


Figure 11. Flame images of multi-tube IDF at different air velocities.

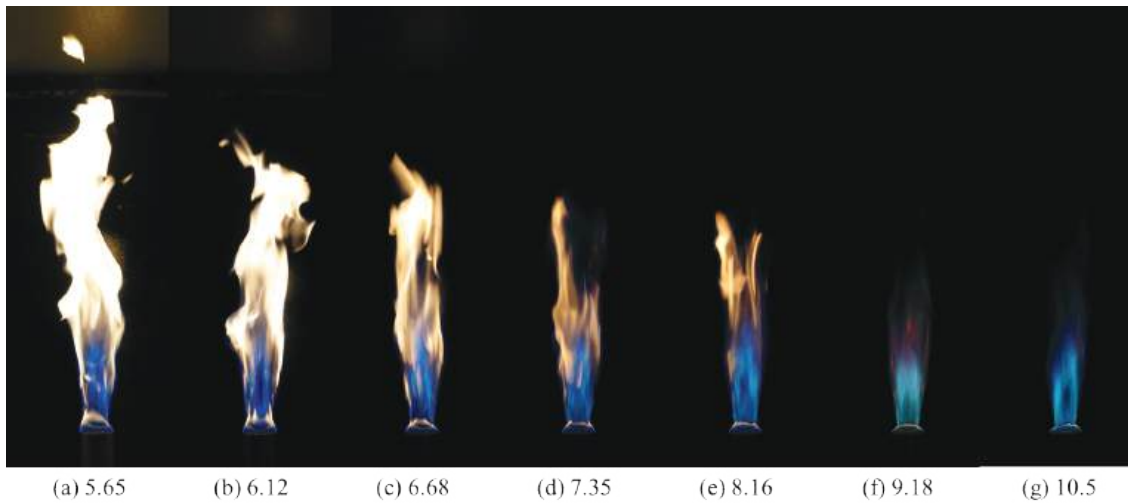


Figure 12. Flame images of single-tube IDF at different air velocities.

region starts from the flame base and burner exit. This makes it difficult to identify separate flame for the different tubes of multi-tube IDF.

Flame Length

Variation of flame length with change in air velocity for single and multi-tube IDF is described in Figure 13. It is observed from Figure 13 that flame length reduces as air flow increases. At lower air flow, because of the lower momentum of the air, the flame has less turbulence, which results in a longer flame length. At a higher air flow rate, air and fuel momentum have more differences, which helps increase the strain rate in the near base region. This

higher strain rate increases turbulence and reduces flame length. In all air velocity cases, multi-tube IDF produces a shorter flame length. The flame length of a multi-tube IDF gets shorter by an average of 12.7% (37 mm) compared to a single-tube IDF. The maximum variation in flame length is 54 mm when the velocity is 9.18 m/s. An increase in the number of tubes helps increase the air-to-fuel contact area, and smaller tubes help in increasing turbulence [42]. This reduces flame length.

Flame Emissions

Figure 14 depicts CO and NO_x variation with respect to change in air velocity. Trends indicate that at higher

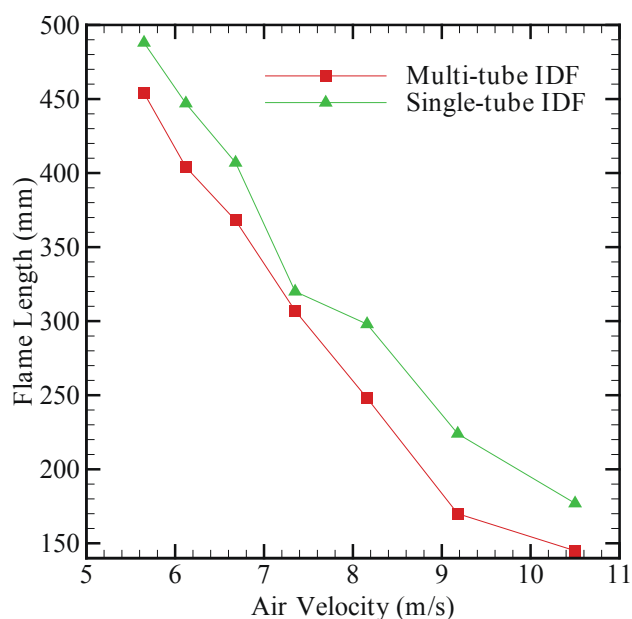


Figure 13. Flame length at different air velocities.

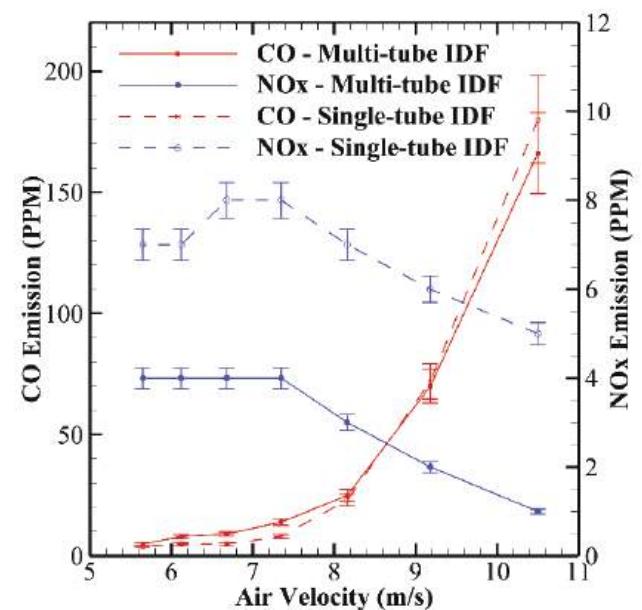


Figure 14. CO and NO_x emissions at different air velocities.

momentum of air, CO emission is higher for both single-tube and multi-tube IDF. At lower velocities, multi-tube IDF has slightly higher CO emissions. Figure 14 shows that at higher velocities, single-tube IDF has higher CO emission. The increase in CO emissions at higher air momentum is mainly because of the quenching effect [21,35]. In IDF, when a fuel impinges on a central air jet in the entertainment zone, part of the fuel cools down by the cold air jet, which produces the quenching effect. Also, shorter flame length in multi-tube offers shorter residence time for the oxidation of CO into CO_2 . Thus, the quenching effect and small reaction time are the reasons behind higher CO emissions in case of multi-tube IDF. NO_x emission in single-tube IDF is significantly higher than in multi-tube IDF. In velocity variation case, the single-tube IDF emits higher NO_x by 3 PPM compared to the multi-tube IDF. Low residence time due to shorter flame length and low flame temperature due to quenching favours low emissions of NO_x in multi-tube IDF.

Flame Temperature

Flame temperature is one of the parameters used to study flame characteristics. In present study flame temperatures are measured on a vertical plane passing through the centre of the central tube and the centre of a circumferentially

arranged tube. Figure 15, 16, and 17 show flame temperatures at different air velocities. At a low velocity of 5.65 m/s, the flame has a uniform temperature after a height of 60 mm. At low velocity, Figure 15 shows that the entrainment effect is reduced. Central and circumferential tube air will create a low temperature near the base of the IDF. Lower air velocity will provide less entrainment of the fuel towards center. At low velocity, less air is available for complete fuel burning. This will increase flame length and the amount of unburnt fuel burned in the post-combustion zone. A high temperature can be seen in the post-combustion zone.

At air velocity of 7.35 m/s, both high-temperature zones are merging, creating a single larger post combustion zone. The bright yellow region in Figure 16 is because of fuel accumulation in this region, which results in a yellow sooty region. Below 40 mm height, cold air jets try to cool down the main reaction zone. Air from the central tube has a greater quenching effect compared to circumferential air tubes. The distance between two annular regions shown by shade lines acts as the anchoring point of the flame. High temperature zone width is getting reduced as compared to 5.65 m/s case. Low temperature zone near base of the flame is physical mixing zone which is getting narrow. High temperature zone come closer to flame base due to increased

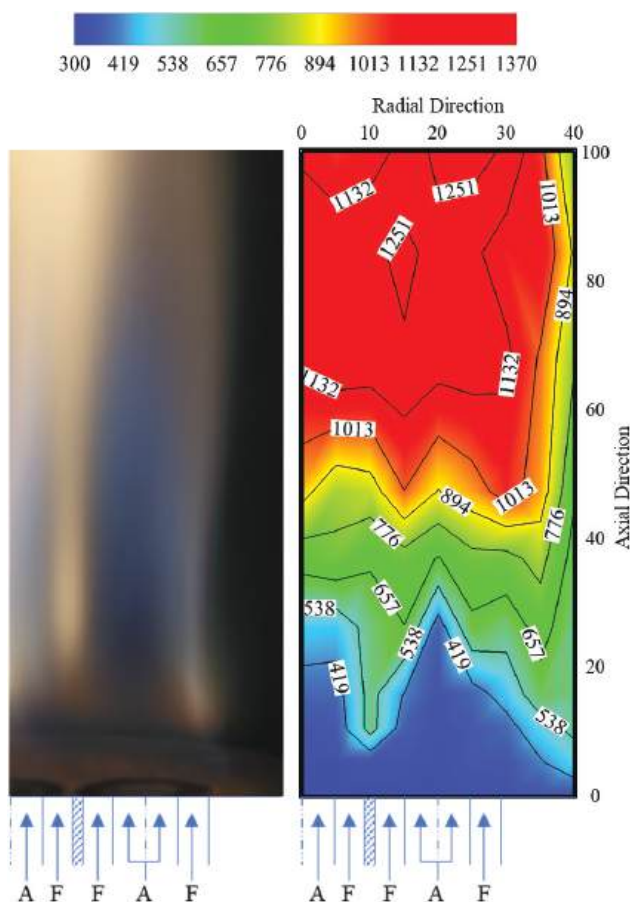


Figure 15. Temperature (K) contour at velocity 5.65 m/s.

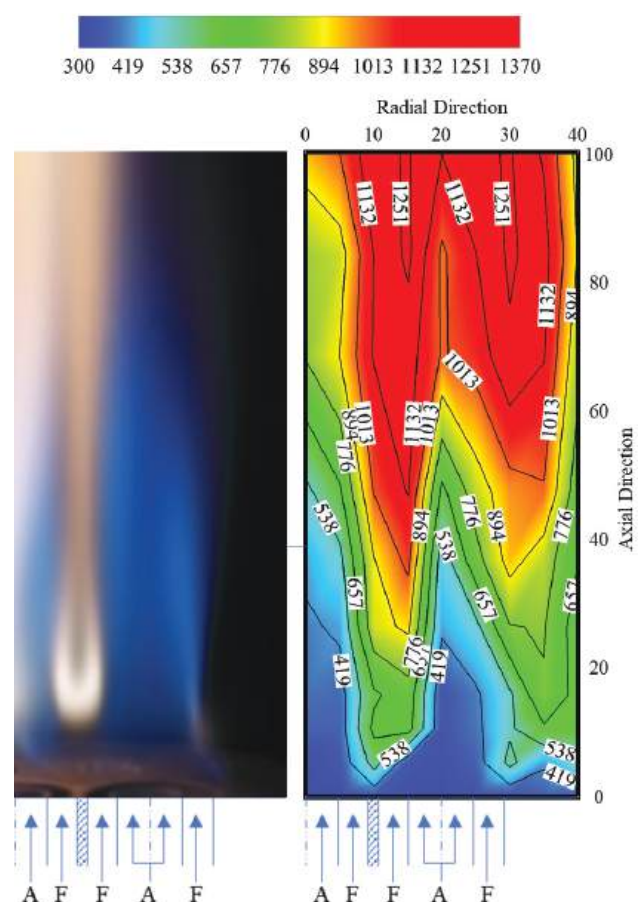


Figure 16. Temperature (K) contour at velocity 7.35 m/s.

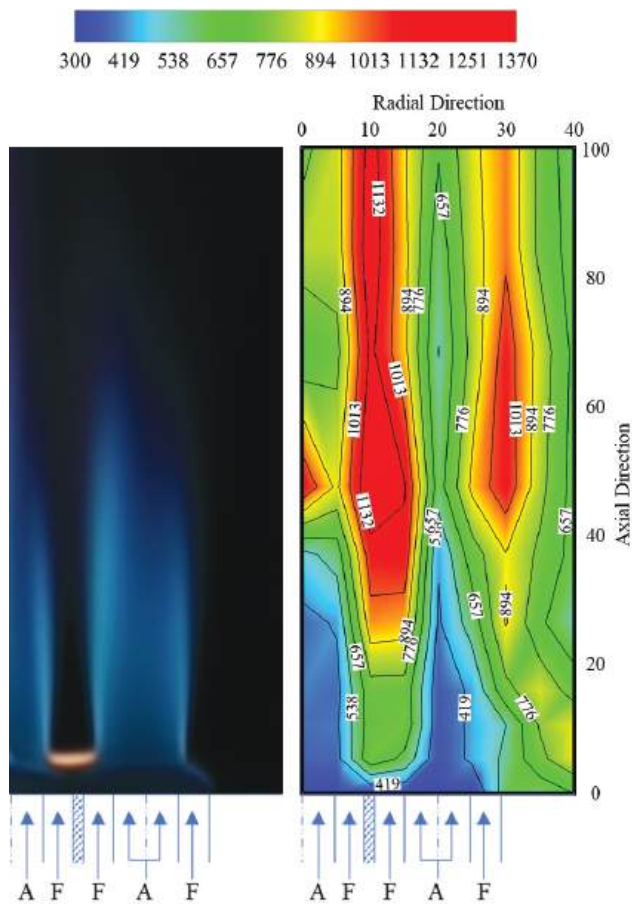


Figure 17. Temperature (K) contour at velocity 10.5 m/s.

air velocity. Increase in turbulence provides better mixing in the mixing zone cause the small low temperature zone at high velocity. At low velocity, yellow sooty region is not clearly visible. Yellow sooty region described in flame structure is clearly visible in 7.35 m/s case.

It is observed that at 10.5 m/s velocity, the flame is completely divided into two separate zones. As discussed in the previous section, a high-velocity air jet cools the fuel, and quenching will divide the flame into two zones. The entrainment zone at the base of the flame is clearly visible in Figure 17. This entrainment is mainly because of high velocity air tries to entrain fuel towards central air jet. At low velocity, air entrainment is less compared to high velocity.

As air jet velocity increases the width of high temperature zone is also getting reduced. At air velocity 10.5 m/s, narrow high temperature zone is observed compare to other cases. Increased entrainment at high air velocity helps to accelerate mixing in mixing zone so area of mixing zone also gets smaller. This accelerated fuel burning and shift beginning of fuel burning towards burner exit. Thus, high temperature zone begins almost from the burner base. Yellow sooty region is smaller at 10.5 m/s as compared to 7.35 m/s. This results in overall reduction of soot generation associated with high velocity air case.

CONCLUSION

The study compares the thermal and emission characteristics of single-tube and multi-tube inverse diffusion flames (IDFs) using LPG as a fuel. Equivalence ratios (ϕ) and air velocities are varied for parametric investigations. Major outcome from the current investigations are:

- Multi-tube IDF exhibits shorter flames, around 12.7% shorter than single-tube IDF. This is due to improved air-fuel mixing at larger contact surface area and higher turbulence produced by the smaller diameter air tubes. Flame length decreases with lower equivalence ratios and higher air velocities. A maximum reduction of 54 mm in flame length was observed at an air velocity of 9.18 m/s.
- Multi-tube IDF emits less CO emissions by 18 PPM at low equivalence ratios compared to single-tube IDF. This reduction is attributed to enhance mixing that prevents quenching effects caused by cold air jets.
- Multi-tube IDF reduces NO_x emissions by 4 PPM at higher equivalence ratios and 3 PPM across all other velocity variations compared to single-tube IDF. Multi-tube configuration reduces maximum flame temperature and also brings uniformity in distribution of high temperature in flame envelope. These helps in achieving low NO_x emissions.
- Multi-tube IDF produces shorter flame which emits low CO and NO_x. Increased surface area available due to multi-tube configuration enhances mixing and improves flame's thermal and emissions characteristics.

ABBREVIATIONS

ϕ	Equivalence ratio
d_a	Diameter of air-port
d_f	Diameter of fuel-port
V_{air}	Air jet velocity
CAP	Circumferentially Arranged Ports
CO	Carbon monoxide
CO ₂	Carbon dioxide
CoA	Coaxial or Co-annular
HC	Hydrocarbons
IDF	Inverse Diffusion Flame
MFR	Momentum Flux Ratio
N ₂	Nitrogen
NDF	Normal Diffusion Flame
NO _x	Nitrogen oxides
PPM	Parts Per Million
LPG	Liquefied Petroleum Gas
VFD	Variable Frequency Drive
SCADA	Supervisory Control and Data Acquisition
SLPM	Standard Liter Per Minute

CONFLICT OF INTEREST

The authors declare that they have no competing interests.

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AUTHORS CONTRIBUTION

Mayur Vadoliya develops lab setup, conducted experiments, drafted manuscript and arrange data. **Rupesh Shah** conceptualized research work, derive methodology and supervised overall study. **Nikhil Baraiya** co-supervised study, review and edited article.

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Research Article

Numerical investigation of sulfur trioxide decomposition in microchannel heat exchangers

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ABSTRACT

Due to global increasing in the energy demand and the need to reduce greenhouse gases, hydrogen is considered as an important energy carrier. In this work, numerical modeling of sulfur trioxide decomposition in the sulfur-iodine cycle in micro-channel heat exchangers has been considered. Different baffles shapes have been investigated. Results show by using these baffles in the flow path, the rate of decomposition increases. The highest decomposition percentage of 91.84% was obtained for the alternative design for the channel with asymmetric rectangular baffles. Also, the effect of the operating conditions such as the direction of inlet fluid flow, the inlet temperature of the helium channel and the type of reaction heat supply gas have been investigated. Results show that in the model with counter-current flow, the molar decomposition of sulfur trioxide is about 7% and the thermal performance coefficient is about 31% more than the parallel flow. Also, by increasing the inlet temperature of the helium flow, by about 18% compared to the base model, the molar sulfur trioxide decomposition increases by about 32%. But there is an about 21% reduction in the thermal performance coefficient. This study can provide a useful consideration of the effective parameters on the performance of a high temperature heat exchanger which worked as a sulfuric acid decomposer for hydrogen production within the sulfur-iodine thermochemical cycle.

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INTRODUCTION

With the development of human societies, the need of human beings for transportation and vehicles has increased, so consequently the consumption of fuel, especially fossil fuels such as gasoline and diesel, is increasing exponentially [1]. Hydrogen is one of the most essential commercial commodities and is known as an economical fuel source. In recent

years, the use of hydrogen gas as a fuel for vehicle power, heating, electricity generation, and some industrial affairs has been given priority [2]. The primary challenge for increasing hydrogen use is its cost of producing, storing and transporting, which in the following, we will focus on its production issue.

Hydrogen can be produced by various processes, such as from non-renewable sources or by water splitting [3].

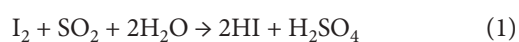
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The most common processes for producing hydrogen from non-renewable sources are the reforming process (especially methane reforming). However, this method produces pollution along with the production of hydrogen. Hydrogen production by water splitting is possible from two methods: water electrolysis and thermochemical cycle. Electrolysis requires low temperatures but is not economically justified due to its low hydrogen production and high cost of electricity consumption. Since 1980, more than 200 thermochemical cycles have been studied, the best of which is the sulfur-iodine thermochemical cycle, abbreviated as the sulfur-iodine cycle. In the sulfur-iodine cycle, all reactants and products are reproduced and recycled. This cycle requires heat and water at the inlet, and its outlet is pure oxygen and hydrogen. The three reactions exist in the sulfur-iodine cycle:



Equation (3) is the most important reaction of the sulfur-iodine cycle [4]. Sulfuric acid is one of the most important products of industrial units. The decomposition reaction of sulfuric acid includes two steps.



Equation (4) decomposes completely at temperatures above 450 °C and occurs very rapidly. Reaction (5), the decomposition reaction of sulfur trioxide, is a reaction with favorable kinetics. Decomposition of sulfur trioxide requires a high temperature, which can be provided by clean energy sources such as solar and atomic energy. Sulfur trioxide decomposes slowly at high temperatures, and a suitable catalyst must be used to accelerate the decomposition process [5]. However, this reaction (Equation (5)) can lead to lower temperatures inside the catalyst particle. So, some researchers have considered this viewpoint [6-7]. Pathak and Upadhyayula [6] studied the catalysts for this reaction. They studied the activity of various catalysts (noble, non-noble, complex metal, supported and unsupported metal oxide). They considered the effect of preparation methods, temperature, flow rate, inlet composition, and pressure.

In other works, simulation and optimization of sulfuric acid decomposition process had been investigated. Gao et al. [8] had experimentally and numerically studied this process for hydrogen production. They measured the kinetic factor of sulfuric acid decomposition reaction in experimental section. In the modelling section, the thermal-hydraulic characteristics and the parameters of

chemical reaction had been predicted. Their results showed good agreement. In another work, done by Sujesh et al. [9], modeling and optimization of packed bed reactor for sulfuric acid decomposition. They experimentally studied the effect of flow rate, wall temperature and with two catalyst loadings. A two-dimensional model was considered. In their model, fluid flow, heat transfer, and species transport with reaction in the porous domain were investigated. They found the suitable performance of this reaction in this reactor, which can be helpful for optimization study.

On the other hand, the overall efficiency of the process depends on the rate of decomposition of sulfur trioxide. One of the factors, which is affected on the efficiency of the reaction, is the residence time of the reactant in the reactor [10]. The design of sulfur trioxide decomposer is one of the main challenges in the development of hydrogen production. Microchannel heat exchangers have baffles with different arrangements in the fluid flow path, which affect the molar decomposition of sulfur trioxide by changing the residence time [11]. Because the sulfur trioxide decomposition is a surface reaction, the contact surface between the stream and the channel wall is an essential parameter in the decomposition rate [12].

Many researches had been studied on the sulfuric acid decomposers based on different issues such as the structures and efficiencies of hydrogen production. One of these studies is Ginosar's work [12]. They used three different catalysts. The influence of exposure on the reaction conditions was considered. Their results showed that the catalyst with higher surface area leads to the higher activity of catalyst, but their deactivation will be rapidly. Pt/TiO₂ catalyst had a lower surface area catalyst. In another work studied by Nagarajan et al. [13] a particular type of heat exchanger was studied. In their study, the temperature of wall is constant which one is measured experimentally. Different types of pellets in the packed bed region were used. The decomposition percentage of sulfur trioxide is considered. In the other their work [2], four different types of fins had been numerically simulated under three different arrangements and proposed a proper design scheme.

In the study done by Sun et al. [14], the design of the decomposer and its pipeline have been studied. Their design is based on a shell-and-tube heat exchanger with a bayonet heat exchanger as the core. They also investigated the heat transfer in their system. They were numerically simulated this system with accounting the phase change of sulfuric acid and the decomposition reaction. They found that the decomposition fraction of sulfuric acid decreases with the enhancement of the flow rate. Also, the sulfuric acid decomposition rate increases and then, with increasing the velocity, it will be stable [15]. Gao et al. [16] focused on the temperature profile, phase change, and chemical reactions of sulfuric acid in this system. The evaporation-condensation model was used to simulate the different stages of the phase change process.

In some other works, the effect of the geometry with different fins or grooves have been considered on the fluid flow and heat transfer, different shape such as: rectangular and semi cylindrical [17], wavy [18] and some other types [19–21]. In the study by Zhu et al. [19], microchannel heat sinks with grooves were studied. The effect of the channel geometry on the performance of this system was considered. Five different types of structures were designed. These different structures were compared, and the optimum structure was determined. Their results showed that the arrangement of the grooves on walls can be affected on performance. Wang et al. [20] found that the use of ribs will induce high-flow disturbances and a blocking-flow effect, so, heat transfer can be improved, although, the pressure drop increases. Their results show that the system with grooves can be more enhanced in heat transfer and smaller pressure drop related to the system with ribs. With consideration of related studies, it can be found that a comprehensive study in various conditions (different operational and geometrical conditions) is needed to better study of the performance of the system.

In this study, sulfur trioxide decomposition in the microchannel heat exchangers has been considered. the thermal performance coefficient, which studied the effect of friction factor and Nusselt number together, was considered in different conditions in this work. In the base model, the rectangular baffles were used. The location of these baffles (only on the bottom wall, asymmetric baffles on the upper and lower walls) were compared. At the next step, the effect of some operating parameters such as the direction of the fluid flow in microchannels, the temperature of each inlet streams, and some geometrical parameter (different baffle shapes) on the molar decomposition of sulfur trioxide had been investigated.

CFD MODELING

The modeling method is based on Computational Fluid Dynamics (CFD). CFD is the analysis of systems involving

fluid flow, heat transfer, and associated phenomena, such as chemical reactions, based on computer simulations. The structure of the CFD program is a numerical method, so fluid flow problems can be solved using this method [22]. CFD modeling and simulation is as a valuable tool for modeling of some processes. There are different studies done by CFD modeling [23–24].

Geometry of the system

Figure 1 shows the base model of a micro-channel heat exchanger. There are three sections in this microchannel heat exchanger: a helium passage channel, a reaction channel and the middle part between these channels made of silicon carbide to transfer heat from the hot helium channel to the reaction channel. The direction of the inlet flow in these channels is counter-current. The height of the helium and reaction channel is 0.85 mm and 0.424 mm, respectively. In this base model, 20 rectangular baffles were located on the walls. The height of the rectangular baffles is 0.2 mm and their width is 0.1 mm. Also, the distance between the baffles is 0.3 mm. The location of the baffles has been considered in this work. Figure 2 shows a schematic view of the baffle location.

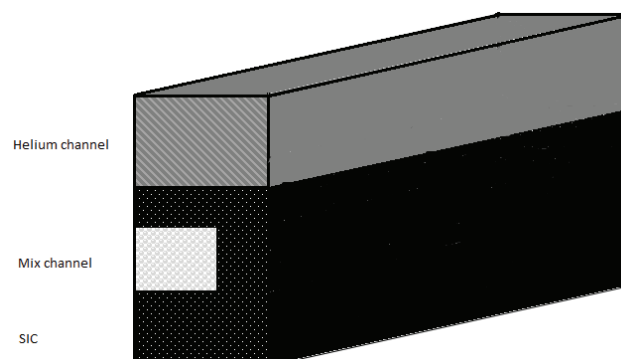
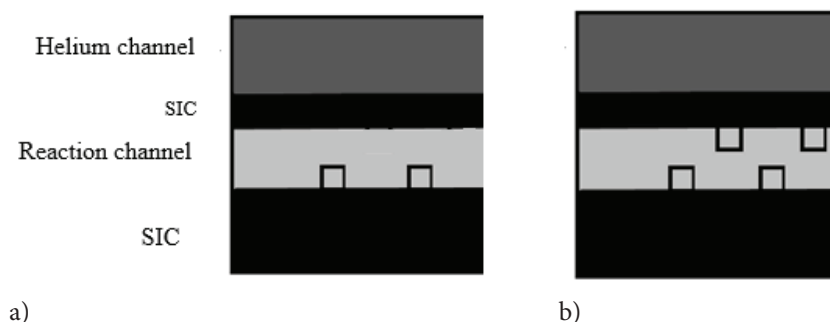


Figure 1. Schematic view of three sections of the micro-channel heat exchanger.



a)

b)

Figure 2. Location of baffles, (a) only on the bottom wall, (b) asymmetric baffles on the upper and lower walls.

Governing Equations

The mathematical model of steady state flow in a micro-channel heat exchanger is described using the Navier-Stokes three-dimensional flow equations, which include a set of partial differential equations (PDE) for the equations of continuity, momentum, energy, and species transfer which are expressed in Equations (6) to (11). The governing equations are as follows [25]:

Continuity equation:

$$\frac{\partial}{\partial x}(\rho v_x) + \frac{\partial}{\partial y}(\rho v_y) + \frac{\partial}{\partial z}(\rho v_z) = 0 \quad (6)$$

Momentum equation:

$$\rho(v_x \frac{\partial v_x}{\partial x} + v_y \frac{\partial v_x}{\partial y} + v_z \frac{\partial v_x}{\partial z}) = -\frac{\partial p}{\partial x} + \mu[\frac{\partial^2 v_x}{\partial x^2} + \frac{\partial^2 v_x}{\partial y^2} + \frac{\partial^2 v_x}{\partial z^2}] \quad (7)$$

$$\rho(v_x \frac{\partial v_y}{\partial x} + v_y \frac{\partial v_y}{\partial y} + v_z \frac{\partial v_y}{\partial z}) = -\frac{\partial p}{\partial y} + \mu[\frac{\partial^2 v_y}{\partial x^2} + \frac{\partial^2 v_y}{\partial y^2} + \frac{\partial^2 v_y}{\partial z^2}] \quad (8)$$

$$\rho(v_x \frac{\partial v_z}{\partial x} + v_y \frac{\partial v_z}{\partial y} + v_z \frac{\partial v_z}{\partial z}) = -\frac{\partial p}{\partial z} + \mu[\frac{\partial^2 v_z}{\partial x^2} + \frac{\partial^2 v_z}{\partial y^2} + \frac{\partial^2 v_z}{\partial z^2}] \quad (9)$$

Energy equation:

$$\rho \hat{C}_P \left(v_x \frac{\partial T}{\partial x} + v_y \frac{\partial T}{\partial y} + v_z \frac{\partial T}{\partial z} \right) = k \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right] + S_h \quad (10)$$

Species transfer equation:

$$\rho \left(v_x \frac{\partial Y_A}{\partial x} + v_y \frac{\partial Y_A}{\partial y} + v_z \frac{\partial Y_A}{\partial z} \right) = \rho D_{AB} \left[\frac{\partial^2 Y_A}{\partial x^2} + \frac{\partial^2 Y_A}{\partial y^2} + \frac{\partial^2 Y_A}{\partial z^2} \right] + S_c \quad (11)$$

In above equations, ρ , μ , k , C_p , and D_{AB} are mixture density, viscosity, thermal conductivity, specific heat at constant pressure, and diffusion coefficient, respectively. The density was calculated using the ideal gas law for an incompressible flow. Other parameters were calculated by equations (12-15) based on kinetic Theory. The diffusivity of the mixture (D_{AB}) was calculated using the Chapman-Enskog kinetic theory. S_h is the heat of the reaction and S_c is the net production rate by chemical reaction.

In the helium channel, the Continuity, momentum and energy equations (Equations 6-10) are solved. There is no reaction in this channel. The term heat of reaction (S_h) in equation (10) is zero due to the lack of reaction in the helium channel. In the reaction channel, in addition to those equations (Equations 6-10), the species transfer equation (Equation 11) is also established. In the middle part, only the energy equation (Equation 10) is solved. In this part, due to the lack of the reaction, the heat term of reaction (S_h) in Equation (10) is zero.

$$\mu = \sum_i \frac{x_i \mu_i}{\sum_j x_j \phi_{ij}} \quad (12)$$

$$\phi_{ij} = \frac{\left[1 + \left(\frac{\mu_i}{\mu_j} \right)^{\frac{1}{2}} \left(\frac{Mw_j}{Mw_i} \right)^{\frac{1}{4}} \right]^2}{\left[8 \left(1 + \frac{Mw_j}{Mw_i} \right) \right]^{\frac{1}{2}}} \quad (13)$$

$$k = \sum_i \frac{x_i k_i}{\sum_j x_j \phi_{ij}} \quad (14)$$

$$C_p = \sum_i Y_i C_{pi} \quad (15)$$

The wall surface reaction model is done on the wall of the reaction channel and is implemented to determine the mole fractions of SO_3 , SO_2 and O_2 . The reaction rate was obtained by using the Arrhenius equation and the reaction is first order reaction (Equations 16 -17) [26].

$$K = A e^{\left(\frac{-E_a}{RT} \right)} \quad (16)$$

$$Rrxn = -KCso3 \quad (17)$$

Where E_a is the activation energy (equals to 3.267×10^7 J/kmol) and A is pre-constant reaction (equals to 0.05).

Solution Method and Boundary Conditions

In this work, the finite volume method is used to discretize the governing equations by the SIMPLE algorithm. Discretization of energy, momentum and pressure equations is performed by first-order upwind, first-order upwind and standard methods, respectively. Convergence is considered when the residuals of the values reach 10^{-6} . A combined mesh of rectangular and triangular mesh was used in these simulations. Mesh dependency was considered, too. The effect of mesh number on the sulfur trioxide decomposition was investigated (as sample, Figure 3 shows for two systems, without baffles and with 20 baffles). Finally, the proper mesh, which had little effect on the results was selected for each system.

In the base model, a hot helium gas stream with a temperature of 1223.15 °K enters the helium channel. The reactants mixture with a temperature of 974.9 °K enters the reaction channel in the opposite direction. Boundary conditions at the entrance are based on specific mass flow rate and specific inlet temperature. The no-slip condition is established on the walls. The right, upper, and lower walls of the geometry are adiabatic. The left wall is considered a symmetrical boundary. The reaction takes place in the reaction channel and on the wall surface. The reaction takes place under a specific pressure (1.5 MPa). In this research,

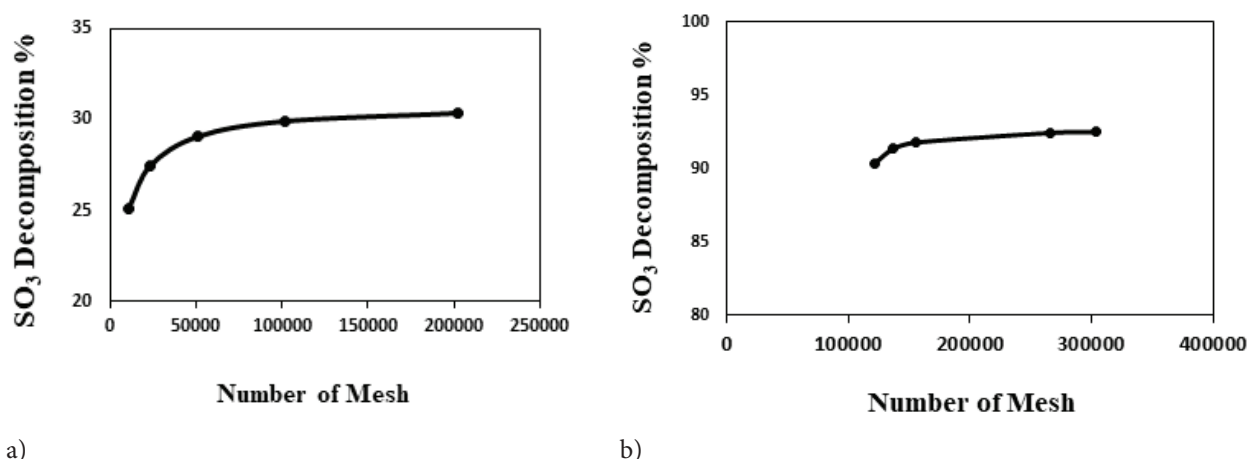


Figure 3. Mesh independency in system (a) without baffles, (b) with 20 baffles.

four parameters (Nusselt number, thermal performance coefficient, molar percentage of sulfur trioxide decomposition, and pressure drop) have been investigated for all models. The thermal performance coefficient is defined as Equation (18):

$$\eta = \frac{\left(\frac{Nu}{Nu_0}\right)}{\left(\frac{f}{f_0}\right)^{\frac{1}{3}}} \quad (18)$$

Nu_0 and f_0 are the Nusselt number and the friction coefficient in a channel without baffles, respectively. It is clear that the value of thermal performance coefficient equals to one for the model without baffles. Also, when the value of the thermal performance coefficient is greater than one, it means that the system has better performance. Equation (19) is used to calculate the molar percentage of sulfur trioxide decomposition:

$$SO_3 \text{ Decomposition \%} = \left[\frac{(SO_3 \text{ concentration})_{inlet} - (SO_3 \text{ concentration})_{outlet}}{(SO_3 \text{ concentration})_{inlet}} \right] * 100 \quad (19)$$

RESULTS AND DISCUSSION

Validation

To validate the model, a study by Ponyavin [27] has been reviewed. In this work, rectangular channels with dimensions similar to Ponyavin's work have been used. A stream of hot helium gas enters the helium channel at 1223.15 °K and the reactants mixture enters the reaction channel at 974.9 °K in the opposite direction. The comparison of the results of Ponyavin's work and this simulation results show that the average error is 4%. The highest error equals to 7%, is observed at the flow rate $3.148 * 10^{-6}$ Kg/s. So, it can be said there is a good agreement between the results and the present model is acceptable.

Effect of the baffles

In this section, 20 rectangular baffles were used in the reaction channel asymmetrically on the upper and lower walls of the channel. Figures 4 shows the variations of molar decomposition of sulfur trioxide and the thermal performance coefficient versus inlet mass flow rate for this model and the model without baffles. The black line corresponding to the model with 20 baffles asymmetrically and the dash line represents the model without the baffles. Other operating conditions are the same as the base model. Results show SO₃ decomposition decreases with increasing the mass flow rate for both models. Also, the model with 20 baffles asymmetrically has higher decomposition rates than the model without baffles. By using these baffles in the flow path, the area of the reaction surface increases and, due to the superficial reaction of sulfur trioxide decomposition, its molar decomposition rate increases by about 12% compared to the model without baffles at the same flow rate.

Moreover, from Figure 4 (b), it can be found that the value of the thermal performance coefficient of the model with 20 baffles asymmetrically is greater than one. So, it is desirable. With increasing the mass flow rate, initially the thermal performance coefficient increases, but with more mass flow rates, its value decreases. Corresponding to Equation (16), the thermal performance coefficient is related to the ratio of Nusselt number to friction factor (which can be calculated from pressure drop) for both models.

Figure 5 shows the variations of the ratio of Nusselt number and friction factor versus mass flow rates. As shown in Figure 5 (a), the Nusselt number in the model with 20 baffles asymmetrically is greater than the model without baffles. The Nusselt number has a direct relationship with the surface convection heat transfer coefficient. Due to the higher reaction surface of the model with 20 baffles asymmetrically, the surface convection heat transfer coefficient increases and, so, the Nusselt number increases for this model respect to the model without baffles. However,

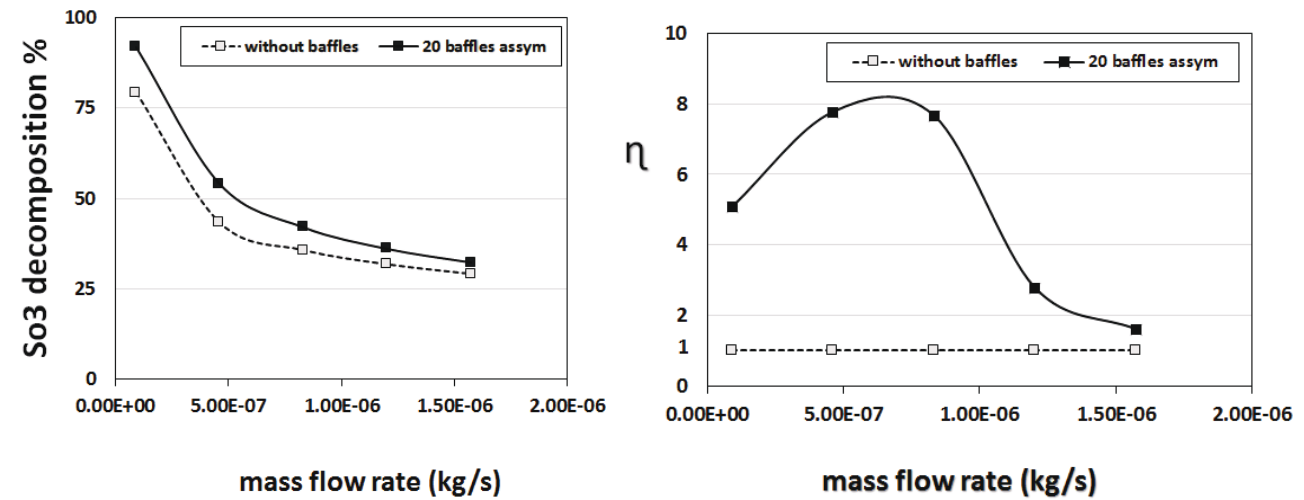


Figure 4. Comparison of (a) molar decomposition of sulfur trioxide and, (b) the thermal performance coefficient versus mass flow rates for the model with 20 asymmetric rectangular baffles and the model without baffles.

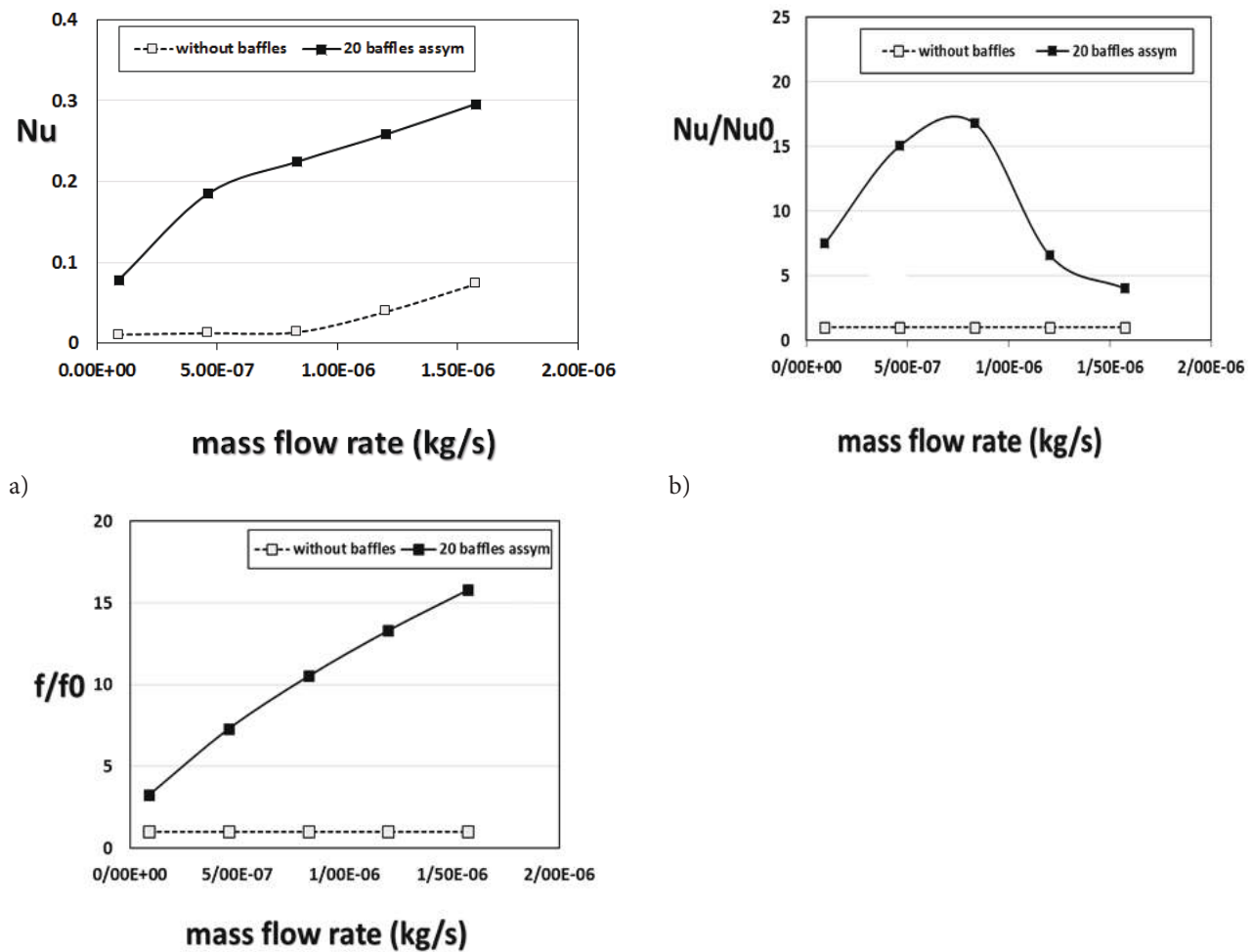


Figure 5. Comparison of (a) the Nusselt number, (b) Nu/Nu₀, (c) f/f₀ versus mass flow rates for the model with 20 asymmetric rectangular baffles and the model without baffles.

the ratio of the Nusselt number of these models is changed with increasing the mass flow rates, so, it can be found from Figure 5 (b), the ratio of the Nusselt number of these models (Nu/Nu_0) initially increases and then decrease. Also, by adding the baffles, the friction factor increases due to increasing the pressure drop (Figure 5 (c)).

Effect of operating conditions

After the above consideration, it was found that the model with 20 asymmetric rectangular baffles had a better performance than the model without baffles. Therefore, this model was examined as the base model. In this base model, inlet flow rate equals to 1.574×10^{-6} (Kg/s) and mass fraction of sulfur trioxide and water equal to 0.8163 and 0.1837, respectively. The inlet temperatures of the reaction and helium channel are 974.9 K and 1223.15 K, respectively. The operating pressure is 1.5 MPa. In this section, the effect of the operating conditions such as the direction of inlet fluid flow, the inlet temperature of the helium channel and the type of reaction heat supply gas is investigated. In all sections, only that parameter is changed and other parameters will be constant (equals to their values at the base model).

Figures 6 shows the variations of the molar decomposition of sulfur trioxide and thermal performance coefficient with different flow direction. When the inlet flow of reaction and helium channels enters from the same side of the microchannel heat exchanger, it was named as parallel. When inlet flow of reaction and helium channels enters in the opposite direction, it was named as counter-current flow. Other parameters are the same as the base model and only the direction of the inlet flow was changed. The black bar chart shows the parallel flow and the gray bar chart shows the counter-current flow. From Figure 6, it can be

found that the molar sulfur trioxide decomposition, and the thermal performance coefficient with counter-current flow are higher than the parallel flow. It is consistency with the previous study [4]. In the model with counter-current flow, the molar sulfur trioxide decomposition is about 7% and the thermal performance coefficient is about 31% more than the parallel flow. For non-aligned motion, the surface convection heat transfer coefficient is a higher than in the aligned position. In fact, in the counter-current flow, the average temperature difference along each part of the channel is greater than the parallel flow. It therefore has a better thermal performance coefficient than other flows. Due to the direct relationship between the Nusselt number and the surface heat transfer coefficient, the higher value of the surface convection heat transfer coefficient leads to increasing the Nusselt number, which effects on the sulfur trioxide decomposition due to the superficiality of the decomposition reaction.

Figure 7 shows the variations of the molar sulfur trioxide decomposition and thermal performance coefficient versus the inlet temperature of the helium flow for two different mass flow rates (mass flow rate 1 > mass flow rate 2). Other parameters are similar to the base model. Results show that by increasing the inlet temperature of the helium flow by about 18% compared to the base model, the molar sulfur trioxide decomposition increases by about 32%. However, there is an about 21% reduction in the thermal performance coefficient. As the inlet temperature of the helium stream in the helium channel increases, the rate of heat transfer to the surface of the reaction channel increases. As a result, fluid contacts with the hot wall of the reaction channel, the molar sulfur trioxide decomposition increases. Also, as shown in Figure 7(b) the thermal performance coefficient decreases with increasing the inlet temperature of the helium flow.

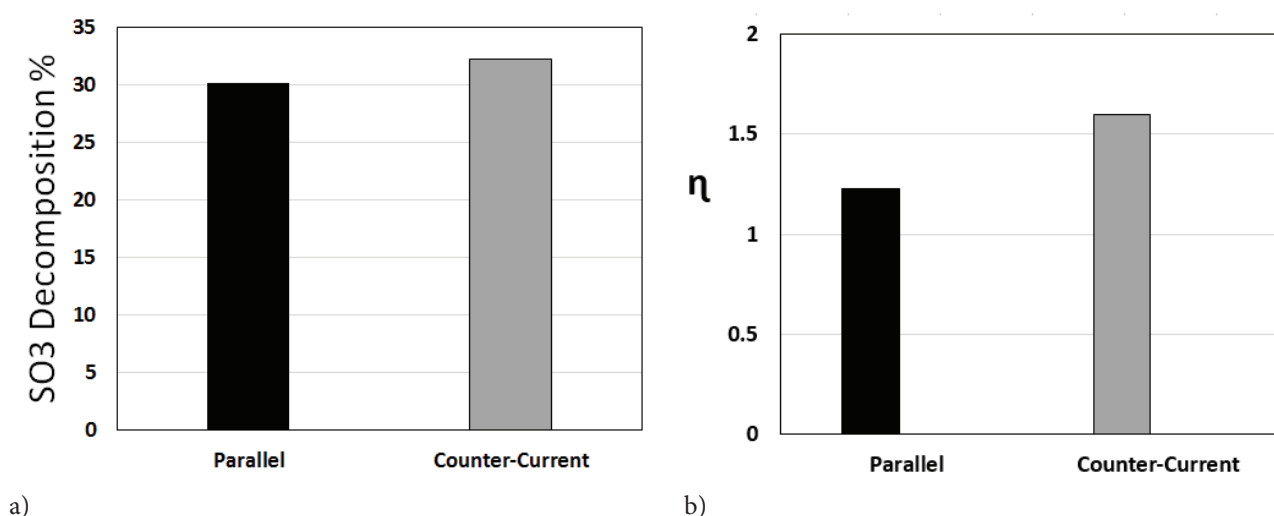


Figure 6. Comparison of (a) the molar decomposition of sulfur trioxide and (b) the thermal performance coefficient of parallel and counter-current flows in the model with 20 asymmetric rectangular baffles, $P = 1.5$ MPa, $T_{\text{mix}} = 974.9$ K, $T_{\text{he}} = 1223.15$ K, mass flow rate = 1.574×10^{-6} kg/s.

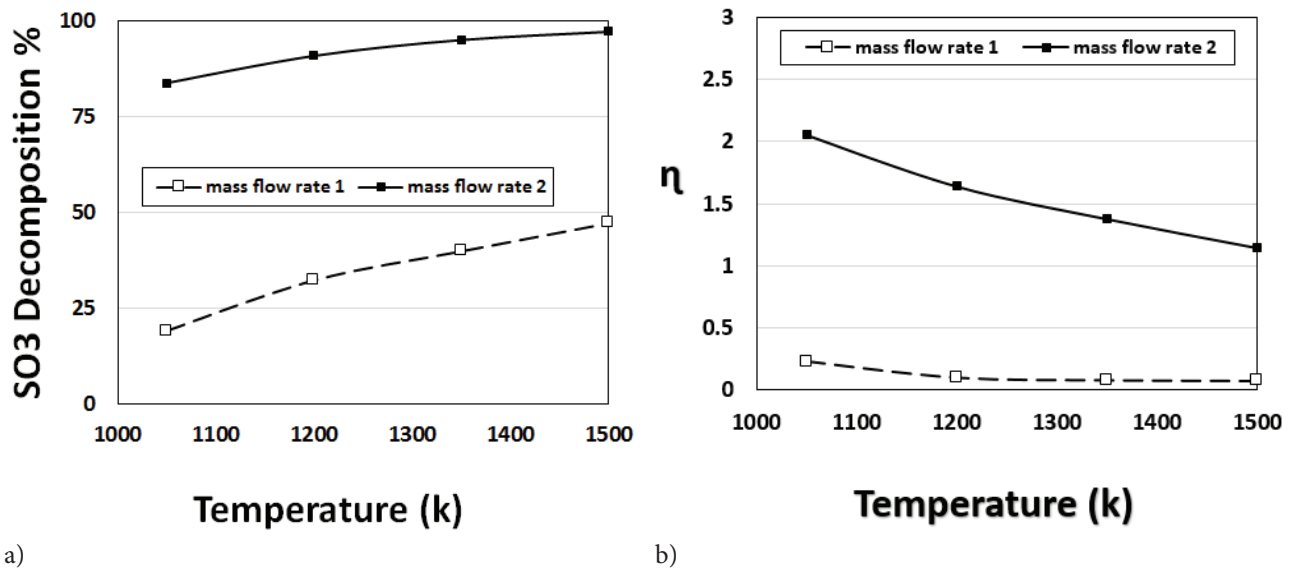


Figure 7. Variations of (a) molar decomposition of sulfur trioxide, (b) the thermal performance coefficient in the model with 20 asymmetric baffles at different helium channel inlet temperatures, $P = 1.5$ MPa, $T_{\text{mix}} = 974.9$ K, mass flow rate 1 = 1.574×10^{-6} kg/s, mass flow rate 2 = 9×10^{-8} kg/s.

With increasing the inlet temperature of the helium flow, the Nusselt number decreases. So, the thermal performance coefficient decreases.

In the base model, the gas used to supply the heat required for the reaction is helium hot gas. In this section, the effect of using hot argon and air gases is investigated and other condition are the same as the base model. Figure 8 shows the molar decomposition of sulfur trioxide and the thermal performance coefficient with these hot gases. From Figure 8, it can be found that the use of helium gas is much better than the other gases mentioned. Because the specific heat capacity of helium gas is more than two other gases, so,

it can supply more heat than other mentioned gases. As a result, the sulfur trioxide decomposition increases.

Effect of Baffles shape

In this section, the effect of using different baffles shapes such as triangle, semi-ellipse, and trapezoid is investigated. These baffles have equal height and width. The geometric schematic view of these shapes of baffles is shown in Figure 9 (a-c). In this section, based on above considerations, the counter-current flow with inlet mass flow rate of 9×10^{-8} (Kg/s) and the inlet temperature of helium channel of 1350 °K is accounted. The reaction was done on the baffle walls.

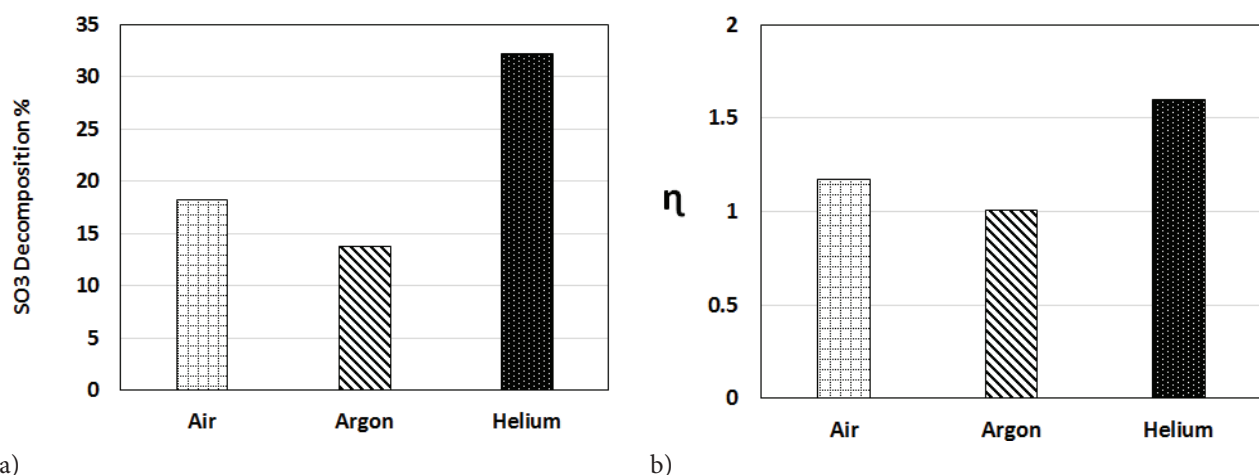


Figure 8. Variations of (a) the molar decomposition of sulfur trioxide, (b) the thermal performance coefficient in the model with 20 asymmetric baffles with different gases supplying the reaction heat, $P = 1.5$ MPa, $T_{\text{in,gas}} = 1223.15$ K, $T_{\text{mix}} = 974.9$ K, mass flow rate = 1.574×10^{-6} kg/s.

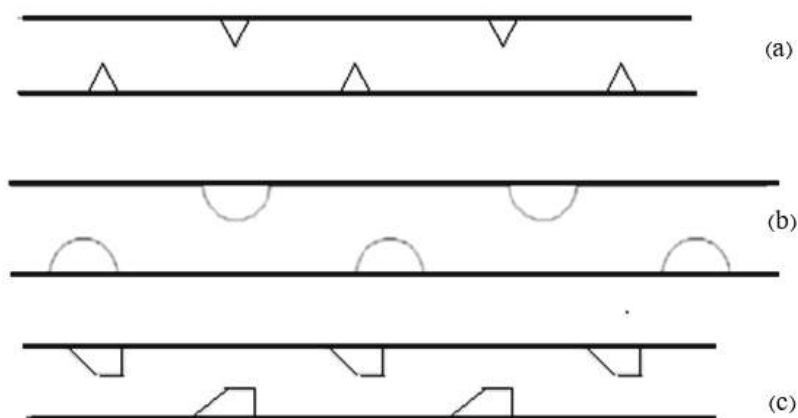


Figure 9. Geometric schematic view of (a) triangular, (b) semi-ellipse, (c) trapezoidal baffles.

Table 1. Reaction surface

Baffle shape	Area (mm ²)
Rectangular baffles	24.8686
Triangular baffles	23.44102
Trapezoidal baffles	24.1548
Semi-ellipsoid baffles	19.7546

With changing the shape of the baffles, the reaction surface was changed. Table 1 lists the reaction surface for different shapes of baffles. Due to the superficiality of the reaction and its effect on the rate of molar decomposition, the reaction area of baffles can be affected.

Figure 10 shows the molar decomposition of sulfur trioxide for the models with different baffle shapes. Due to the proximity of the reaction surface of the geometry of the baffles used in the fluid flow path, different molar decompositions are observed. The molar sulfur trioxide decomposition for the model with rectangular baffles is slightly higher than other models. The molar of sulfur trioxide decomposition for the model with rectangular baffles is about 1.5% higher than the triangular model, about 0.3% higher than that of the trapezoidal model and about 8.5% higher than that of the semi-ellipse model. The critical point is the effect of the structure of these baffles on parameters such as pressure drop and Nusselt number. The type of curvature, the sharp edge of the baffles and the slope of the sides affect the amount of friction, pressure drop and Nusselt number.

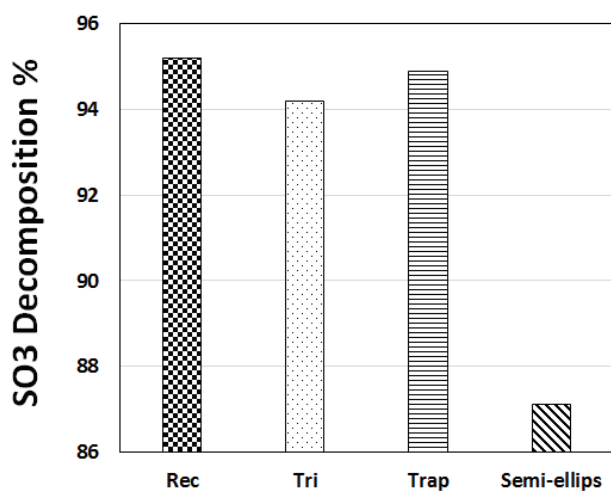


Figure 10. Comparison of the molar decomposition of sulfur trioxide in the model with 20 asymmetric baffles with different baffles shapes, mass flow rate = 9×10^{-8} Kg/s, $P = 1.5$ MPa, $T_{\text{mix}} = 974.9$ K, $T_{\text{he}} = 1350$ K.

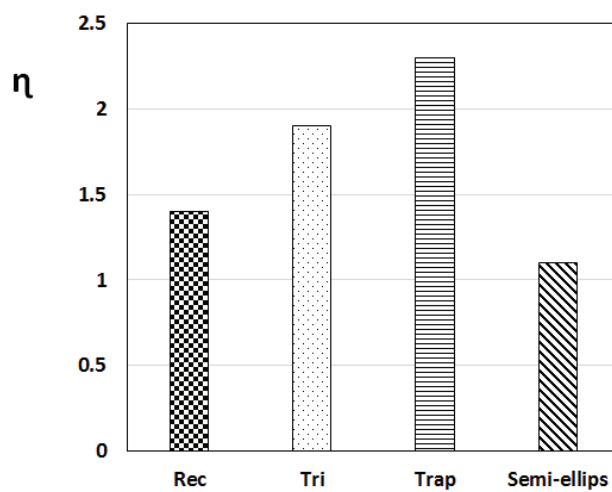


Figure 11. Comparison of thermal performance coefficients in the model with 20 asymmetric baffles with different baffles shapes, mass flow rate = 9×10^{-8} Kg/s, $P = 1.5$ MPa, $T_{\text{mix}} = 974.9$ K, $T_{\text{he}} = 1350$ K.

Figure 11 shows the thermal performance coefficient of the models with different baffles. The lowest thermal efficiency is related to the semi-ellipse model. Due to the curvature of the semi-ellipse baffle heads, after each baffle, the low number of vortices formed reduces the rate of heat transfer from the surface of the channel to the fluid compared to other models. Also, the thermal performance coefficient of the trapezoidal model is higher than other models due to its better Nusselt number and lower pressure drop. The thermal performance coefficient of the model with triangular baffles is about 29% higher than the model with rectangular baffles, while, the model with trapezoidal baffles is about 41% higher than the model with rectangular baffles. In the model with semi-ellipse baffles, the thermal performance coefficient is about 25% lower than the model with rectangular baffles.

CONCLUSION

In this work, three-dimensional computational fluid dynamics analysis was performed on fluid flow, heat transfer and chemical reaction in a high temperature heat exchanger. Due to the superficiality of the decomposition reaction of sulfur trioxide, the surface area of the reaction was increased to improve its decomposition rate. This was done by adding rectangular baffles to the reaction channel. From this consideration, it can be found:

- With the addition of asymmetric rectangular baffles, about 30% was added to the reaction surface area compared to the no-baffle model.
- The molar decomposition of sulfur trioxide increases by about 12% compared to the no-baffle model.
- The conversion rate of sulfur trioxide in the system with counter-current flow was about 7% and the coefficient of thermal performance was about 31% higher than the parallel flow.
- By increasing the helium flow inlet temperature by about 18% compared to the base model, the sulfur trioxide conversion rate increases by about 32% but the coefficient of thermal performance decreases by about 21%.
- The use of argon and air gases instead of helium significantly reduces the conversion rate and thermal efficiency.
- In the model with 20 asymmetric triangular baffles, compared to the base model, the sulfur trioxide molar decomposition decreases by about 1.5%, but the coefficient of thermal performance increases by about 40%. While, in the system with 20 asymmetric semi-ellipse baffles, the molar decomposition of the system with semi-ellipse baffles reduces by about 7.5%. Also, its thermal performance coefficient is lower than the base model.

NOMENCLATURE

A	pre-exponential factor, s^{-1}
C	molar concentration, mol/m^3
Cp	specific heat at constant pressure, $J.kg^{-1}.K^{-1}$

D_{AB}	diffusion coefficient for species A in the B, $m^2.S^{-1}$
E_a	activation energy, $J/kmol$
f	friction factor
K	reaction rate, s^{-1}
k	thermal conductivity, $W.m^{-1}.K^{-1}$
m	mass flow rate, $kg.s^{-1}$
Mw	Molecular weight, $kg/kgmol$
Nu	Nusselt number
R_{rxn}	net rate of production of species i by chemical reaction, $kg.m^{-3}.s^{-1}$
S_c	net rate of production of species A by chemical reaction, $kg.m^{-3}.s^{-1}$
S_h	heat of chemical reaction, $J.mol^{-1}$
T	static temperature, K
x_i	mole fraction of species i
Yi	mass fraction of species i

Greek symbol

η	thermal performance coefficient
μ	Viscosity, $kgm^{-1}.s^{-1}$
ρ	density, $kg.m^{-3}$
Φ_{ij}	constant of ideal gas law

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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

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Research Article

Numerical analysis of thermo-hydraulic performance of molten salt coolant flowing through a louvered-fin compact heat exchanger

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ABSTRACT

New types of nuclear reactors have been proposed in recent years to meet the growing energy demand. These reactors operate at high outlet temperatures, which must be efficiently removed from the system. For this purpose, compact heat exchangers are considered important candidates. The study examines the thermohydraulic behavior of LiF-BeF₂ (FLiBe) molten salt in a compact heat exchanger with multi-louvered fins as a part of a Molten Salt Reactor's cooling loop at the steady state. This numerical benchmark study aimed to collect data for both baseline scenarios and cases of slow coolant circulation within the reactor for the first time in the literature. Numerical tests were conducted for low Reynolds numbers ranging from 100 to 500, over a flow region characterized by varying fin pitches and louver angles to compare FLiBe's performance as a coolant in a Molten Salt Reactor. The findings indicate that as the louver angle increases, the molten salt flow becomes more directed through the louvers, enhancing heat transfer between the fins and the molten salt. Conversely, increasing fin pitches directs the flow more towards the ducts, reducing interaction with the FLiBe and diminishing thermal performance. As regard to outcomes, the highest heat transfer coefficient is determined to be about 27 kW/(m²K) at the smallest fin pitch of 1.5 mm, the biggest louver angle of 36° and a molten salt inlet velocity of 0.8 m/s. Under the same conditions, the heat transfer coefficient is calculated as 24.7 kW/(m²K) and 24.2 kW/(m²K) for the pitches of 2.0 mm and 2.5 mm, respectively. The louver angle has a significant effect on the pressure drop especially for the fin pitch of 1.50 mm. At 0.8 m/s molten salt inlet velocity and fin pitch of 1.5 mm, the pressure drop with a louver angle of 36° is 58% higher than with a louver angle of 20°.

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INTRODUCTION

In light of nuclear mishaps, public dissent, and the critical issue of climate change due to hazardous carbon emissions, the Next Generation Nuclear Plant (NGNP) initiative and the Fourth Generation Nuclear Reactor Systems (Gen IV) have identified Molten Salt Reactors (MSRs) as one of six promising candidates for reliable and sustainable nuclear power generation with minimal carbon emissions in the future. Molten salt nuclear reactors have been a subject of interest in nuclear engineering since 1940s, beginning with initial studies in Tennessee [1]. MSRs have been applied in various fields, including Aircraft Nuclear Propulsion (ANP), with successful experiments such as the aircraft reactor experiment (ARE) and aircraft reactor test (ART). Building on research from Oak Ridge National Laboratory (ORNL), small MSRs, known as Fuji, were developed by Furukawa et al. [2-5] within the Thorium Molten Salt Nuclear Energy Synergetic (THORIMS-NES) system. FLiBe, a fluid that serves both as fuel carrier and coolant, is considered one of the most suitable for use in Fuji-MSRs, facilitating heat exchange between the reactor core and the steam turbine side. Recent studies by emerging MSR firms have presented various MSR designs, yet they often lack detailed information on heat exchangers. Designing heat exchangers for MSRs poses significant challenges. Historically, ORNL has conducted extensive research on MSR heat exchanger design, particularly focusing on shell-and-tube types, and has established key design considerations well-documented in the literature [6-9]. These designs were shaped by the prevailing technology and standards applicable to shell-and-tube heat exchangers. In a novel approach, Erbay [10] explored the integration of MSRs with a Stirling heat engine, highlighting the critical need for high-performance heat exchangers to function as the hot-end head of the Stirling engine, with molten salt FLiBe flowing over and helium within the channels. In another original approach, Akram et al. [11] investigated a solar-assisted trigeneration system with molten salts from exergy aspect. Erbay's expertise [12] also shed light on the heat removal system of Fuji-MSRs [3]. Although new heat exchanger designs have been proposed in projects such as EVOL and SAMOFAR [13-15], these designs are yet to be finalized and manufactured.

Liquid coolants exhibit superior performance in gas cooling systems by minimizing temperature gradients across heat exchangers and reducing pumping power. Molten salts serve as both fuel and coolant in MSRs, chosen for their thermal efficiency and exceptional properties [16-18]. These coolants maintain chemical and radiological stability at elevated temperatures, exhibit low vapor pressures, and possess high boiling points, specific heat, and thermal conductivity. Their compatibility with high-temperature materials and cost-effectiveness further enhances their appeal. Notably, the volumetric heat capacity of molten salt coolants is nearly 25% greater than that of pressurized

water and about five times that of liquid sodium, facilitating the design of compact heat exchangers due to their high heat capacity and safe operational pressures. However, molten salt experiments present several challenges. The high melting point, while beneficial, complicates convection heat transfer experiments as it requires heating the entire system, including the cooling equipment. Additionally, corrosion of pipes adds further complexity. Consequently, computational fluid dynamics (CFD) studies have become a vital tool for assessing the thermal-hydraulic properties of molten salts. Recent experimental [19-23] and numerical [1, 24-28] research has focused on elucidating the convective heat transfer characteristics of molten salts and verifying the accuracy of established empirical correlations.

To address the individual challenges of both numerical and experimental investigations on this unique topic, researchers study shell-and-tube heat exchangers for molten salt refrigerants using a comparative approach, either numerically or experimentally [29, 30]. These studies have examined various molten salts, including LiNO_3 , LiF-NaF-KF (FLiNaK), $\text{NaNO}_2\text{-KNO}_3\text{-NaNO}_3$, $\text{LiF-BeF}_2\text{-ThF}_4\text{-UF}_6$, $\text{NaBF}_4\text{-NaF}$, and LiF-BeF_2 (FLiBe). The convective heat transfer with molten salt in different types of channel geometries; namely circular tubes, annular ducts, outside the tube bundles, and double pipes were conducted in these useful studies. The heat transfer enhancement in MSRs is important for reducing cost and fluid inventory. The amount of fuel out of the core in the MSRs should be adjusted to prevent a new critical mass. The enhancement of heat transfer and the need for a small volume to avoid hold-up issues in MSRs make compact heat exchanger configurations suitable. The Printed Circuit Heat Exchanger (PCHE), a novel design, offers compactness, low-pressure drop, high effectiveness, and the capacity to handle significant pressure differentials, and is produced by Heatric. [31, 32]. A compact heat exchanger (CHE) featuring finned surfaces tailored for a suitable working fluid is poised to become crucial in Molten Salt Reactors (MSRs). Various molten salts have undergone thorough analysis by Oak Ridge National Laboratory (ORNL) to aid the MSR Experiment and the Molten Salt Breeder Reactor Programs. [31]. Due to its convenience for both MSRs fuel salt and coolant, FLiBe has been reported as a promising candidate during the fluid selection process. Therefore, the thermal and hydraulic characteristics of FLiBe flowing over the finned surface must be investigated. Regrettably, there isn't any work that reports neither experimental nor numerical findings for a louvered fin flat-tube compact heat exchanger associated with MSRs in open sources. Computational Fluid Dynamics (CFD) can be used to fulfill the lack of data and information in the field.

This research examines a louvered fin flat-tube compact heat exchanger designed to achieve high heat transfer rates and extensive heat transfer areas. FLiBe (LiF-BeF_2), commonly used in MSRs and Molten Salt Fast Reactors (MSFRs), serve as the working fluid. The study focuses on the

steady-state thermo-hydraulic performance of FLiBe in the cooling loop of an MSR, particularly within a low-velocity range, which could represent either a primary or secondary heat exchanger. Numerical simulations were performed at low Reynolds numbers ranging from 100 to 500 to emulate scenarios like coolant pump failures. Various louver pitches and fin pitches and fin louver angles were tested. The resulting velocity profiles, temperature, and pressure fields were analyzed. Performance metrics for FLiBe as a coolant were quantified using the heat transfer coefficient, pressure drop data, Colburn j -factor, fanning friction factor, and the volume goodness factor. Despite concerns regarding the use of molten salts in nuclear reactors, this study aims to enrich the MSR literature by exploring alternative heat exchanger configurations. The paper begins by detailing the geometry of the louvered finned flow channel, followed by a description of the Computational Fluid Dynamics (CFD) model used to simulate FLiBe convection, and concludes with the presentation of validated results.

MATERIALS AND METHODS

The technical drawing of the louvered fin geometry under consideration is depicted in Fig. 1. The material used for the fin is Hastelloy-N, a nickel-based alloy developed by ORNL and the International Nickel Company for use in MSRs. The louvered fin's performance was evaluated for fin pitches of 1.5, 2.0, and 2.5 mm and louver angles of 20°, 24°, 28°, 32°, and 36°, while keeping the louver pitch and fin thickness constant at 1.7 mm and 0.1 mm, respectively.

As previously discussed, this study numerically investigates a compact heat exchanger featuring a louvered fin on a flat tube with FLiBe as the working fluid, to provide initial results on its suitability for use in MSRs. While a variety of working fluids are applicable for MSRs, FLiBe was selected for this research. The criteria for choosing materials are detailed in the first part of this section, followed by an explanation of the mathematical methodology in the second part.

Considerations for Coolant Selection

To improve the efficiency of the reactors, several types of working fluids have been proposed and used by researchers. Since a variety of molten salts can be used in the reactors, the selection of the right working fluid is crucial. To ease the selection, a series of the figure of merits (FOM) has been suggested by Kim et al. [33] as regards to the general characteristics that are required in the coolant (Table 1). By considering the suggested FOMs for each general characteristic of the coolants, molten salts considered in MSRs could be compared from a better aspect. The physical properties of typical molten salts used in the reactors and FOM's are summarized in Table 2. The original table could be addressed in ref. [33] in which results of not only the molten salts but also gas and liquid metals could be found as well. In Table 2, in addition to the typical molten salts reported by Kim et al. [33], the physical properties and FOM's of the FLiBe is also enlisted at the bottom in the present study.

The first one, FOM_{ht} , among the suggested FOMs, is regarding the heat transfer performance of the coolant.

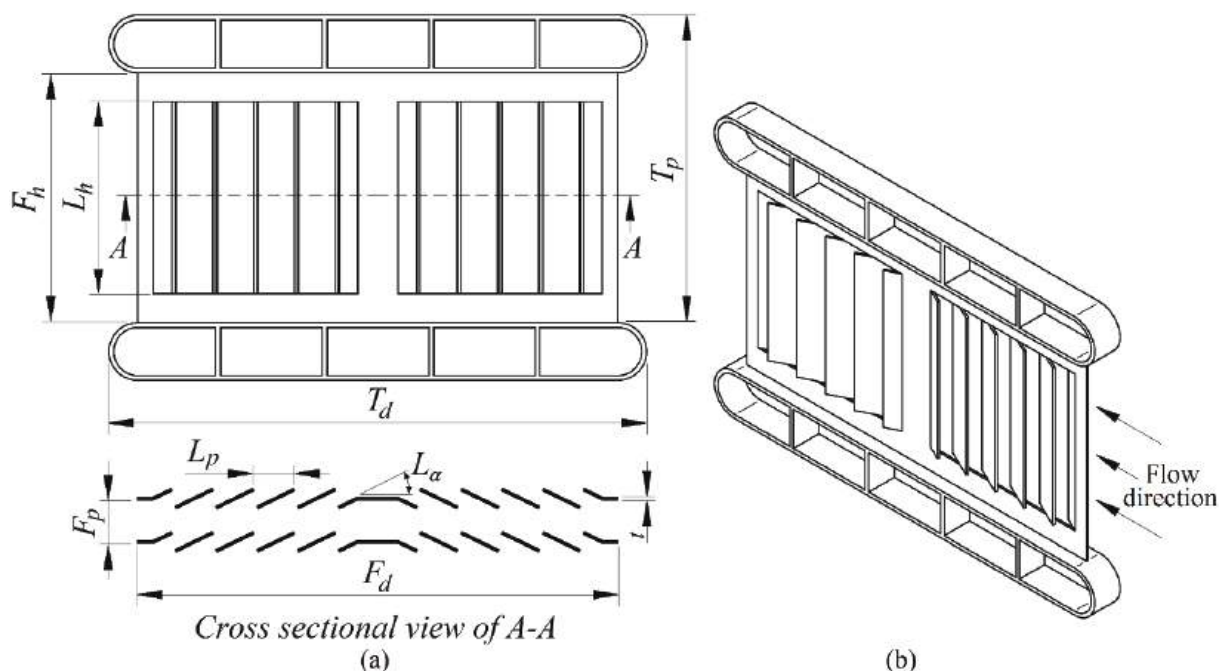


Figure 1. Technical drawing of the considered louvered fin (a) 2-D cross-sectional view of A-A (b) 3-D appearance.

Table 1. Definition of the figure of merits (FOMs)

Definition	FOM
Heat transfer performance of coolant where, $FOM_{ht,0} = (k)^{0.6} (\rho)^{0.58} (C_p)^{0.4} (\mu)^{-0.47}$	$FOM_{ht} = \frac{(k)^{0.6} (\rho)^{0.58} (C_p)^{0.4} (\mu)^{-0.47}}{FOM_{ht,0}}$ (1)
Pumping power of coolant where $FOM_{p,0} = (\rho)^{-2} (C_p)^{-2.8} (\mu)^{0.2}$	$FOM_p = \frac{(\rho)^{-2} (C_p)^{-2.8} (\mu)^{0.2}}{FOM_{p,0}}$ (2)
Volume of coolant where $FOM_{cv,0} = (\rho)^{-0.84} (C_p)^{-1.16} (\mu)^{0.1}$	$FOM_{cv} = \frac{(\rho)^{-0.84} (C_p)^{-1.16} (\mu)^{0.1}}{FOM_{cv,0}}$ (3)
Volume of structural materials where $FOM_{ccv,0} = (P) (\rho)^{-0.84} (C_p)^{-1.16} (\mu)^{0.1}$	$FOM_{ccv} = \frac{(P) (\rho)^{-0.84} (C_p)^{-1.16} (\mu)^{0.1}}{FOM_{ccv,0}}$ (4)
Heat loss of the coolant where $FOM_{hl,0} = (k)^{0.6} (\rho)^{0.34} (C_p)^{0.06} (\mu)^{-0.44}$	$FOM_{hl} = \frac{(k)^{0.6} (\rho)^{0.34} (C_p)^{0.06} (\mu)^{-0.44}}{FOM_{hl,0}}$ (5)

Table 2. Comparison of FOMs for different coolants

Coolant	Melting Point (°C)	k (W/mK)	ρ (kg/m ³)	C_p (J/kgK)	μ (Pa·s)	P (atm)	FOM_{th}	FOM_p	FOM_{cv}	FOM_{ccv}	FOM_{hl}
Water (25°C)	0	0.61	997	4181	0.00089	1	1.00	1.00	1.00	1.00	1.00
LiF-NaF-KF	454	0.92	2020	1886	0.00290	1	0.80	2.87	1.57	1.57	0.92
NaF-ZrF ₄	500	0.49	3140	1173	0.00510	1	0.45	5.02	1.98	1.98	0.56
KF-ZrF ₄	390	0.45	2800	1046	0.00510	1	0.38	8.70	2.50	2.50	0.51
LiF-NaF-ZrF ₄	436	0.53	2920	1233	0.00690	1	0.40	5.36	2.05	2.05	0.50
LiCl-KCl	355	0.42	1520	1198	0.00115	1	0.55	14.99	3.07	3.07	0.76
LiCl-RbCl	313	0.36	1880	890	0.00130	1	0.47	23.08	3.67	3.67	0.70
NaCl-MgCl ₂	445	0.50	1680	1096	0.00136	1	0.58	16.28	3.18	3.18	0.81
KCl-MgCl ₂	426	0.40	1660	1160	0.00140	1	0.50	14.31	3.02	3.02	0.70
NaF-NaBF ₄	385	0.40	1750	1507	0.00090	1	0.71	5.66	2.04	2.04	0.88
KF-KBF ₄	460	0.38	1700	1305	0.00090	1	0.64	8.98	2.47	2.47	0.84
RbF-RbF ₄	442	0.28	2210	909	0.00090	1	0.54	14.63	3.01	3.01	0.75
FLiBe	459	1.00	1940	2414	0.00560	1	0.67	1.78	1.30	1.30	0.73
LiF – BeF ₂											

As for the evaluation of the findings of suggested FOM's, Kim et al. [33] reported that FOM_{ht} would get higher values while the remaining ones (FOM_p , FOM_{cv} , FOM_{ccv} and FOM_{hl}) would get lower values to achieve a better performance. From the highlighted point of view, it is seen that FLiBe is superior than the alternative molten salts. Though it has moderate FOM_{ht} , it has the lowest FOM's in remaining four FOMs.

Numerical Method

Considering that the maximum velocity in each section of the heat exchanger must be limited to 5 m/s to prevent erosion and corrosion [14], the FLiBe flow was presumed to be laminar for $Re_{lp} \leq 1300$ [34]. The mass, momentum, and

energy equations for two-dimensional, steady, laminar, and incompressible flow are provided respectively in Eqs. 6-8.

$$\nabla \cdot V = 0 \quad (6)$$

$$(V \cdot \nabla)V = -\frac{1}{\rho} \nabla P + \nu \nabla^2 V \quad (7)$$

$$(V \cdot \nabla)T = \alpha \nabla^2 T \quad (8)$$

In modeling and analysis of the primary and secondary heat exchangers with louvered fins in MSRs, the expected properties of FLiBe and Hastelloy-N were assumed to be constant as shown in Table 3.

Table 3. Properties of FLiBe and Hastelloy-N

Material	ρ (kg/m ³)	c_p (J/kgK)	k (W/mK)	μ (kg/ms)	Pr
Hastelloy-N [14]	8860	578	23.6	-	-
FLiBe [16]	1940	2414.17	1.0	0.0056	13.525

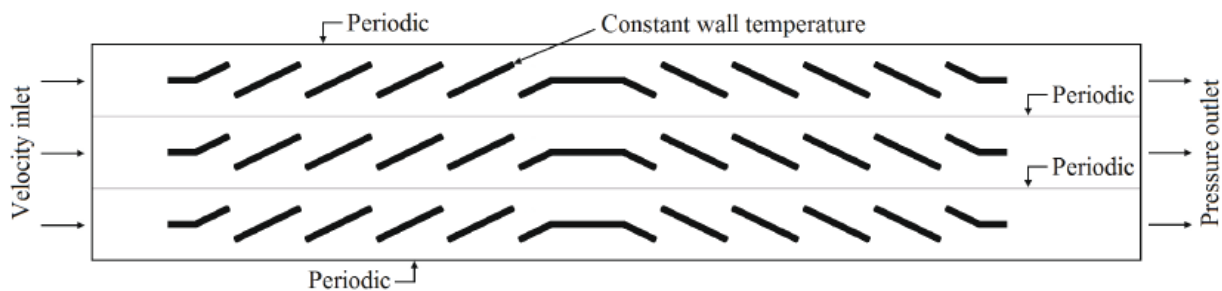
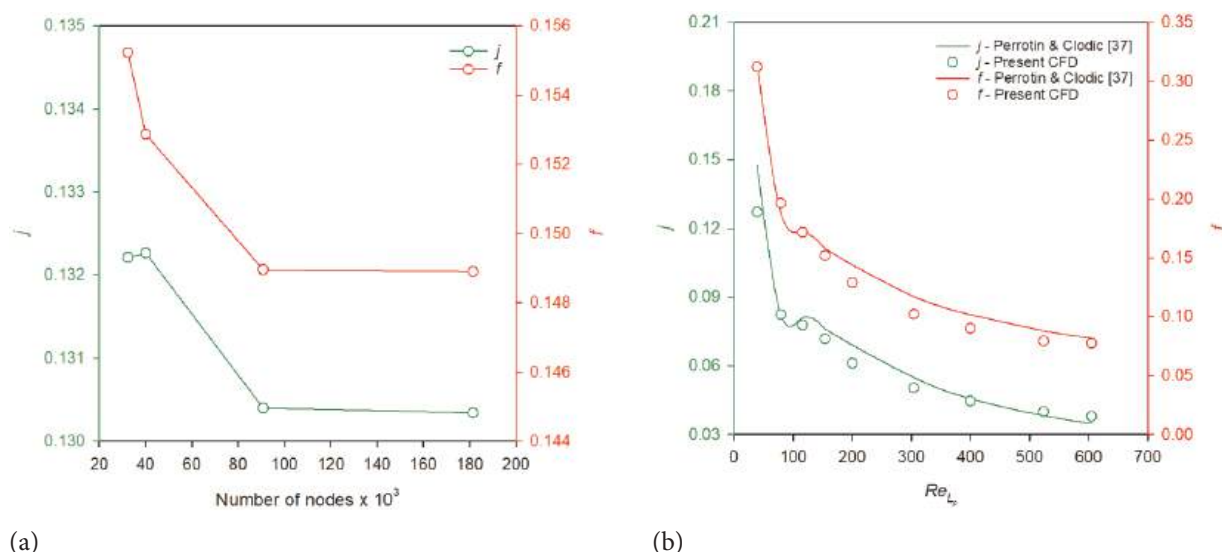
Boundary Conditions

The 2-D computational domain under consideration is depicted in Fig. 2. Louvered fins are characterized as periodic surfaces to reflect their repetitive nature, and a no-slip boundary condition is applied on these surfaces. A constant temperature of 973 K is maintained on the fin walls, considering the melting point of Hastelloy-N lies between 1300–1400°C. The coolant, entering the louvered fins, is kept at 773 K, aligning with the fuel and coolant temperature range of 490–800°C [35]. FLiBe's inlet velocity is set between 0.15–0.80 m/s to ensure laminar flow at the entrance of the louvered fins, and a gauge pressure of 0 Pa is established at the fins' exit. The numerical model's mass, momentum, and energy equations are resolved using Ansys Fluent 14's second-order upwind scheme [36], while the simple algorithm

facilitates the efficient coupling of velocity and pressure fields across the coolant's louvered fins [35].

Validation of the Computational Domain

Four different quadrilateral cell sizes were utilized to achieve mesh independence in the 2-D numerical model, as depicted in Fig. 3a. The discrepancies in the solutions for j and f were approximately 0.16% and 0.96%, respectively, for meshes exceeding 90741 nodes. The scarcity of information on FLiBe molten salt flowing over compact heat exchangers with louvered fins presents significant validation challenges for the computational results against literature. The primary difficulty arises not only from the molten salt and compact heat exchanger combination but also from the minimal flow depth in the flat tube used in the CHEX construction.

**Figure 2.** The 2-D numerical model and the boundary conditions.**Figure 3.** (a) Mesh independency (b) Comparison of the results with the literature.

In light of these challenges, a fundamental comparison with the numerical study by Perrotin and Clodic is employed for computational model validation. Despite the absence of experimental results on this specific subject, the model's reliability is corroborated by the numerical study's findings. Among the various conditions examined in the study, the configuration with $F_p=1.30$ mm, $L_p=0.95$ mm, $L_a=20^\circ$, $F_d=16.5$ mm aligns most closely with our current model. There is a notable concordance between the CFD outcomes of present model and the numerical results of Perrotin and Clodic [37].

Mathematical Method

The mathematical model as a result of the numerical approach is presented in this section. The Reynolds number for the coolant side based on the louvered pitch is given as,

$$Re_{Lp} = \frac{u_c L_p}{\nu} \quad (9)$$

where

$$u_c = u_{in} \frac{A_{fr}}{A_c} \quad (10)$$

The heat transfer coefficient of the coolant side is determined by using Eq. 11.

$$h = \frac{\dot{Q}}{F A_a \Delta T_m} \quad (11)$$

where the heat transfer rate and logarithmic mean temperature difference are defined as,

$$\dot{Q} = \dot{m} c_p (T_o - T_i) \quad (12)$$

$$\Delta T_m = \frac{T_i - T_o}{\ln[(T_s - T_o)/(T_s - T_i)]} \quad (13)$$

In the design of heat exchangers, overall performance criteria are preferred over the direct use of the heat transfer coefficient. The pressure gradient is a crucial design parameter to consider when molten salt flows over louvered fins. The thermal and hydraulic performance of a louvered fin heat exchanger is characterized by the Colburn j -factor and the friction factor f , as given in Eqs. 14 and 15.

$$j = St Pr^{2/3} = \frac{Nu}{Re_{Lp} Pr^{1/3}} \quad (14)$$

$$f = \frac{2\Delta P}{\rho u_c^2} \left(\frac{A_c}{A_a} \right) \quad (15)$$

where

$$St = \frac{h}{\rho u_c c_p} \quad (16)$$

The volume goodness factor of $j/f^{1/3}$ is used for expressing new sights about the overall performance of the louvered fin heat exchanger.

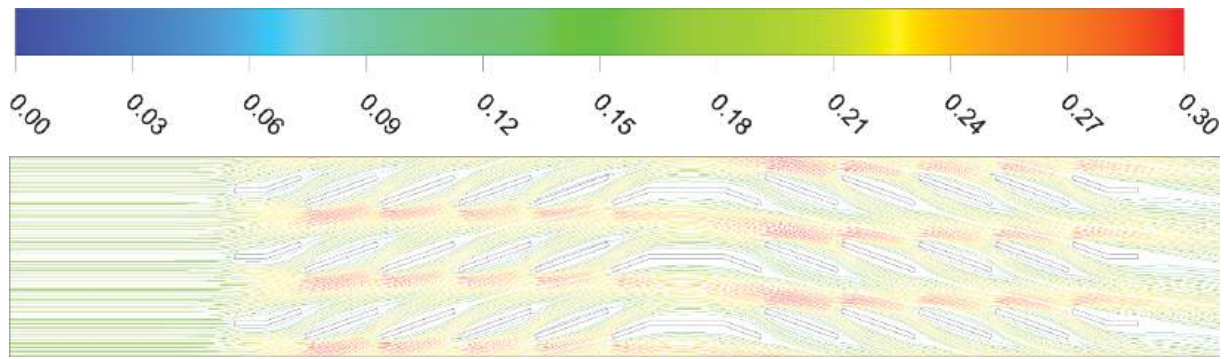
RESULTS AND DISCUSSION

In the present study, the thermo-hydraulic performance of the molten salt FLiBe is investigated numerically. The investigation is conducted for varying fluid velocities in the range of 0.1 m/s–0.8 m/s, fin pitches in the range of 1.5–2.5 mm, and fin louver angles in the range of 20° – 36° . The findings of the simulations are presented either in dimensional form (heat transfer coefficient vs. frontal velocity, pressure drop vs. frontal velocity) or non-dimensional form (Colburn j -factor vs. Re_{Lp} number, f -factor vs. Re_{Lp} number). As for the initial step, the contour plots for streamline, pressure, and temperature are demonstrated to provide a better perspective on the flow characteristics of the studied molten salt. Then the thermo-hydraulic performance of the FLiBe is presented by the diagrams that are driven for heat transfer coefficient, pressure drop, Colburn j -factor, friction factor f , and volume goodness factor $j/f^{1/3}$.

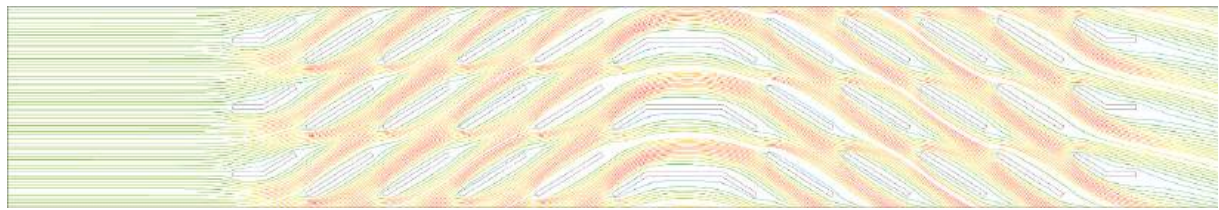
The streamlines which are tangent to the local velocity vectors during the flow of FLiBe are presented in Fig. 4 for the fin pitch $F_p=1.5$ mm and for the louver angles $L_a=20^\circ$ and 32° where $Re_{Lp}=100$. These lines are used to observe the inclination of the flow direction during the analyses, as regards to the given figures, when the stream becomes stronger in any direction, the color turns to red from blue. While blue streamlines demonstrate the weaker streams, red corresponds to a stronger flow stream. It could be seen from the schematic, when the louver angle is small ($L_a=20^\circ$), the molten salt tends to flow through the space between louver fins which causes a lesser interaction between the FLiBe and fin surfaces. On the contrary, the interaction between fin surfaces and FLiBe improves as the louver angle becomes steep. As it is presented in Fig. 4, the streamlines of molten salt flow become stronger, and the color of the streams turns red when FLiBe flows over the fins. It can be noted that, as the louver angle is steeper the flow of the molten salt can be more directed to louver fins and a “louver directed flow” can be obtained vice versa if the louver angle is moderate, molten salt's flow turns to a duct directed flow instead.

The pressure changes along with the louver fins and consequently the drop between the upstream and downstream can be observed from Fig. 5. As it could be seen, as the louver angle turns from $L_a=20^\circ$ to 32° , the pressure changes from upstream to downstream increases. As regards to the presented figure, the pressure of FLiBe is high at the upstream and rises locally in particular at the leading edge in both presented louver angles. The rise at the leading edge is bigger when the fin is steep since the resistance against the flow is higher. As the molten salt flows over the fin, the pressure falls along the flow path and becomes the lowest at the downstream. The pressure difference between the inlet and exit of the molten salt flow is higher when the fin is steeper, and this variance descends as the louver angle becomes more moderate where the resistance against the flow is lower.

Streamline [m/s]



(a)



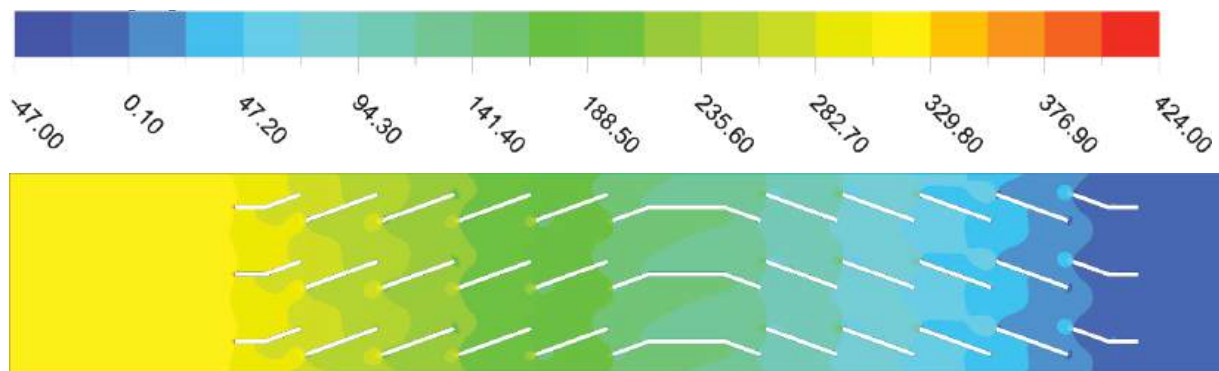
(b)

Figure 4. Streamlines of FLiBe flowing through the louvered fins for $F_p=1.5$ mm and $Re_{lp}=100$ (a) $L_a=20^\circ$ (b) $L_a=32^\circ$.

The temperature distribution along the louver fins is presented in Fig. 6 for the $F_p=1.5$ mm and $L_a=20^\circ$ and 32° at $Re_{lp}=100$. Temperature rises of the molten salt from the upstream to the downstream can be seen clearly in the

presented contour plots. While the FLiBe's temperature at the upstream is 773 K at the downstream temperature can rise to 853–873 K locally as regards to heat transfer over the fins to the molten salt. Due to the interaction between the

Pressure [Pa]



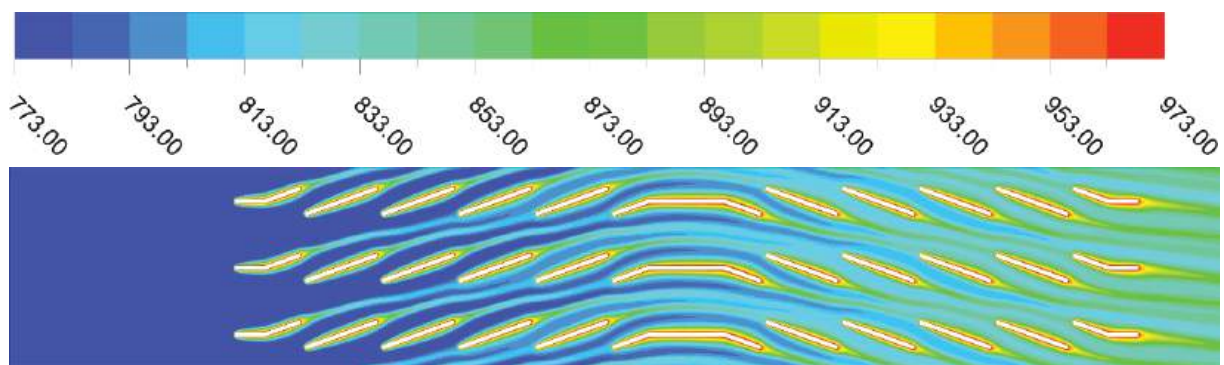
(a)



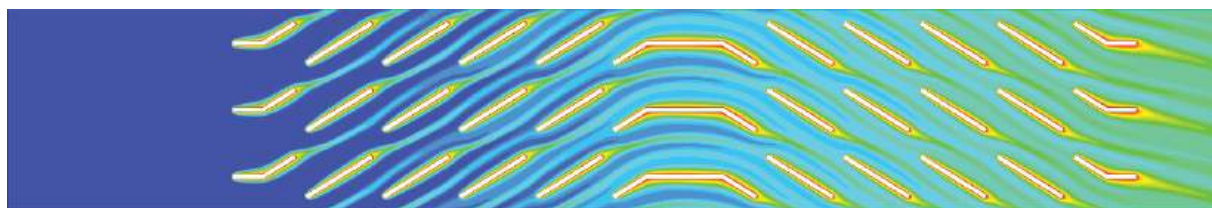
(b)

Figure 5. Pressure fields of FLiBe flow through the louvered fins for $F_p=1.5$ mm and $Re_{lp}=100$ (a) $L_a=20^\circ$ (b) $L_a=32^\circ$.

Temperature [K]



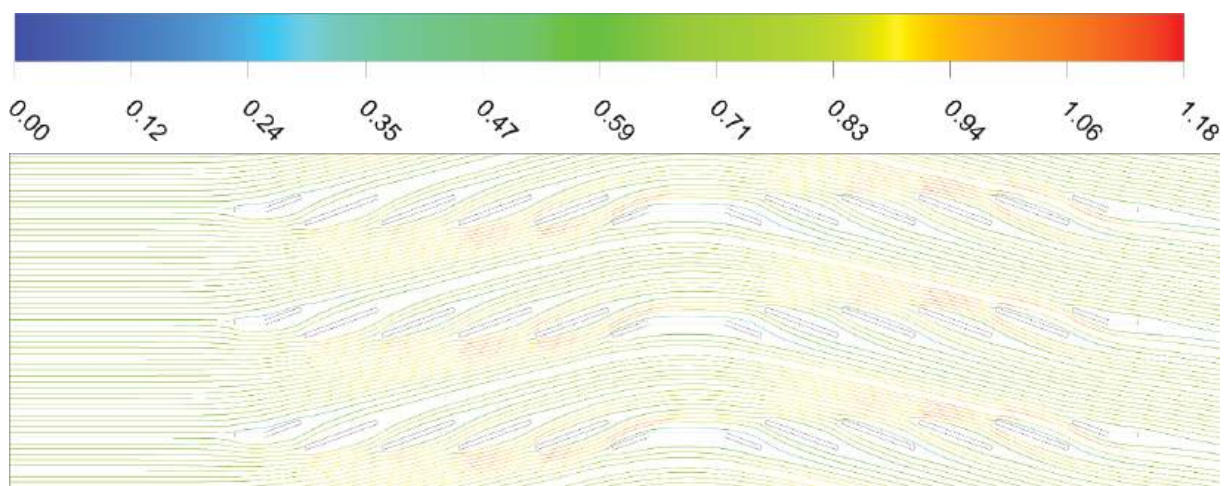
(a)



(b)

Figure 6. Temperature fields of FLiBe flow through the louvered fins for $F_p=1.5$ mm and $Re_{Lp}=100$ (a) $L_a=20^\circ$ (b) $L_a=32^\circ$.

Streamline [m/s]



(a)



(b)

Figure 7. Streamlines of FLiBe flowing through the louvered fins for $F_p=2.5$ mm and $Re_{Lp}=400$ a) $L_a=20^\circ$ (b) $L_a=32^\circ$.

fin surface and FLiBe, more heat is transmitted to the molten salt in the neighboring regions of the fins in which temperature can go up to 950 K as it is plotted in the figures by red color. In accordance with the louver angle increment, the molten salt gains more heat, and the temperature of the molten salt ascends gradually as the L_a rises from 20° to 32° . The temperature of the FLiBe is higher when the louver angle is steep, the major reason for this increment relies on the flow characteristics which are presented in Fig. 4. When the louver angle rises, the flow turns from duct directed to louvered directed which improves the interaction between the fins and molten salt, and consequently, the heat transfer is improved.

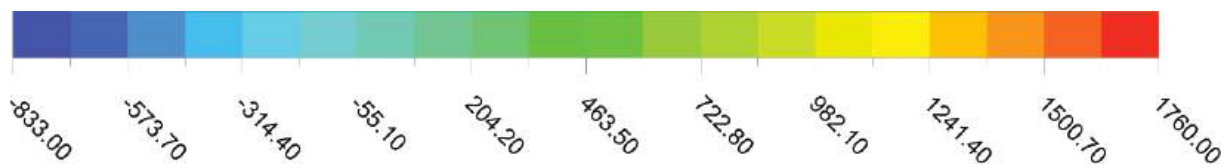
The evolution of the streamlines, pressure, and temperature by the contour plots are presented for the spacing $F_p=2.5$ mm in Fig 7-9. To present the findings from a different aspect. A wider fin spacing ($F_p=2.5$ mm) and higher Reynolds number ($Re_{Lp}=400$) are chosen in the mentioned figures and temperature, pressure, and streamline contours are plotted for these cases. Once again, the louver angles $L_a=20^\circ$ and 32° are chosen to demonstrate the impact of

the louver angle change on the flow characteristics of the molten salt. Similar trends are observed in each figure for the corresponding parameters, such as, while the “duct directed” flow is observed for $L_a=20^\circ$, it turns to “louvered directed flow” when the louver angle becomes steep ($L_a=32^\circ$) in Fig 7.

The pressure reduces across the flow direction, the higher pressure is observed at the upstream and lower pressure is seen at the downstream in both louver angles of $L_a=20^\circ$ and 32° . Since the resistance is more when the louver fins are steep, the pressure at the upstream is higher $L_a=32^\circ$ and therefore the pressure difference between inlet and exit is more as it could be seen in Fig. 8.

The temperature distribution over the molten salt increases along the flow direction as well in Fig. 9 during the interaction of the molten salt with fins. Though identical trends are observed, some notable features are detected as well in the demonstrated figures. For instance, even though flow directions are identical, streamlines are sparse due to the higher fin pitch used in this model (Fig. 7). The major reason for this remarkable change is the space

Pressure [Pa]



(a)



(b)

Figure 8. Pressure fields of FLiBe flowing through the louvered fins for $F_p=2.5$ mm and $Re_{Lp}=400$ (a) $L_a=20^\circ$ (b) $L_a=32^\circ$.

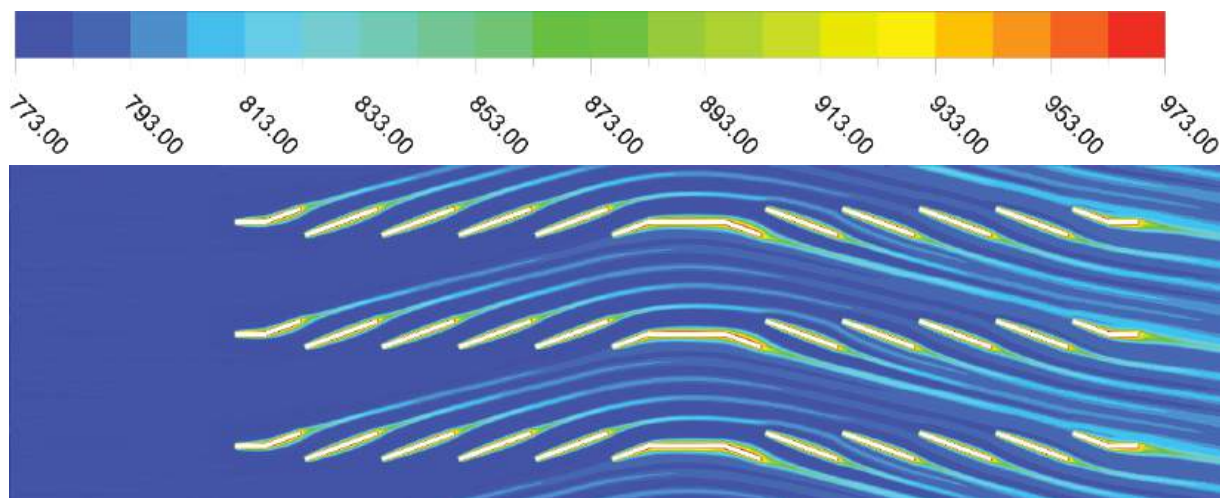
between louver fins indicated by the pitch which is $F_p=2.5$ mm instead. As the space grows, the bigger portion of the molten salt flows through that space instead of flowing over the fins and flow becomes more “duct directed” instead of “louvered directed”. This is also the major reason for the worsening thermal performance observed. Fig. 9. As it is seen, the region that gains heat from the fins is very limited since the interaction between the FLiBe and fins gets worse. As the contact reduces and molten salt has more free space to flow, the heat transfer from the fins to the FLiBe gradually descends and the temperature plot has more blue regions which refer to inlet temperature.

Heat transfer coefficient (h) and pressure drop (ΔP), variation with respect to molten salt frontal velocity is presented in Fig.10 for three different fin pitches ($F_p=1.5$, 2.0, and 2.5 mm). As the velocity increases, the heat transfer performance improves regardless of the variation of the louver angle and fin pitch of the louvered fins. The left-hand side of the figures pertains to heat transfer coefficients, which refers to the curve cluster at the top, while

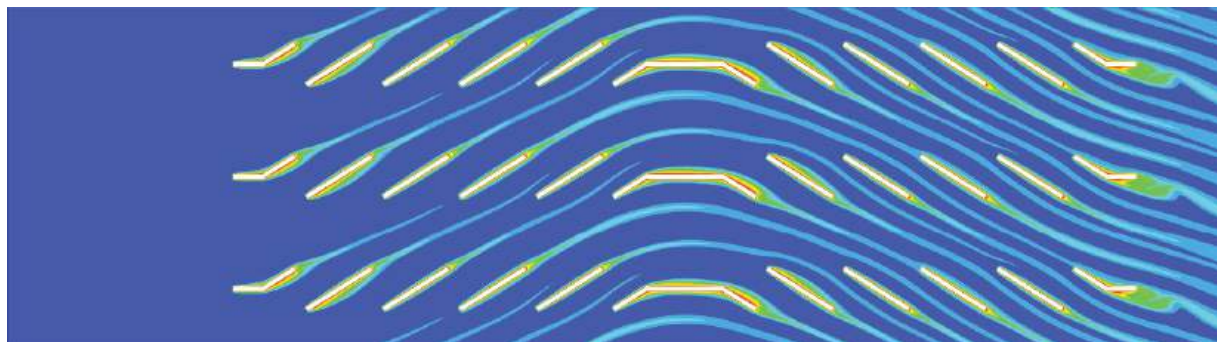
the right-hand side pertains to pressure drop, which refers to the curve cluster at the bottom of the figures. Among the investigated louver angles, the $L_a=36^\circ$ provides the highest heat transfer which gets higher values as the frontal velocity ascends. The major reason for this improvement relies on the flow direction change provided by the variation of the louver angle as presented before with the contour plots. Since the molten salt flow becomes more louver-directed when the louver angle is steep, it enhances the interaction between the FLiBe and fins which leads to a thermal performance increment. The maximum h is obtained as $27000 \text{ W/m}^2\text{K}$ at this louver angle ($L_a=36^\circ$) and for the velocity of 0.8 m/s among all tested cases, while the lowest is obtained as $12000 \text{ W/m}^2\text{K}$ at ($L_a=20^\circ$) and for the velocity of 0.1 m/s .

Even though there is a noticeable difference between the louver angle of $L_a=36^\circ$ and the remaining louver angles, this difference is not obvious as the L_a decreases, so h values begin to converge at lower louver angle values and starts to present identical evolution trends by the frontal velocity.

Temperature [K]



(a)



(b)

Figure 9. Temperature fields of FLiBe flowing through the louvered fins for $F_p=2.5$ mm and $Re_{Lp}=400$ (a) $L_a=20^\circ$ (b) $L_a=32^\circ$.

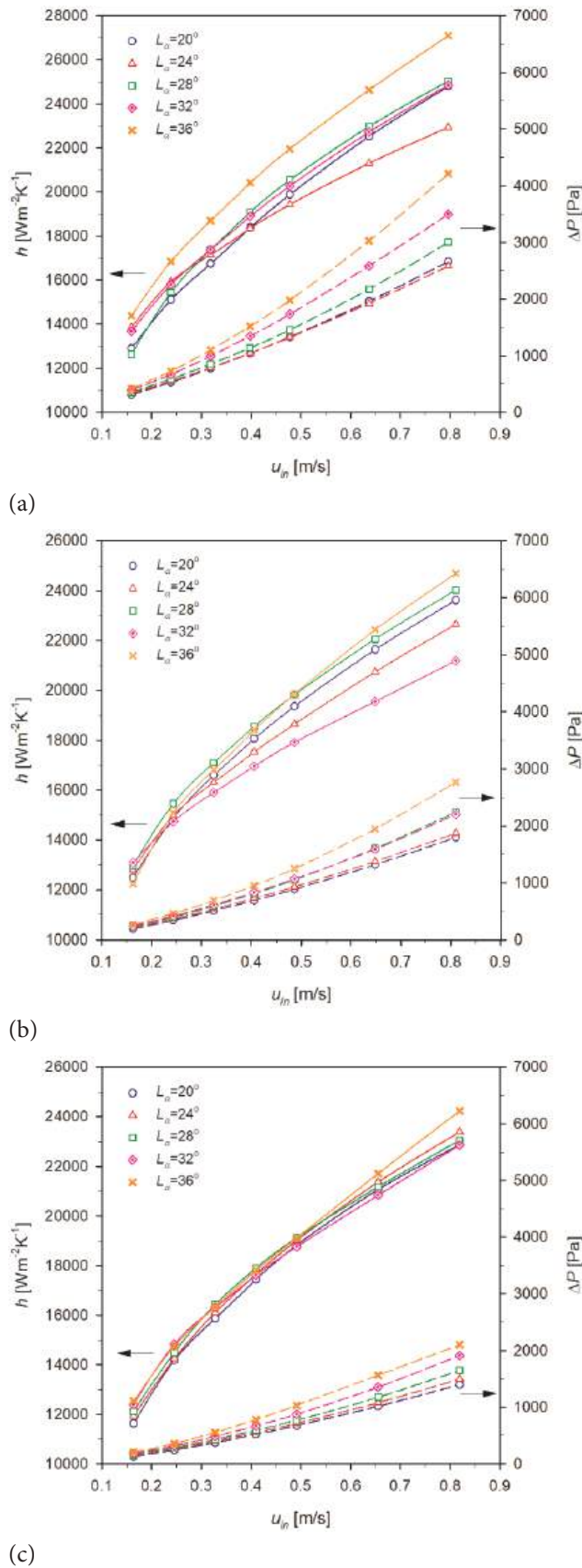


Figure 10. Deviation of heat transfer coefficient (h) and pressure drop (ΔP) with respect to frontal velocity for FLiBe (u_{in}) (a) $F_p=1.5$ mm (b) $F_p=2.0$ mm (c) $F_p=2.5$ mm.

The evolution characteristics are more obvious for pressure drop (ΔP) curves in the same figure (Fig. 10a). As the louver angle grows, the fins become steep and as a result of increasing pressure at the leading edge, the difference between the upstream and downstream becomes more obvious. The highest pressure drop is obtained for the highest louver angle at the maximum frontal velocity while the lowest is observed for the lowest louver angle and the lowest velocity. The possible reason for the reported change can be related to the louver angle. As the louver angle increases, the resistance against molten salt flow ascends too, which leads to a higher pressure drop. Similarly, as the velocity increases, the pressure drop rises as well. Contradiction to this, as the frontal velocity reduces, pressure drop reduces and when this fall approaches about 0.1 m/s, almost all louver angles show a similar pressure drop across the flow over the fins at this particular velocity. It is seen that the resistance that emerges by the louver angle rise doesn't have an impact at lower frontal velocities as in the higher ones.

Similar evolution trends for heat transfer (h) and pressure drop (ΔP) can be observed for $F_p=2.0$ mm in Fig. 10b. As it is seen, either h or ΔP increases as the frontal velocity of the molten salt ascends, in addition, both of them rise as the louver angle steeps. Unlike Fig. 10a, h curves of varying louver angles are more obvious at higher frontal velocities. The gradual increment of h plots in each louver angle is clearer than the previous fin pitch. The highest h is obtained around 25000 W/m²K when the $L_a=36^\circ$ and $u_{in}=0.8$ m/s. As for ΔP , the difference between the plots becomes smaller at this fin pitch but is still apparent. The louver angle changing from $L_a=32^\circ$ to 28° doesn't have much impact on pressure drop and is located between $L_a=36^\circ$ and 24° . Even the $L_a=20^\circ$ has the least ΔP among the analyzed louver angles, ΔP at $L_a=24^\circ$ is very close and there is a vague difference between these two. Fig. 10c demonstrates the heat transfer coefficient (h) and pressure drop (ΔP) evolution for the fin pitch $F_p=2.5$ mm. Though the trend is identical with the earlier findings, h 's are becoming closer than earlier plots which are belonging to fin pitches of 1.5 mm and 2.0 mm. Since the space between the louver fins increases, the duct-directed flow becomes more prominent for the molten salt and as a result, the interaction with the fins descends and thermal performance reduces. Associated with the previous reason, the impact of the louver angle on the thermal performance descends as well and h could be able to reach to 24000 W/m²K, though the ΔP trend and the order among the louver angles are similar, the values of ΔP and curves are plotted on a narrower range, unlike the earlier figures. While pressure drops can go beyond 3000 Pa and be able to reach 4000 Pa for the smaller fin pitches, when the $F_p=2.5$ mm, the ΔP barely exceeds 2000 Pa.

The Colburn j -factor and friction factor f evolution as regards to Re number for three different fin pitches ($F_p=1.5$, 2.0, and 2.5 mm) are given in Fig. 11 to see the impact of the louver angle and velocity on the performance from a non-dimensional perspective. The Colburn j -factor and friction

factor f reduce as the Re number increases as regards to the nature of these factors' definitions. The highest Colburn j -factor is observed for the biggest louver angle in accordance with the earlier findings. The Colburn j -factor plots begin to cross over when the louver angle is less than $L_a=36^\circ$ and it seems below this value, louver angle degree does not have much impact on thermo-hydraulic performance when fin pitch is $F_p=1.5$ mm (Fig 11a). Even though there is a vague difference between the curve for $L_a=24^\circ$ and the rest of them, the variance cannot be reported as a remarkable one. The lowest performance for the Colburn j -factor can be marked at this particular condition at around 0.035.

The flow direction which is oriented by the louver angle change has an unavoidable impact on the thermal performance. As the louver angle steeps the flow turns to a louver-directed one and improves the interaction of the fin and molten salt and consequently improves the performance. As for the friction factor f , the evolution curves are more observable than the Colburn j -factor curves. The overall trend of the friction factor f reduces by increasing Re_{Lp} number which can be observed in all investigated louver angles. The highest f factor is recorded as 0.52 for the $L_a=36^\circ$ when Re_{Lp} is 100 while the lowest is obtained for $L_a=20^\circ-24^\circ$ when Re_{Lp} is 500. As it is seen, the friction factor at the louver angles $L_a=20^\circ-24^\circ$, becomes very similar when Re_{Lp} is more than 200. Beyond that threshold, two curves overlap and show identical evolution. As it could be seen, as the louver angle becomes higher, the louver fins position becomes steep against flow which causes higher resistance against the FLiBe flow. Consequently, the pressure drop becomes higher and so the friction factor increases. The exact opposite can be noted too, as the louver angle gets smaller, the flow encounters lower resistance during the flow in particular at lower Reynolds number and it allows smoother transition over the fins. In Fig. 11b, Colburn j -factor and friction factor f evolution are presented, for $F_p=2.0$ mm. The overall trend is almost similar with $F_p=1.5$ mm except for the range that the curves are plotted. While the plots are in the range of 0.10 - 0.04 for $F_p=1.5$ mm, this range turns to 0.09 to 0.03 for $F_p=2.0$ mm. The evolution of the Colburn j -factor for all louver angles drowns as a cluster for the fin pitch of $F_p=2.0$ mm. The only observable plot appears for lower louver angles such as $L_a=20^\circ$. In this particular condition, the lowest performance is observed and plotted below the other investigated louver angles. As for friction factor f , the trend is similar to $F_p=1.5$ mm as well. The highest and lowest plots are pertaining to $L_a=36^\circ$ and 20° respectively. The difference between $L_a=20^\circ$ and 24° at higher Re_{Lp} is more prominent in this case. The highest friction factor f is obtained as 0.07 when the $L_a=36^\circ$ and $Re_{Lp}=100$. The major reason for the highest friction factor for $L_a=36^\circ$ is the higher resistance that molten salt encounters during the flow which escalates as the louver angle steeps.

Colburn j -factor and friction factor f change for $F_p=2.5$ mm is demonstrated in Fig. 11c. Even though they have the same trend, it is becoming hard to distinguish the most

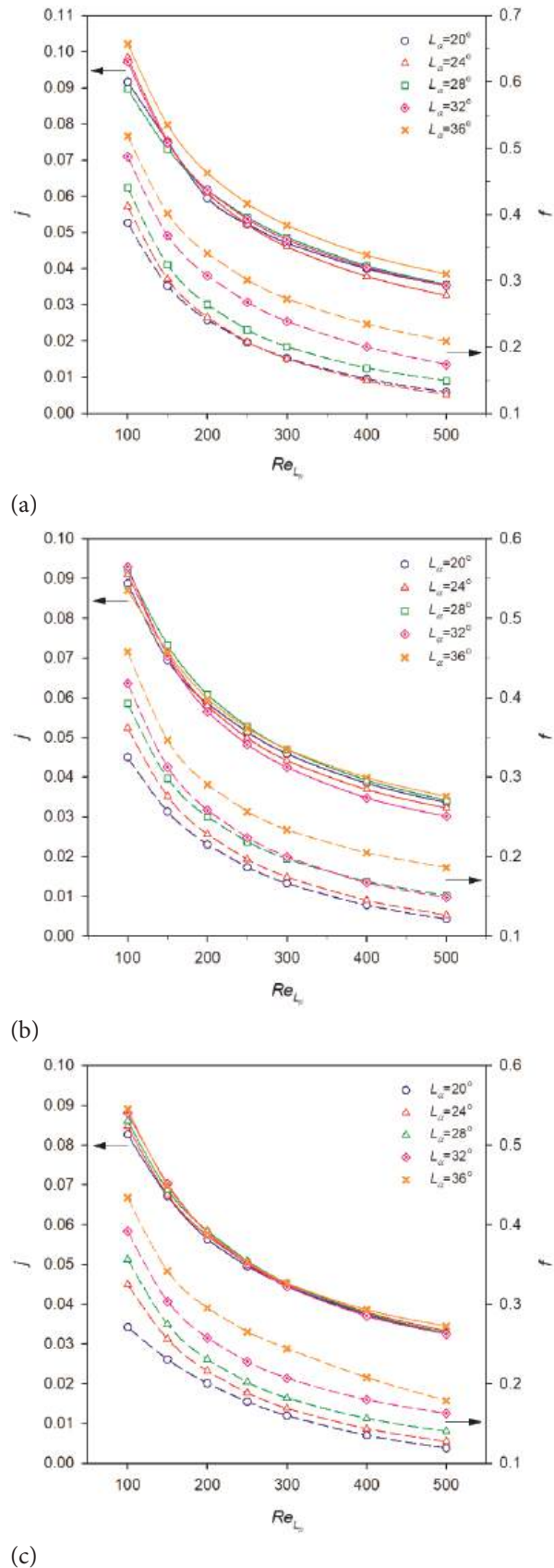


Figure 11. Deviation of Colburn j - and f -factors with respect to frontal velocity for FLiBe (a) $F_p=1.5$ mm (b) $F_p=2.0$ mm (c) $F_p=2.5$ mm.

efficient louver angle at this fin pitch since the plot cluster is more congested. The Colburn j -factor plots are packed in a narrower range than earlier fin pitches and the impact of the louver angle on the performance becomes very difficult when the space between the fins grows and flow becomes more duct oriented. Unlike Colburn j -factor, friction curves are more separated, and the difference between the curves is more obvious as the F_p increases. The highest and lowest friction factors can be identified more clearly. Though the trend has not changed, the maximum friction factor f is observed at around 0.43 at the $L_a=36^\circ$ when Re_{Lp} is 100 while the minimum f is obtained at around 0.03 at $L_a=20^\circ$ when Re_{Lp} is 500. When the overall picture is considered by taking all diagrams into account, it is seen that all value ranges for h , ΔP , j and f reduce as the F_p increases. With regard to summarized findings, thermal performance measures diminish when the fin pitch increases as a consequence of the enlargement of the uninterrupted flow region which refers to the duct-directed flow. As fin pitch becomes lower, the molten salt flow turns to louvered directed flow which enhances the interaction of the fin and FLiBe that allows improving the thermal performance. In addition, the louver angle has an important role in the pressure drop and friction either, as the louver angle steeps, the molten salt encounters more resistance beginning from the leading edge. The trends observed for parameters like the Colburn j -factor and f -factor are consistent with findings in the literature regarding louver angle and fin pitch [38]. However, differences in magnitude and range arise due to the unique thermophysical properties of molten salt.

The performance of the fins is investigated by some other analogies in literature as well. One of the widely preferred criteria is the volume goodness factor ($j/f^{1/3}$) which allows considering the thermal and hydraulic performance

together. The volume goodness factors evolution for all considered louver angles and fin pitches with respect to Re_{Lp} is presented in Fig. 12. In the figures, the findings are presented in three individual schemes which correspond to the fin pitches analyzed in the study. From left to right, the diagrams correspond to the fin pitches of $F_p=1.5$ mm, 2.0 mm, and 2.5 mm. It could be seen that the overall trend of the volume goodness factor is to descend as the Re_{Lp} increases. Even though every case considered has a similar tendency and obtains values in a similar range which looks like a cluster in each figure, minimum and maximum values are noticeable at each Re_{Lp} . Besides, the unlikely characteristics can be identified among the curve cluster as well. From a wider aspect, for most of the considered Re_{Lp} , the curve pertaining to $F_p=2.0$ mm and $L_a=20^\circ$ shows better performance while the curve belongs to $F_p=2.5$ mm and $L_a=36^\circ$ shows the worst performance from the volume goodness factor aspect. The maximum volume goodness factor at $Re_{Lp}=100$, is obtained as 0.13 and for the case where $F_p=1.5$ mm and $L_a=24^\circ$ while the lowest value is 0.112 for the louver fins $F_p=2.0$ mm and $L_a=36^\circ$. When the Re_{Lp} number is slightly increased there is a minor change in the response of the fins for this increment, unlike the first case, $F_p=2.0$ mm and $L_a=20^\circ$ shows better performance than other fin pitches and louver angles while $F_p=2.5$ mm and $L_a=36^\circ$ gets the lowest value. The observed highest and lowest values for this Re_{Lp} are 0.11 and 0.09 respectively. Though the difference isn't very distinct, for the Re_{Lp} number 300 and 400, the highest and lowest performance from the volume goodness aspect, are observed for the louver fins with $F_p=2.0$ mm and $L_a=20^\circ$ and $F_p=2.5$ mm and $L_a=36^\circ$, respectively. Even though the order hasn't changed a lot when Re_{Lp} number is 500, the lowest performance is observed for the louver fin with $F_p=2.0$ mm and $L_a=32^\circ$ instead. It could be seen; the

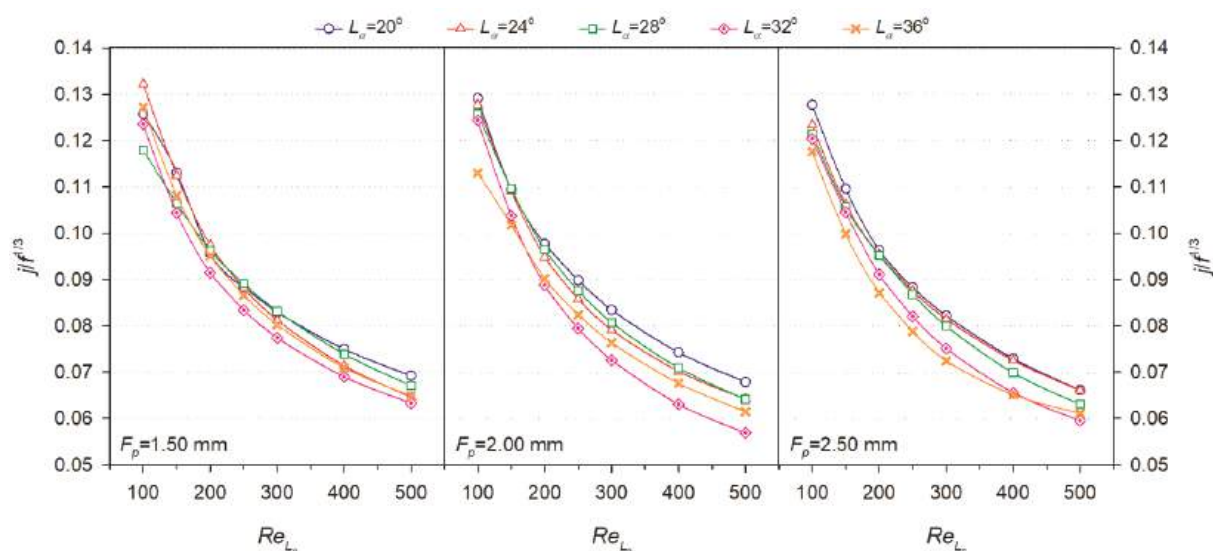


Figure 12. Deviation of volume goodness factor ($j/f^{1/3}$) with respect to Reynolds number.

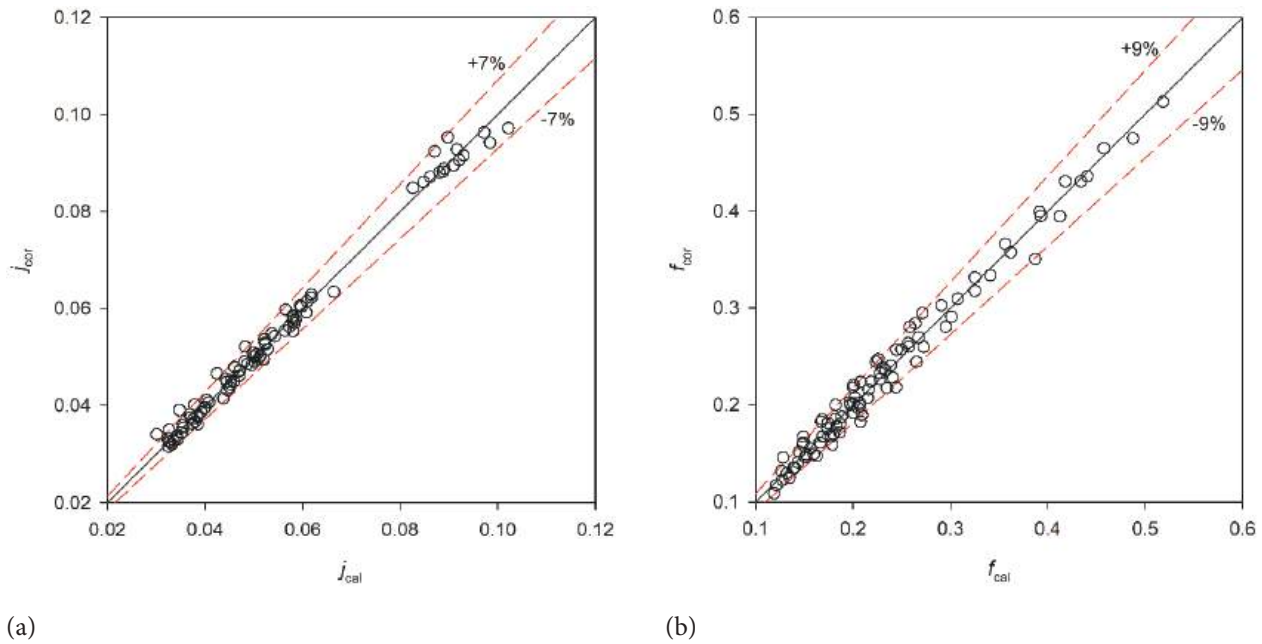


Figure 13. Comparison of numerical data and correlation for (a) j -factor and (b) friction factor.

highest volume goodness factor is obtained as 0.13 while the lowest is observed as 0.06. As regards to outcomes, though the louver fin models with $L_a=20^\circ$ don't show higher performance from both heat transfer and Colburn j -factor aspect, when the combined effect of Colburn j - and f -factors are considered, it becomes the most preferable option among the others since the pressure loss and friction factor values are considerably lower than the other cases.

Finally, the correlations for the Colburn j -factor and the friction factor f are derived by considering ninety data points, presented in Eqs. 17 and 18, respectively. Fig. 13a and Fig. 13b show the comparison of numerical results with the present correlated data for j - and f -factors. As can be seen, 95% of the j - and f -factors data are correlated with in a range of $\pm 7\%$ and $\pm 9\%$, respectively. The proposed j -factor correlation has a mean deviation of 2.55% while the proposed f -factor correlation has a mean deviation of 4.55%.

$$j = 1.73(Re_{Lp}^{-0.615}) \left(\frac{L_a}{90}\right)^{0.078} \left(\frac{F_p}{L_p}\right)^{-0.175} \quad (17)$$

$$f = 15.36(Re_{Lp}^{-0.619}) \left(\frac{L_a}{90}\right)^{0.647} \left(\frac{F_p}{L_p}\right)^{-0.341} \quad (18)$$

CONCLUSION

Minimizing the fuel inventory in the heat exchanger of a molten salt reactor (MSR) is highly desirable. This can be achieved through the use of a compact heat exchanger that maximizes the heat transfer area while minimizing its volume. One of the most promising candidates for such a

design is the use of louvered fins over mini-channelled flat tubes. The physical properties of the molten salt in both the primary and secondary fluid circuits influence not only the size and cost of the heat exchanger but also the required pumping power, piping dimensions, fuel inventory, and overall design cost. For the contribution to the molten salt reactor literature, which hasn't been reported before, the steady-state thermo-hydraulic performance of the FLiBe flowing through a compact heat exchanger with multi louvered fins has been investigated with regard to the typical MSR's thermal conditions. A series of numerical tests were conducted for three different fin pitches and five different louver angles in the low-velocity range as part of a preliminary study for an ongoing investigation. Steady-state velocity profiles, temperature distributions, and pressure fields were obtained for five different Reynolds numbers. The heat transfer coefficient and pressure drop were analyzed in terms of the Colburn j -factor, the Fanning friction factor f , and the volume goodness factor.

The effect of louvered fins, designed to enhance heat transfer, was evaluated. While the heat transfer coefficient improved, it came at the expense of an increased pressure drop. However, the increased pressure drop has a beneficial effect in reducing the fuel inventory. This benchmark study was conducted for low-velocity profiles to assess the suitability of compact heat exchangers for molten salt reactor (MSR) applications. It investigates the thermo-hydraulic behavior of FLiBe as it flows through the louvered fins in a compact heat exchanger (CHX). The outcomes obtained can be highlighted as in the following:

- As the louver angle increases, the upstream pressure rises while the downstream pressure decreases, resulting in a greater overall pressure drop.
- As the louver angle increases, the flow of FLiBe becomes more aligned with the louvers, improving the interaction between the molten salt and the louvered fins, which enhances thermal performance.
- As the fin pitch increases, the flow becomes more duct-directed due to the larger spacing between the fins. This reduces the interaction between the molten salt and the fins, leading to a decline in the thermal performance of the compact heat exchanger (CHX).
- As the velocity of the molten salt flow increases, both the heat transfer coefficient and the pressure drop rises for all investigated fin pitches and louver angles. The highest heat transfer coefficient range is obtained for $F_p=1.5$ mm by 12000-28000 W/m²K, while this range changes as 12000-25000 W/m²K and 12000-24000 W/m²K for the $F_p=2.0$ mm and $F_p=2.5$ mm respectively. A similar decrement is observed when the fin pitch impact is observed on the pressure drop. The ΔP range is 4500-0 Pa for $F_p=1.5$ mm, while it reduces to 3000-0 Pa and 2000-0 Pa for $F_p=2.0$ mm and $F_p=2.5$ mm respectively.
- The Colburn j -factor and f -factor decrease as the Reynolds number increases, regardless of the fin pitch or louver angle. As the fin pitch increases, thermal performance declines due to reduced interaction, resulting in a corresponding decrease in the Colburn j -factor range. While it evolves in the range of 0.10–0.04 for the $F_p=1.5$ mm, it drops to 0.095–0.03 and 0.09–0.03 for the $F_p=2.0$ mm and $F_p=2.5$ mm respectively. The same trend is observed for the friction factor too.

This study bridges two key fields: MSRs and the design of compact heat exchangers with enhanced heat transfer surface areas. As such, it provides valuable insights for researchers, product designers, and manufacturers working on the thermohydraulic systems of MSRs. As highlighted by the summarized findings, this comprehensive investigation into the thermohydraulic of FLiBe under boundary conditions relevant to MSRs contributes to the development of more efficient compact heat exchangers. These advancements enhance heat transfer from the reactor core, enabling reduced system volume and minimizing the fuel salt inventory outside the core.

NOMENCLATURE

A_a	air side heat transfer area, m ²
A_c	minimum flow area, m ²
A_{fr}	frontal area, m ²
c_p	specific heat at constant pressure, J·kg ⁻¹ ·K ⁻¹
f	friction factor
F	correction factor
F_d	flow depth, mm
F_h	fin height, mm

F_p	fin pitch, mm
h	heat transfer coefficient, W·m ⁻² ·K ⁻¹
j	Colburn j -factor
k	thermal conductivity, W·m ⁻¹ ·K ⁻¹
L_α	louver angle, °
L_h	louver height, mm
L_p	louver pitch, mm
$\dot{m}$	mass flow rate, kg·s ⁻¹
P	pressure, Pa
Pr	Prandtl number
$\dot{Q}$	heat transfer rate, W
Re_{Lp}	Reynolds number based on louver pitch
St	Stanton number
t	thickness of the louvered fin, mm
T	temperature, K
T_i	inlet temperature, K
T_d	tube depth, mm
T_p	tube pitch, mm
T_o	outlet temperature, K
T_s	surface temperature, K
ΔT_m	logarithmic mean temperature difference, K
V	velocity, m·s ⁻¹
u_c	maximum velocity at minimum flow area, m·s ⁻¹
u_{in}	free flow velocity at the inlet, m·s ⁻¹

Greek letters

α	thermal diffusivity, m ² ·s ⁻¹
μ	dynamic viscosity, kg·m ⁻¹ ·s ⁻¹
ρ	density, kg·m ⁻³
ν	kinematic viscosity, m ² ·s ⁻¹

Abbreviations

ANP	Aircraft Nuclear Propulsion
ARE	Aircraft Reactor Experiment
FOM	Figure of Merit
MSBR	Molten Salt Breeder Reactor
MSRE	Molten Salt Reactor Experiment
MSRs	Molten Salt Reactors
NGNP	Next Generation Nuclear Plant
ORNL	Oak Ridge National Laboratory
TEMA	Tubular Exchanger Manufacturers Association
THORIMS-NES	Thorium Molten Salt Nuclear Energy Synergetic

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Research Article

Examining the effect of temperature and thicknesss on polycrystalline semi-conductor CIGS solar cell

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ABSTRACT

Solar panels can convert the sunlight into electricity, which can be used in industry and homes. Using sunlight which is solar energy is the most plenty renewable energy sources on Earth. We acknowledge the criticism regarding the lack of clarity about the innovation in our article. The main objective of this study is to provide a detailed analysis of the factors influencing the performance of CIGS solar cells, specifically focusing on the impact of temperature and thickness on efficiency of solar cell. SCAPS 1D which is a numerical simulation software that (simulating the electrical properties of thin film heterojunction solar cells, considering both DC and AC characteristics.) has been used in this study of copper indium gallium selenide (CIGS) solar cell. In this investigation, the effect of temperature and thickness on efficiency of the solar cells has been investigated. The measurements have been shown and discussed using SCAPS 1D version 3.3.11. The experiment result clearly observed that temperature and thickness diversity directly affect the efficiency of CIGS solar cell. These parameters have been used for CIGS solar cell by electrical properties of I-V measurements. These measurements provide a critical snapshot of the cell's performance under varying operating conditions. By analyzing the I-V curves at different temperatures and thicknesses, the experiment gathered empirical evidence to corroborate our theoretical understanding. The efficiency value for indium gallium selenide (CIGS) solar cell at 300K (Working point of SCAPS 1D is 300 K and this temperature serves as a convenient reference point for calibrating and comparing models, establishing a consistent starting point for further simulations.) has been found optimum value 10.88%. This study showed that as temperature and thickness rises from 200 K to 350 K and from 1 μ m to 5 μ m, the efficiencies of solar cell decrease from 17.2% to 7.5% for CIGS respectively.

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INTRODUCTION

Energy is essential for life. It is the driving force behind all physical and biological processes. Energy is the very foundation of our modern world, indispensable for powering our homes, businesses, and the vital services that keep us healthy and informed. Renewable energy is an energy which comes from natural environment, such as rain, sunlight, wind, geothermal heat, waves, tides. That is crucial since renewable energy sources can help us to meet our growing energy needs while decreasing our reliance on fossil fuels and struggling climate change. Renewable energy sources most common wind, sunlight is free all over the world, and they can be used to generate electricity 24/7. Solar panels can convert the sunlight into electricity, which can be used in industry and homes. There are many studies about solar energy.

Govindasamy and Kumar. [1] performed an experiment for rising the effectiveness of solar cells using composite materials with phase change. They examined that solar panel efficiency decreases as temperature increases. Phase change materials (PCMs) can be used to keep solar panels cool, which improves their efficiency. PCMs are substance that is able to absorb and set free heat at a specific temperature. When used in solar panels, PCMs can help to keep the panels cool, which improves their efficiency. They used four different PCMs. Paraffin jelly alone, Paraffin jelly with expanded perlite, Paraffin jelly expanded graphite, Paraffin jelly expanded vermiculite. The study found that the paraffin jelly with expanded perlite PCM was reduced the biggest surface temperature of the solar units by 10.29 °C. It also increased the electrical efficiency of the solar panels by 2.92% and the power output by 8.04%. The results showed that the paraffin jelly with expanded perlite PCM was the most effective at reducing the temperature and improving the efficiency of the solar panels.

Guesmi et al. [2] investigated CCTS-based solar cell's performance, which have received less attention than CZTS-based solar cells. The CCTS thin film was formed in substrate of glass by thermal evaporation and then operated on SCAPS 1D software to simulate the acts of a CCTS based solar cell. It was found that the CCTS film had big coefficient of absorption is 104 and energy of bandgap is 1.4 eV. This means that the CCTS film is able to absorb considerable part for solar spectrum, which leads to an improved conversion of photons into electricity and a new photovoltaic structure was modeled that included CCTS as the absorber material. The CCTS cell performed significant efficiency of conversion is 14.76% afterwards fixing of the absorber layers and buffers. This effectiveness was comparable to that of high-efficiency solar cells, which suggests that CCTS-based solar cells have the potential to be a viable technology for commercial applications.

Zhao et al. [3] showed to make high efficiency silicon heterojunction solar cells (SHJ) through united theoretical works and investigational works. They first optimized

the Si-based thin-film layers and then implemented electrodes with rised use of indium and silver. They also used up-to-date opto-electrical simulations to find clear paths for device design and experimental optimizations. They produced FBC and SHJ solar cells at the same time monofacial solar cell and bifacial solar cell the values with the best efficiencies is 24.18% and 23.25%. They introduced various articles SHJ based solar cell architectures and found that the biggest efficiency of 27.60% can be experimentally for FBC and SHJ solar cells.

Gopal and Subash. [4] investigated the performance of mono-crystalline and poly-crystalline solar panels at Hyderabad latitude 17.43° N, 78.43° E from February to May 2018 and on April 1, 2023. The authors found that the SPV panels of both materials showed an 8% increase in energy output in 2023 compared to 2018. This suggests that solar panel technology has improved over the past 5 years, and that solar panels are becoming more efficient at converting sunlight into electricity. The authors also found that the wind speed had a small but measurable effect on the output power generation from the panels. At higher wind speeds, the panels generated slightly more power. This is likely due to the fact that the wind helps to cool the panels, which improves their efficiency. Overall, the study found that both mono-crystalline and poly-crystalline solar panels are viable options for solar power generation in Hyderabad. The authors recommend that consumers choose the type of panel that best suits their needs and budget.

Kim et al. [5] have developed functional polymeric HTMs that can passivate the perovskite/HTL interface. This improved both reliability and effectiveness of the device lasting for a considerable time. Perovskite solar cells (PSCs) were hopeful future posterity solar cell technology with high level of efficiency (25.7%) contrasted to silicon solar cells. Hole-transporting materials (HTMs) were a key component of PSCs that play a vital role in transporting holes from the perovskite layer to the electrode. The most common HTM used in PSCs is spiro-OMeTAD, but it had naturally low hole mobility and required doping with Li-TFSI and tBP to improve its performance. This doping decreased the wetness and heat balance of the PSC material. To develop the reliability and effectiveness of PSC, researchers are developing new HTMs. The last studies on efficient and stable devices based on HTM using functional polymers, small molecules, polymers, spiro OMeTAD. They focused on polymeric HTM and useful polymeric HTM, which have the potential to overcome the limitations of traditional HTM. In conclusion they developed functional polymeric HTM that can passivate the perovskite/HTL interface. This improved both the long-term reliability and effectiveness of the device.

Patil et al. [6] researched that developing an air-cooling system for solar panels to decrease the photovoltaic panel's temperature. They designed and attached an experimental air-cooling setup to the backside of a PV panel. They then conducted an experiment to contrast the effectiveness of

the photovoltaic panels (PV) with the air-cooling system and without it. The study revealed that the air-cooling design significantly reduced the running temperature of the photovoltaic panel. The values of photovoltaic panel temperature was reduced by 9 degrees Celsius. They also found that the air-cooling system increased the electrical properties performance of the photovoltaic panel. Photovoltaic panel's electrical energy conversion efficiency increased from 7% to 12.6%

Khan et al. [7] examined that ground-mounted solar systems required a lot of land, while rooftop-mounted solar panels could save land. However, rooftop structures could be damaged by strong winds. This study investigated how strong winds affect rooftop-mounted solar panels using computer simulations. The simulations found that the truss and mounting structure were the most sensitive to wind damage. The truss fails at a wind speed of -15 m/s, and the mounting structure fails at a wind speed of -25 m/s. The channel and column structures were more resistant to wind damage and can resist wind speeds of up to ± 25 m/s without failing. The existing truss design is vulnerable to wind damage and fails at a wind speed of 7.53 m/s. The new truss design was more resilient and could resist winds of up to 35 m/s. The existing mounting structure was not reasonable for far-away areas with high wind speed. It failed at wind speeds above 25 m/s. The new mounting structure can resist winds of up to 25 m/s. The existing channel design is within the safe limit and does not require any changes. The existing column design was highly stable and could resist winds of up to 45 m/s.

Salehi et al. [8] showed that cooling photovoltaic panels with nanofluids is a new and effective way to improve their efficiency. The aluminum nanoparticle's effects on the conversion efficiency and cooling performance of photovoltaic panels were studied experimentally. Nanofluids which had aluminum nanoparticles, cooled solar panels the reduction value of surface temperature was 13°C compare to water cooling. Photovoltaic panels has been cooled with aluminum nanoparticles and the power harvest and performance of solar panels increased by 13.5% and 13.7%. The improvements were ascribe to more surface area available for the exchange of aluminum nanoparticles and to the efficient heat transfer enable by aluminum nanoparticles high thermal conductivity contributed to the enhanced performance. The performance of aluminum nanoparticles in various sort of coolants, solar panels and different concentrations are conduct further research. Investigate the stability of aluminum nanoparticles in liquid cooling and the effects of difference in surrounding temperature on their ability to boost conversion efficiency in photovoltaic panels. Overall, this study suggests that aluminum nanoparticles are a promising new technology for improving the efficiency of solar panels.

Urdirroz et al. [9] studied that how to better the bifacial solar unit's performance with rear cell (PERC) architecture and passivated emitter by texturing the glass surface

and the silicon surface (solar cell). Light trapping used textured surfaces to guide light deeper into the solar cell, where it can be more efficiently converted into electricity. Radiative cooling uses the cold night sky to passively cool the solar panel, which can reduce thermal losses and improve efficiency. They found that texturing both surfaces could increase thermal power and photocurrent by 15% compared to a flat surface panels. However, they also found that the back of the panel can deliver up to 30% more power, but the thermal effects depend heavily on the setting up details and the conditions of atmosphere. In short, this research showed that texturing both surfaces of a bifacial solar panel can improve its performance, but that it is difficult to achieve optimum light shielding and radiant cooling at the same time.

Alomar et al. [10] investigated the energetic, economical, and exergetic performance of photovoltaic solar systems using fixed, individual axis, and dual axis tracking systems in the location of Zakho, Iraq. Tracking system improved the exergy, thermal, and electrical performance of photovoltaic solar panels. Tracking systems reduced thermal exergy losses. Cell temperatures were higher for PV solar panels with tracking systems due to direct exposure to solar radiation. Tracking systems increased the electrical energy gain of PV solar panels, with the maximum enhancement at 8 a.m. and 17 p.m. Tracking systems increased the annual yielded energy of PV solar panels by 19.6% for individual axis tracking and 26% for dual axis tracking compared to fixed panels. The setting up costed for tracking solar panels energy system were higher than for fixed systems, but the price of energy was lower. Tracking systems reduced CO₂ emissions by 473 tCO₂, 566 tCO₂ (single-axis), and 635 tCO₂ (dual-axis) compared to natural gas and gasoline. Overall, this study suggested that tracking systems improved the performance and economics of PV solar panels, and they also reduced CO₂ emissions.

Mirjalili et al. [11] showed that buildings used a lot of energy, and commercial-office buildings were the worst offenders. To create more sustainable cities, we need to adopt energy effectiveness tactics in the construction of buildings. Solar cells could help us to reduce energy consumption in buildings. They used architectures energy softwares to investigate a merchant building based on green construction guidelines, examining nearness of electrical cars and straightforward photovoltaic panels. The study found that these structures with solar panels on the top could reduce its energy consumption by 9%, and with straightforward photovoltaic panels on the windows, the reduction could be up to 18%. The refunding for the photovoltaic panels were estimated in 5 years and 9 months for the first part and 6 years and 2 months for the second part. With incentive policies, the payback period could be reduced to 2 months and 20 days for the first part and 3 years and 3 months for the second part. Overall, this study suggests that solar panels and transparent solar cells are promising technologies for

reducing electricity consumption and costs in commercial-office buildings.

Ekinci et al. [12] studied that reducing the effectiveness of photovoltaic panels (PV) by cleaning. They used the effective of three different chemical solutions were made to clean photovoltaic panels. In this respect solution 1 was 2-propanol (5% v/v), solution 2 was ethanol (5% v/v) and solution 3 was acetone (5% v/v). First they handled a 3D printer to develop a solar panel cleaning robot after that chemical liquids were made in ideal status and used to photovoltaic panels with the cleaning robot. Studies showed that the most effective chemical solution was solution 1 which was (2-propanol), in conclusion solar panels can get dirty, which makes them less efficient. Using a cleaning robot on solar panels with a chemical solution can increase their efficiency. This method increased the power harvested from the photovoltaic panels aside 15%.

Tilmatine et al. [13] developed a cleaner gadget for photovoltaic panels. The cleaning gadget which uses corona wind generated by corona discharge plasma. The gadget was made up of a high-voltage electrode and a grounded frame electrode and also the gadget was moved with two wheels and is put a couple of millimeters over the photovoltaic panel surface. Experiment values that using sand dust showed that the gadget could achieve performance of 95% and the velocity of corona wind about 2 m/s. The pollution particles were removed from the plane of photovoltaic panels using the corona wind appeared of a 5 mm width at the bottom side of the gadget. The gadget could achieve corona wind velocity that 2 m/s and effectiveness of cleaning was 95% with 20 W used power. Overall, researchers suggest that the ionic wind solar panel cleaning device is a hopeful new technology for cleaning solar panels efficiently and with low power consumption.

Xu et al. [14] studied that solar photovoltaic panels were less efficient when their temperature was so high. Therefore, they used the way to get colder photovoltaic panels which was using the phase change materials (PCMs). Paraffin wax which is an organic phase change material (PCMs), it could cool very effectively solar photovoltaic panels and (PCMs) increased their power harvesting. Paraffin wax could lower the center temperature and the top surface of a photovoltaic panel aside 33.94 °C and also the bottom surface's center by 36.51 °C in five hours, under the condition that there is no wind, the light irradiation is 1000 W/m² and the initial environmental temperature was 7.3 °C. As a result, paraffin wax can increase the average maximum value of harvesting power of photovoltaic panels aside 1.35 W and 1.63% maximum average efficiency.

Dambhare and Moharil. [15] examined that the energy losses when the sunlight hit the solar panel. Since not all of it is converted into electricity when solar cell absorbed the sunlight. There are some energy losses that take place during this process, the most common of which is heat generation. The heat causes the solar cell to warm up, which decreases its efficiency remarkable. They discuss the various

mechanisms of energy loss in solar cells and methods for spectral correction. Spectral modification is a technique for changing the wavelength of light. Upconversion is a process in which is lower energy of two or more photons are combined to creating a higher energy photon. Downconversion involves the division of a photon with higher energy into two or more photons with lower energy. By using upconversion and downconversion, we can shift the wavelength of sunlight to match the solar cell's absorption spectrum, which can lead to increased efficiency. Overall, the spectral modification techniques that could guide to a significant reduction in the cost of photovoltaic panel technology because of spectral modification which can rising the efficiency of photovoltaic panels, that means we can generate more electricity from the same amount of sunlight. This is a promising technique for developing the effectiveness and decreasing the cost of photovoltaic panels.

Laseide and Ramere. [16] studied that solar farms are a growing source of renewable energy, but their efficiency can be reduced by high temperatures. Researchers used a method which is based on microcontroller water spraying set-up that uses a thermal check report system to optimize the process. Water spraying is a simple and effective way to cool solar panels, but it is important to optimize the process to avoid wasting water and reducing efficiency. In another words this method is a new way to spray water on solar panels that is more efficient and uses less water than previous methods. The new system uses a microcontroller to control the water flow and a temperature sensor to make sure the panels don't get too cold and this method has been shown to rise solar panel's effectiveness by over to 16.65% and decreased the construction cost.

Hatamleh et al. [17] studied that nanofluids (Nfs) are a mixture of conventional liquids and particles in the submicrometer range. They have improved properties compared to conventional liquids, particularly in terms of cooling. Nanofluids are progressively used in solar panel (SP) systems to improve efficiency. The researchers manage an experimental operation to investigate the electrical efficiency and thermal efficiency of solar panel systems with common fluids and nanofluids. They started up an artificial neural network to devise a data equation to predict nanofluids' cooling performance in solar panel systems. In conclusion is that the use of nanofluids instead of the usual working fluids can significantly improve the electrical and thermal properties of solar panels. The researchers have also developed an equation that can predict the cooling efficiency of nanofluids in photovoltaic panel systems, which can help to decrease the time and cost of future research studies.

Almarzooqi et al. [18] investigated that using of optical filters to increase the effectiveness of solar panel systems. The researchers knew that the effectiveness of photovoltaic panels related to the panel's temperature. So they used that method that using a Plexiglas box filled with water can decrease the water's temperature until 46 °C. They also

covered the top plane of the photovoltaic system that has optical filters can decrease the panel's temperature and also improves the electric harvest in conclusion that the use of Plexiglas as an optical filter has the future to rise the efficiency of photovoltaic panels. Researchers recommend that using this method in large-scale photovoltaic systems to figure out how plexiglass can increase the electrical harvest in photovoltaic centrals.

Aly and Clarke. [19] studied that using machine learning and computational fluid dynamics to calculate loads of wind on photovoltaic panels. The problem is that Windstorms can damage photovoltaic (PV) systems, most commonly single axis trackers. Currently used method is constructional codes dont perform exactly loads of wind for sun tracking systems. So the studies showed that this new approach can succumb various results from the typical method. The researchers propose stowing photovoltaic panels at a -15° angle during wind case to decrease damage. They also demonstrate that combining CFD and ML can accelerate operations (10,000 times faster) without failing. This new approach could be used to design more resilient PV systems.

Burgelman et al. [20] explain the numerical modeling requirements for polycrystalline thin-film solar cells, where deep traps and interfacial band offsets strongly affect device behavior. They implement a multilayer semiconductor model in SCAPS that solves Poisson and continuity equations while accounting for defect recombination. Simulated I-V, C-V and spectral-response curves closely match experimental CdTe and CIGS data, confirming its predictive accuracy. The study positions SCAPS as a reliable tool for thin-film solar-cell design and analysis.

Hossain et al. [21] investigate how different ETL/HTL combinations influence CsPbI₃-based perovskite solar-cell architectures. They integrate DFT calculations with SCAPS-1D and wxAMPS simulations to evaluate band alignment, carrier extraction, and material thickness effects. Screening 96 configurations, they identify ITO/TiO₂/CsPbI₃/CBTS/Au as the best structure, achieving about 17.9% conversion efficiency. The results highlight that appropriate transport-layer selection is critical for improving perovskite-device performance.

Zare et al. [22] focus on the stability requirements of free-piston Stirling engines, where maintaining a stable limit cycle is essential for continuous operation. They derive dynamic-error equations and use a practical-stability framework to determine sufficient operating conditions. The analysis provides nine parametric constraints that ensure stable oscillations during design. These findings guide engineers in selecting feasible engine configurations early in development.

Mobini et al. [23] analyze the dynamic performance of free-piston engines by comparing hydrogen with conventional working gases. They assess closed-loop poles and phase-plane trajectories to evaluate startup behavior and oscillation characteristics. Hydrogen significantly enhances

operating frequency, output power, and stability, outperforming gases like air and nitrogen. The study recommends hydrogen as an optimal working fluid when safety constraints are satisfied.

Tangestani et al. [24] develop a neural-network model to predict hydrogen generation in the water–gas–shift process. They incorporate catalytic and operating-condition inputs into the model to estimate hydrogen output accurately. The approach reduces experimental effort while maintaining high prediction capability. The results show that machine learning can capture complex kinetic behavior in hydrogen-production systems.

Bumararaja et al. [25] evaluate the thermal performance of a hybrid-nanofluid heat pipe using multiple machine-learning regression algorithms. They train models with a large experimental database covering different heat loads, flow rates, and inclination angles. The Extra-Tree regression model delivers the best accuracy, with less than 5% deviation from measurements. Their results confirm that data-driven techniques can provide reliable thermal predictions for heat-pipe systems.

Bumataria et al. [26] review recent advances in heat-pipe technologies using mono and hybrid nanofluids. They summarize fabrication approaches, nanoparticle selection, fluid concentration effects, and operational variables such as inclination and heat load. The review emphasizes significant thermal-conductivity enhancements achieved with hybrid nanofluids. Future research needs are also discussed to overcome stability and aggregation issues.

Laseinde et al. [16] examine thermal-management improvement in polycrystalline PV panels using an automated water-spray cooling system. Their microcontroller-based control strategy activates cooling to reduce cell temperature when needed. The system achieves about a 16.65% performance improvement under hot-climate conditions. Results demonstrate that cost-effective cooling can significantly boost PV electrical output.

MATERIALS AND METHODS

Numerical Methods

SCAPS 1D is a one dimensional solar cell simulation which the version is 3.3.11. This software is free to use to the Photovoltaic research association such as universities and research institutes, companies [21] [22]. The researcher can create a solar cell which can be various proterties with layers which can be seven; such as temperature, optical absorption, thickness, defect densities etc. And software is able to show that the measurements of: I-V, Q-E, C-f, C-V. Solar Cell Capacitance Simulator (SCAPS 1D) software's working fundamentals are related to steady state electron hole continuity equations and Poisson. The Poisson equation is shown in (1).

$$\frac{\partial^2 \varphi(x)}{\partial x^2} + \frac{q}{\epsilon_r \epsilon_0} [p(x) - n(x) + N_D^+ - N_A^-(x) + p_t(x) - n_t(x)] = 0 \quad (1)$$

φ represents the electrostatic potential at position x , q denotes the electron charge N_D^+ indicates the number of positively charged donor atoms per unit volume which is ionized donor concentration, N_A^- represents the number of positively charged donor atoms per unit volume which is ionized acceptor density, the terms of p and n represents separately electron density and hole density, ϵ_0 and ϵ_r stands for permittivity of free space and relative permittivity of the semiconductor material, p_t represents the trapped hole density at position x , n_t stands for the trapped electron density at position x and (x) denotes the position in the x -coordinate system.

$$-\frac{\partial J_n}{\partial x} + G - R = 0 \quad (2)$$

$$-\frac{\partial J_p}{\partial x} + G - R = 0 \quad (3)$$

Equations (2) and (3) describe the steady-state continuity relations for electrons and holes. In simpler terms, they express how the concentrations of these charge carriers vary within the material over time. Such variations arise from three main processes: the inflow and outflow of carriers, their generation under illumination, and their recombination within the semiconductor. In these expressions, J_n and J_p denote the current densities of electrons and holes, respectively, R represents recombination processes (both direct and indirect), and G indicates the carrier generation rate.

$$J_n = qn\mu_n E + qD_n \frac{\partial n}{\partial x} \quad (4)$$

$$J_p = qp\mu_p E - qD_p \frac{\partial p}{\partial x} \quad (5)$$

The symbols μ_n and μ_p represent the mobilities of electrons and holes, while D_n and D_p denote their corresponding diffusion coefficients. The term E refers to the electric field acting within the semiconductor. The overall current in a semiconductor results from the combined effects of two fundamental transport mechanisms: drift and diffusion. The drift component describes the motion of charge carriers under the influence of an external electric field, whereas diffusion accounts for their movement from regions of higher concentration to lower concentration. Equations (4) and (5) express these principles mathematically for electrons and holes, respectively.

The optical absorption constant α :

$$\alpha(\lambda) = \left(A + \frac{B}{h\nu}\right)^2 \sqrt{(h\nu - E_g)} \quad (6)$$

The optical absorption coefficient (α) is a basic property that tells how much light a material can take in at different wavelengths. In solar cell studies, α plays a key role because it directly affects how well the device absorbs sunlight and turns it into electricity.

Equation (6) shows how the absorption coefficient (α) relates to the photon energy ($h\nu$) and the material's band gap (E_g). The constants A and B change from one material to another. In simple terms, α isn't a fixed value—it moves up or down depending on the energy of the incoming photons. When the photon energy goes beyond the band gap ($h\nu > E_g$), the material absorbs light strongly, creating plenty of charge carriers. But if the photon energy is lower ($h\nu < E_g$), absorption becomes weak, and the material is almost transparent to that light. In our work, this equation was used to describe how the CIGS layer absorbs light under real conditions. Considering that α changes with wavelength made the simulation more realistic and gave a clearer idea of how light actually behaves inside the cell. This also made the overall performance analysis more dependable.

Theoretical Background

Solar cell's characteristic is simply related to the I-V graph measurements. It shows that open circuit voltage (V_{OC}), short circuit current (I_{SC}), voltage-current-power at maximum power point, fill factor (FF) and efficiency. Short circuit current (I_{SC}) is the maximum current that a solar cell can generate when its terminals are directly connected, essentially bypassing any load or resistance. In this scenario, the voltage across the cell is essentially zero. Open circuit voltage (V_{OC}) is the maximum voltage that a solar cell can produce when no current is flowing through it. It's essentially the theoretical voltage difference between the cell's positive and negative terminals under ideal conditions with no load connected. The maximum power point (P_{max}) is that the solar cell can generate its maximum power; in this case the current and voltage defines as I_{max} and V_{max} . The fill factor (FF) (7) is a parameter that in conjunction with V_{OC} and I_{SC} determines the maximum power from a solar cell. Efficiency is quantified as the quotient of P_{max} divided by the input light irradiance (E) and the surface area A_c of solar cell.

$$FF = \frac{P_{max}}{I_{SC} V_{OC}} \quad (7)$$

$$\eta = \left(\frac{FF \times I_{SC} V_{OC}}{E \times A_c} \right) \quad (8)$$

If you look at Figure 1, the I-V curve basically shows how a solar cell reacts when the light or temperature changes. It's the main way to see how well the cell works and where it delivers the most power. The shape of this curve isn't fixed — it varies with the type of technology used, whether that's a crystalline-silicon panel, a thin-film design, or some newer photovoltaic setup.

At the short-circuit point, the voltage drops to zero while the current reaches its peak. That's the maximum

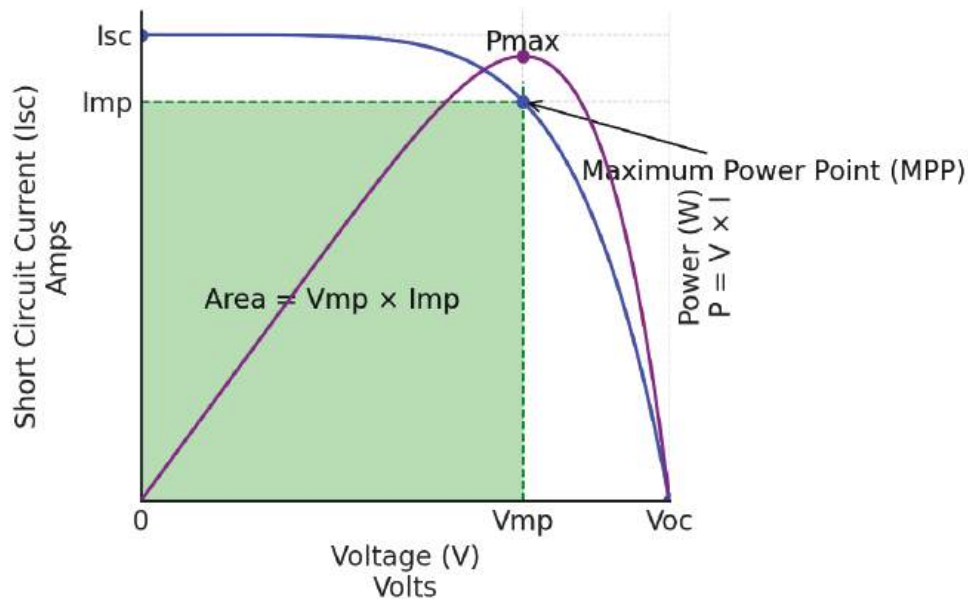


Figure 1. The solar I-V curve visually illustrates the operational behavior of a specific solar cell.

current you can get when the cell's terminals are directly linked together. In the opposite case, the open-circuit point, the current becomes zero and the voltage climbs to its highest value. This is the greatest voltage the device can produce when there's no external load connected.

Between these two limits lies what's called the **maximum power point (Pmax)** — the spot where the product of current and voltage gives the highest output. Knowing this point is crucial for figuring out how efficient the cell really is and for setting its best working conditions in practice.

Device Structure and Simulation Parameters

The development of PV technology has caused to a rise in simulation models and tools. These simulation applications can estimate and approach the simulation of monocrystalline, polycrystalline and matters based on solar cells. Among those simulation softwares, the SCAPS-1D simulator is one of the softwares with an advanced algorithm that takes into account the higher number of parameters. In this study, we used CIGS based photovoltaic panels. In figure 2 illustrates the structure's schematic employed in this simulation.

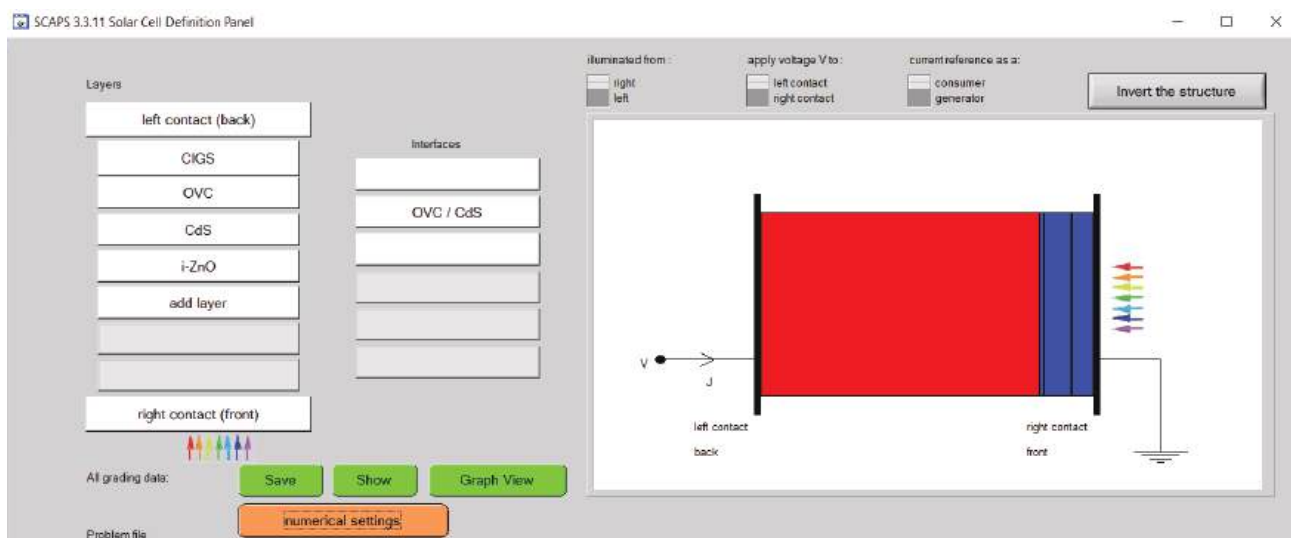


Figure 2. The model's representation that used for the Photovoltaic simulation on SCAPS 1D, four layer structures were chosen.

Table 1. Material properties data for simulating the CIGS solar cell layers

	CIGS	OVC	CdS	i-Zno
Thickness: W	1	0.015	0.100	0.080
Profil	Uniform	Uniform	Uniform	Uniform
Properties	Pure A (y=0)	Pure A (y=0)	Pure A (y=0)	Pure A (y=0)
Gap energy: E_g (eV)	1.2	1.45	2.45	3.4
Electron affinity: χ (eV)	4.5	4.5	4.45	4.55
Relative permittivity: ϵ_r (cm^{-3})	10	10	10	10
Density of state in CB: N_C (cm^{-3})	2.0×10^{18}	2.0×10^{18}	2.0×10^{18}	4.0×10^{18}
Density of state in VB: N_V (cm^{-3})	2.0×10^{18}	2.0×10^{18}	1.5×10^{19}	9.0×10^{18}
Thermale velocity of electrons: V_{thn} (cm/s)	10^7	10^7	10^7	10^7
Thermale velocity of holes: V_{thp} (cm/s)	10^7	10^7	10^7	10^7
Electrons mobility: μ_n (cm^2/V_s)	50	1	50	50
Holes mobility: μ_p (cm^2/V_s)	20	1	20	20
Density of donnors: N_D (cm^{-3})	0	10^{13}	10^{15}	5.0×10^{17}
Density of acceptors: N_A (cm^{-3})	5.5×10^{15}	0	0	0
Coefficient of absorption	sqrt(hv-Eg) law (SCAPS traditional)	sqrt(hv-Eg) law (SCAPS traditional)	sqrt(hv-Eg) law (SCAPS traditional)	sqrt(hv-Eg) law (SCAPS traditional)

The SCAPS-1D program basically works around something called a *Layers List*. It's like a stack of thin material layers that together make the solar cell. Each layer has its own job — the absorber takes in sunlight, and the contacts collect the current. When researchers change the layer settings such as thickness, doping, or defect levels, they can see how light absorption and carrier movement change. Adjusting these things helps them find a balance between better absorption and fewer recombination losses. That's how new and more efficient solar-cell structures are tested before being built.

Table 1 shows the main data used to model the Copper Indium Gallium Selenide (CIGS) cell. It lists physical and electrical properties for each layer — usually the absorber, buffer, and transparent conductor — since every one of them affects performance differently. The important numbers are the band gap, carrier mobility, lifetime, doping, and dielectric constant. For example, the band gap decides what part of sunlight the cell can absorb, while mobility and lifetime show how fast the charges move and how likely they are to recombine before reaching the contacts.

The values in Table 1 mostly come from earlier studies and standard reference data. They're not exact; they just give a realistic range. In real use, these numbers change depending on how the material is made or tested. Outdoor conditions can also shift the behavior of the cell, so the simulation results won't always match perfectly. Because of that, the parameters here should be seen as general guides, not fixed facts. As research goes on, these numbers will be updated, and the models will get closer to what really happens in working solar panels.

RESULTS AND DISCUSSION

Temperature (K) and Thickness (um) Effects

In this part of the experiment, the effects of the temperature and thickness on efficiency of the solar panel has been investigated. Table 2 demonstrated the distribution of the temperature and thickness used in this study.

To clarify the dominant parameter affecting performance, Table 2 has been examined in detail. The data obtained from Table 2, which reveals that temperature has a more significant impact on CIGS solar cell efficiency than thickness within the investigated ranges. Specifically, increasing the temperature from 200K to 350K at a constant thickness results in a substantial decrease in efficiency (around 9.7%), while varying the thickness from 1 μ m to 5 μ m at a constant temperature leads to only a minor change (around 0.02%). This highlights the critical role of temperature control in optimizing CIGS solar cell performance. Our detailed simulations, summarized in Table 2, the operated I-V parameters Jsc (mA/cm²), Voc (V), Fillfactor FF and solar cell efficiency η (%) of single shot at different thicknesses and temperatures for indium gallium selenide (CIGS) solar cell under lighting, that the CIGS solar cell's best experiment values with open circuit voltage, short circuit current density, fill factor and efficiency is 0.558731V, 31.75017336 mA/cm², 61.3, and 10.8845 % at 3.0000 μ m and 300K respectively. However, simulation show has been showed at 300K, 1.000 μ m open-circuit voltage, short-circuit current density, fill factor and efficiency is 0.558663 V, 31.68649100 mA/cm², 61.4 %, and 10.8697 % at same temperature and 2000 μ m open-circuit voltage, short-circuit current density, fill factor and efficiency is

Table 2. I-V curve parameters simulated (V), (mA/cm^2), eta and FF (%) is calculated in single shot at different temperatures (K) and thicknesses (μm) for indium gallium selenide (CIGS) solar cell under lighting

Thickness (μm)	Temperature (K)	Voc	Jsc	FF	η
1.000	200	0.811174	32.51037855	65.3878 %	17.2438 %
	250	0.685849	32.17308486	64.2102%	14.1685%
	300	0.558663	31.68649100	61.4038 %	10.8697 %
	350	0.428152	31.09036540	56.5998 %	7.5342 %
2.000	200	0.811213	32.54778994	65.3738 %	17.2608 %
	250	0.685894	32.21165389	64.1904 %	14.1821 %
	300	0.558703	31.72554985	61.3798 %	10.8797 %
	350	0.428195	31.12918500	56.5702 %	7.5405 %
3.000	200	0.811233	32.57197941	65.3559 %	17.2693 %
	250	0.685918	32.23625417	64.1689 %	14.1887 %
	300	0.558731	31.75017336	61.3562 %	10.8845 %
	350	0.428221	31.15329095	56.5444 %	7.5433 %
4.000	200	0.811235	32.59043407	65.3383 %	17.2745 %
	250	0.685938	32.25452380	64.1484 %	14.1926 %
	300	0.558752	31.76825075	61.3324 %	10.8869 %
	350	0.428243	31.17090562	56.5194 %	7.5446 %
5.000	200	0.811240	32.60452643	65.3216 %	17.2776 %
	250	0.685955	32.26887439	64.1283 %	14.1948 %
	300	0.558769	31.78243479	61.3100 %	10.8881 %
	350	0.428260	31.18473481	56.4950 %	7.5450 %

0.558703 V, 31.72554985 mA/cm^2 , 61.3 %, and 10.8797%. Buffering layer thickness caused to a few rise in efficiency as like, at 4.000 and 5.000 μm respectively open-circuit voltage, short-circuit current density, fill factor and efficiency is 0.558752 V, 31.72554985 mA/cm^2 , 61.3 %, and 10.8869 % and 0.558769V, 31.78243479 mA/cm^2 , 61.3%, and 10.8881 %. Comparing to 200K the greater values single shot at 5.000 μm is open-circuit voltage, short-circuit current density, fill factor and efficiency is 0.811240V, 32.60452643 mA/cm^2 , 65.3 %, and 17.2776 %. So these values tell us that greater values at 200K compared to 300K. The worst values compared to other temperature parameters is at 350K at 1.000 μm with open-circuit voltage, short-circuit current density, fill factor and efficiency is 0.428152V, 31.09036540 mA/cm^2 , 56.5, and 7.5342 %.The performance of the CIGS solar cell was examined under various temperature and thickness conditions. The best results were obtained at 300 K with an absorber thickness of 3.0 μm , where the device produced an open-circuit voltage (Voc) of 0.5587 V, a short-circuit current density (Jsc) of 31.75 mA/cm^2 , a fill factor (FF) of 61.3%, and an overall efficiency of 10.88%. As the temperature increased, charge carriers gained kinetic energy, which led to more frequent collisions and recombination events. This behavior caused both the current and the overall efficiency to drop. In contrast, increasing the absorber thickness improved light absorption but also

raised internal resistance, resulting in additional losses. The relatively stable fill factor across all conditions reflects the electrical stability of the simulated device. Choosing 300 K as the reference temperature in SCAPS-1D simulations was not arbitrary—it reflects a balance between practicality and performance. At this temperature and thickness (3 μm), the solar cell reached its best combination of Voc, Jsc, FF, and efficiency, aligning closely with typical room-temperature conditions. Because most experimental measurements are performed near 25 °C (≈ 298 K), the difference between simulations and real-world data remains minimal. Even slight deviations—on the order of only a few millivolts in Voc—do not significantly affect accuracy. Thus, 300 K serves as a reliable benchmark for both modeling and laboratory validation. The nearly constant fill factor of about 61% further confirms that the CIGS device maintains electrical consistency across these operating conditions.

When you look closely at the results, it becomes clear which factors shape how the device behaves. The interaction between temperature and thickness is especially important because it lets engineers tweak the design of CIGS cells for better output without pushing up manufacturing costs. Temperature changes affect carrier mobility and recombination, while thickness alters how light is absorbed and how much internal resistance builds up. Getting the balance right between the two is what makes the cell perform

well. Interestingly, the data around 300 K fits real industrial conditions quite well, so it serves as a good and repeatable reference point for optimization.

Efficiency, noted as η , is still the simplest way to judge how well a solar cell converts sunlight into power. When the cell heats up, η usually drops because the current-voltage (I–V) curve starts to distort. Higher temperatures lower both V_{oc} and I_{sc} , which means less power overall. The effect of thickness is a bit more complicated. A thicker absorber grabs more photons and raises current, but it also adds resistance and slightly pulls voltage down. Thin layers do the opposite — better voltage, weaker current. So, finding a middle ground between the two is what really pushes efficiency upward.

In general, how temperature and thickness interact sets the limits of what a CIGS device can do. The simulations here show that decent efficiency can be reached even with minimal material use, as long as structural and thermal settings are properly adjusted. The absorber layer matters the most: if it's thinner than about 3 μm , it doesn't take in enough light; if it's thicker, resistance grows and carrier flow slows down. Around 3 μm seems to be the sweet spot, since making it thicker doesn't bring much benefit.

The band gap of the absorber also plays a big role. Making the band gap wider cuts down photon absorption and current, but it slightly raises V_{oc} because voltage scales with band gap energy. In this case, tuning the band gap carefully brought the efficiency up to roughly 17.2%, showing that band-gap engineering can really boost CIGS performance when done within realistic limits.

Overall, these findings point toward a clearer way to design efficient yet affordable CIGS solar cells. By understanding how temperature and thickness work together and adjusting material properties accordingly, it's possible to push the technology much closer to its theoretical best while still keeping production practical and cost-effective.

CONCLUSION

The CIGS solar cell was comprehensively simulated and analyzed using SCAPS-1D (version 3.3.11) under varying temperature and thickness conditions, as detailed in Table 1 and Figure 2. Among all the cases examined, the most favorable performance was achieved at 300 K with a layer thickness of 3.0 μm . Under these conditions, the device reached a short-circuit current density (J_{sc}) of 31.75 mA/cm², an open-circuit voltage (V_{oc}) of 0.5587 V, a fill factor of 61.3%, and an efficiency of 10.88% under illumination. These results clearly indicate that both temperature and absorber thickness play decisive roles in defining the electrical response of CIGS solar cells.

To ensure the validity of our findings, we compared them with the experimental and numerical studies by Laseide and Ramere (2021) and Patil et al. (2023), both of which investigated similar temperature- and thickness-related effects on photovoltaic efficiency. The observed

patterns were consistent. For instance, our simulations showed a noticeable drop in efficiency—from 17.2% to 7.5%—as the temperature increased from 200 K to 350 K, whereas Laseide and Ramere reported a comparable but smaller decline (from 14.8% to 13.5%) over a 7 °C temperature rise. This alignment not only supports the reliability of our simulation but also situates our work within a broader, experimentally validated context. By contrasting our results with those in the literature, we emphasize both the robustness of our data and the distinct contribution of this study to ongoing CIGS research.

The results show that pushing either temperature or layer thickness too far hurts the solar cell's performance. When the temperature rises too much, carrier recombination speeds up; when the absorber layer gets thicker, more defects form and internal resistance goes up. Both effects lower the open-circuit voltage (V_{oc}) and short-circuit current (J_{sc}), which together drag down overall efficiency (η). To avoid that, the fabrication steps have to be carefully tuned—especially the absorber thickness and processing temperature. Keeping that balance not only improves efficiency but also makes the device more durable and easier to reproduce in manufacturing.

Our efficiency values are close to those reported in earlier work on polycrystalline cells [17], but this study looks at the combined effect of temperature and thickness over a wider range (200 K – 350 K and 1 μm – 5 μm). The data make it clear that both factors depend on each other: temperature has the stronger impact, while thickness changes the outcome more subtly but still measurably. The best results showed up again near 3 μm , confirming that making the layer any thicker adds little or no benefit.

Overall, these findings give useful direction for improving how CIGS solar cells are built and operated. Using simulation tools such as SCAPS-1D helps make sense of the complicated behavior inside multilayer solar devices. This kind of modeling links theory and real experiments, helping researchers predict performance more accurately. In the end, it can cut testing costs, speed up material development, and support cleaner, more efficient, and affordable photovoltaic systems.

NOMENCLATURE

CIGS	Copper Indium Gallium Selenide
V_{oc}	Open-Circuit Voltage, V
J_{sc}	Short-Circuit Current Density, mA/cm ²
FF	Fill Factor, %
η	Efficiency, %
T	Temperature, K
μm	Micrometer
SCAPS	Solar Cell Capacitance Simulator
ZnO	Zinc Oxide
CdS	Cadmium Sulfide
Mo	Molybdenum
ITO	Indium Tin Oxide

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.

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Research Article

Experimental investigation of methyl-ester soybean biodiesel: performance and emission characteristics as an alternative diesel fuel

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ABSTRACT

Due to the higher demand for renewable and sustainable energy sources, it is very important to investigate fuels such as soybean biodiesel. This study focuses on a compression ignition diesel engine's performance characteristics and emissions when running on soybean biodiesel (B100 with 5% ethanol) in comparison to normal diesel fuel. The hypothesis is based on soybean being a biodiesel that can perform reasonably close to normal diesel engine with low emissions. Key performance variables such as indicated power, brake horse power, frictional power, thermal efficiencies, mechanical efficiency and specific fuel consumption were obtained during the experimental analysis as depicted in the research methodology. Emissions of CO, HC, CO₂, O₂, and NO_x for the diesel fuel and for various biodiesel blends (B25, B50, B75) were also studied. Findings indicated that soybean biodiesel had comparative fuel economy and power generation as diesel, and CO emission was reduced by 40, while NO_x emissions increased by 48 percent at full load condition. This studies clearly show that it is easy to use soybean oil based biodiesel as a renewable eco-friendly alternative biodiesel for compression ignition engines with much more improvement opportunities in terms of emissions performance. The originality of this work resides in the broader scope to evaluate the performance, emissions and trade-offs of soybean biodiesel blends and prospects that the literature on the sustainability of biodiesel is expanding by bringing in such new perspectives on it. Through the integration of experimental results, relevant performance as well as environmental assessments, this study extends previous endeavors and provides concrete recommendations for researchers, industry players and policy-makers willing to contribute to the advancement of alternative fuels.

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INTRODUCTION

Background and Importance of Soybean Biodiesel

Due to its convenience and a great source of lipids, soybean oil has become an excellent feedstock for biodiesel production. The use and the production of biodiesel from soybeans increases energy independence and pushes for environmental friendly policies through less emissions of greenhouse gases (GHGs). According to recent estimates, U.S. soybean oil production's carbon footprints have been reduced by 22% demonstrating that the oil is a suitable low-carbon feedstock for clean fuels [1-3]. Comparative life-cycle studies demonstrate that with reasonable estimates based on production practices and geographical location, biodiesel from soybean oil is expected to emit 40% to 69% fewer GHGs until 2040 [4]. Further, as new technologies in biofuel production are being developed such as superior transesterification processes and better catalysts, soybean biodiesel is being made more efficient and sustainable all the time. This enables wider acceptance since it leads to increased outputs and decreased costs of production [5]. In addition, the rising interest in biodiesel has led to increased soybean growing and it is anticipated that 14 billion pounds of soybean oil will be used in biofuel production during the period of 2024 to 2025 [6]. But on the other hand, it is also important to think about the land-use change and the GHG emitting consequences that ever increasing soybean farming may have. Such studies show that unless sustainable agricultural practices are put in place, increased production of soybeans will result in more emissions [7]. This study thus aims at investigating the effects of differing reaction times on oil yields when using non-edible oils such as waste cooking oil, rubber seed oil and *Jatropha curcas* linn seed oil in biodiesel synthesis. Looking at these issues will bring forth a revelation concerning why soybeans can stand out as a food security crop even in areas where other food crops cannot grow properly, let alone produce enough to be sold or consumed by people living in those regions [8]. The soybean bio-diesel is one possible solution towards achieving this goal, provided it ends up replacing petroleum diesel fully since it would not burn sulfur-containing compounds in our atmosphere like does ordinary diesel fuel. We strive for a cleaner energy future; soybean biodiesel stands out as a promising contender within this transition period for a more earth-friendly and greener power landscape.

Scope of the Study

As the demand for biodiesel rises, there is an ongoing development of quality standards and specifications that impose restrictions on permissible levels of contaminants in biodiesel. Key elements influencing the economic feasibility of biodiesel production comprise plant capacity, feedstock oil prices, and glycerine and biodiesel yields. An operation with an annual capacity of 100,000 tons demonstrates economic viability, displaying increased ARR and NNP alongside a reduced BBP. Enhancing capacity using

soybean oil or waste cooking oil maintains economic feasibility, supporting the future promotion of biodiesel production by reducing feedstock oil costs and increasing valuable product yields in large-scale plants [9]. Biodiesel indicated a lower exhaust emissions but CO₂ emission slightly higher with the exception of NO_x according to M. Canakci [10]. The potential for lower oxidation stability and lower performance in cold weather due to higher cloud and pour points needs to be improved [11-12]. The improvement can be done by using some additive for cold flow improvement, a slight blend of petro-diesel during winter, the winterization process, hydrotreating technology, or the use of ethanol or methanol blends [13-14].

Soybean Biodiesel is derived from the chemical reaction between soybean oil, methanol, and a catalyst, resulting in the biodiesel manufacturing process (fatty acid methyl esters) and glycerin as a secondary product. Its compatibility with unmodified diesel engines and potential substitution for petroleum-based diesel addresses the growing demand for alternative, biodegradable, and economically viable fuel sources [15-19]. Amid global apprehensions regarding the exhaustion of petroleum reserves, the intensification of global warming, and environmental challenges, there exists an urgent requirement for sustainable alternatives. Recognized as a clean, green, and renewable fuel, biodiesel emerges as a prime substitute for traditional diesel, aiming to reduce dependence on pollution-inducing fossil fuels and mitigate environmental impact. The agricultural soybean yield serves as a valuable protein source as well as the primary feedstock in the production of biodiesel. The use of soybean oil, an abundant renewable resource, makes up 73% of biodiesel and finds applications across various industries [20-21]. Additionally the use of soybean oil as a biodiesel feedstock is novel in the sense that methyl esters are prepared from the oil on a large level. Lapin described the problematic clarification processes of unevenly heated biodiesel produced from the microalgae *Spirulina* in a novel and simple way. Most microalgae lipids that can convert to methyl esters, are thermally stable. Therefore, external heating gently and evenly solves the heating issue, while microalgae that remain unutilized can solve the lack of heating.

Diluting with hexane improves the separation of crystals from liquid methyl esters, offering potential benefits for biodiesel production [22]. In contrast to other oil crops, soybeans exhibit greater harvest and contain approximately 19% of oil content.

The rising demand for energy compels the investigation of alternative energy sources such as biofuels and methods for energy conservation. Chand et al. present a diesel engine-driven combined heat and power system utilizing various blends of biodiesel as fuel [23]. The findings indicate that Eureka Sativa oil biodiesel exhibits the highest brake thermal efficiency and the lowest brake specific fuel consumption compared to all other biodiesel blends. Sekharraj et al. investigate the application of algae-derived

biodiesel, including microalgae oil, in the production of biodiesel [24]. The study also compares Bi₂O₃ nanoparticles with additives to biodiesel fuel blends which have better results in terms of performance and emissions. The study conducted by Rangasamy and co-workers and Ahmadbeigi et al. presents the use of *Chlorella emersonii* methyl ester (CEME) which is a common fresh water green algae growing in Indian subcontinent. Biodiesel presents considerable potential as an energy source, attributed to its renewable characteristics, lower emissions, and versatility in production from diverse materials [25–26]. Sonawane et al. discussed different ethanol-diesel blends, including those close to 5%, noting emission benefits like reductions in particulate matter and smoke [27]. According to Salmani and the work of, 5% ethanol blends offers increased combustion efficiency driven by combustion and emission reduction [28].

Through the use of lipases as biocatalysts, Lloyd and other authors demonstrated the ease and accuracy of the conversion of triglycerides to alkyl esters. Assessments on the efficiency of various lipase biocatalysts revealed a significant difference in conversion potency. These results provide fundamental insight towards building green pathways to large-scale biodiesel production [29]. Schumacher and other authors observed a reduction to the total amount of hydrocarbons emitted, together with the particulate matter and carbon monoxide but more NO_x [30].

Alcantara et al., explored three fatty materials—soybean oil, used frying oil (a 50% mixture of olive and sunflower oil), and tallow. These materials were subjected to transesterification with methanol to generate biodiesel. The cetane indices, determined by ASTM methods based on density and atmospheric distillation, matched those of conventional petrochemical diesel. Utilizing biodiesel like Soygold utilized as a blending component for middle distillate fuels offers numerous advantages. Compliant with or exceeding of ASTM specifications, it provides beneficial synergistic effects when combined with stable and unstable middle distillate fuels. Blended insoluble at 10–20% show stability in storage for at least one year and pass the ASTM filterability test [31]. While the failure of the ASTM filterability test in pure soy-derived diesel is attributed to the presence of polar species, it's important to note their tendency to clog such filters. Additionally, esters in biodiesel enhance lubricity, foster positive synergies, act as an additional oxygen source and interact with fuel system components [32].

Candeia et al. identified that the blend ratios of soybean biodiesel with diesel depended on the biodiesel proportion and the primary alcohol size in the ester. Careful consideration should be taken with diesel engines as increasing concentrations of biodiesel may impact engine performance and operation. All in all, biodiesel-diesel blends are a practical and reasonable alternative to fossil fuels [33]. Additionally, Ozener et al. analyzed the combustion, performance and emissions of conventional diesel and blends with soybean oil derived biodiesel (B10, B20, B50, B100).

They showed that lower heating value of biodiesel could be responsible for the 1–4% decrease in torque and 2–9% increase in brake-specific fuel consumption (BSFC). They noted significant reductions in carbon monoxide (CO) and unburned total hydrocarbons (THC) emissions but higher emissions of NO_x and CO₂ [34]. They found that biodiesel decreases the diesel ignition delay time and the premixed combustion phase duration, which means, it's a promising alternative fuel for the engine which is less harmful to the environment. Improved sustainability practices, such as crop rotation and reduced tillage, are necessary for environmentally responsible biodiesel. Addressing NO_x emissions may require new injection strategies. Bioenergy production, especially biodiesel, faces sustainability challenges. Despite being a crucial energy resource for global fuel diversification, increased efficiency, and pollution reduction, concerns arise. According to the eligible content, the caused concerns can include sustainability, socio economics, and environment impacts such as deforestation, biodiversity loss, and health-related concerns in the case of soybean, which is one of the key biodiesel feedstock. South American soybean production schemes, mainly in Argentina and Brazil, sound the alarm about soil fertility exhaustion, agrochemical poisoning, and negative burdens for the society. Improved sustainability practices, such as crop rotation and reduced tillage, are necessary for environmentally responsible biodiesel.

Nevertheless, the current projection indicates the extinction of soy biodiesel by 2050, pointing towards the necessity of the range of feedstock crops to support the biodiesel industry sustainably [35]. Can et al, showed that incorporating diesel with soybean biodiesel and EGR used in a diesel engine improves the combustion characteristics of the diesel and helps in the reduction of NO_x emission while maintaining the engine performance [36]. These results demonstrate soybean biodiesel's advantage as a renewable substitute for diesel engines and its role in protecting the environment.

Cesar et al discusses the competitiveness of 'social soybeans' for biodiesel production in Brazil. It explores the social, economic, and political factors impacting the supply chain and offers strategies to strengthen the position of subsistence farmers [37]. While Zhu et al. [38] analyze the socio-economic consequences of biodiesel and the potential of biodiesel for inclusive growth. Zhu et al introduces a CaO/Ag nano-catalyst for the transesterification of soybean oil and proves that it has better catalytic activity than the isolated catalysts. With optimized conditions, this catalyst reaches a very high biodiesel yield of $90.95 \pm 2.56\%$, which is a substantial improvement compared to only using CaO. Improved reaction kinetics, thermodynamics, and economics as articulated in the kinetic and thermodynamic studies, demonstrate the growing potential of CaO/Ag for biodiesel catalyst and the catalyst's promise for the industry biodiesel production. The studies were conducted with fresh and used soybean oils into biofuel free of the wax

compounds. The new oil had a higher oleic acid content, and the results indicated variations in peroxide values and free fatty acid concentrations [39]. A green heterogeneous catalyst was developed by transesterifying soybean oil to produce biodiesel. Transesterification was sped up by the catalyst, which was made from *Parkia speciosa* biomass and exhibited strong active sites for calcium and potassium. The catalyst was a good substitute for traditional homogeneous catalysts since it maintained its heterogeneity even after five cycles and produced the maximum biodiesel production of 96.4% at 40 °C [40]. Agbulut et al., produced alternative diesel engine fuels from waste soybean oil. Through a transesterification method, hybrid catalysts such as Ni-doped ZnO, Alkyl-celite, Poly-acrylonitrile nano-fibrous, and Ironoxide are used to make the biodiesel. Blends with diesel were made using the waste soybean oil (BDWSBO) with the highest yields. With 30.85% BTE and 17.96% less smoke opacity than the 25% diesel fuel with 75% IO-II-used BDWSBO blend, it is advised as an alternate fuel for diesel engines [41].

Research Gap and Novelty

So far, existing literature reveals limited investigations on soybean biodiesel, with only a few researchers exploring its performance parameters in four-stroke diesel engines. Some aspects remain unclear, such as the few low content biodiesel blends with petroleum diesel studies. To fill these gaps, the current work examines performance curves, power variation exhaust emissions, and fuel consumption fuel using four blends of soybean biodiesel and 5% (V/V) ethanol, known as B25, B50, B75 and B100. This investigation aims at updating this information by examining the technological features of soybean biodiesel production and analyzing its performance as well as its environmental sustainability. This study, by examining of soybean biodiesel, could present an insightful side for researchers, policy makers and industry that want to produce the sustainable and a cleaner substitution energy.

MATERIALS AND METHODS

Preparation of Soybean Biodiesel: Transesterification Process

FAME, or fatty acid methyl esters, are long-chain mono alkyl esters that are created when oils and fats are converted to biodiesel is justified for bio-diesel blends production. In such process 100 parts of oil or fat and 10 parts of a short-chain alcohol such as methanol are reacted in the presence of a catalyst. Sugar glycerin is generated as a co-product. A common process for making biodiesel from vegetable and animal fats is transesterification. Different methods, including as solvent extraction, supercritical fluid extraction, transesterification, and mechanical presses, are used in the conventional biodiesel production process to extract and convert oil from soybeans. It includes lowering

the viscosity of fats or oils by reacting them with alcohol to create glycerol and alkyl esters. Selecting the right alcohol and catalyst is essential; methanol is a common option because of its low cost and several chemical and physical benefits such as low viscosity (0.54 cP at 25 °C), high reactivity, and low carbon emissions (20-50%) with congenital fuels [42]. Triacylglycerol esters of fatty acids with glycerol make up the majority of vegetable oils.

Description of the Transesterification Process

The transesterification process covers the following procedures:

(a) Reaction Mechanism: In this process, soybean oil which contains triglycerides is reacted with methanol with the use of an alkali catalyst such as sodium or potassium hydroxide. Such a process produces methyl esters (biodiesel fuel) and glycerol as a byproduct.

(b) Optimal Reaction Conditions:

- **Temperature:** The process is usually performed under a temperature of 60 °C which is sufficient to ensure that optimal yield results, while energy requirements are minimized.

- **Time of Reaction:** A time period extending between 1-2 hours is sufficient for the conversion of the triglyceride oils to methyl esters.

- **Alcohol-to-oil:** In most cases a ratio of alcohol-to-oil of 6:1 (methanol to oil) is used to complete the reaction.

(c) Purification: After the reaction, the closely mixed solution is allowed to sediment. The layer containing biodiesel is separated from the glycerol layer. The biodiesel obtained is then washed with water to remove remaining catalyst or impurities, followed with a drying process until the required level of purity is achieved.

Figure 1 displays the basic flowchart of the entire manufacturing process.

Formulation of Blends (B25, B50, B75, B100)

After preparation of soybean biodiesel from the transesterification process, four different blends are prepared in the laboratory. These blends are, namely, B25 (25% biodiesel + 70% petro-diesel + 5% ethanol), B50 (50% biodiesel + 45% petro-diesel + 5% ethanol), B75 (75% biodiesel + 20% petro-diesel + 5% ethanol), and B100 (with 5% ethanol). An additive of 5% ethanol is mixed to keep the charge temperature lower than its usual level and to drop the temperature of the combustion chamber, as higher temperature enhances NO_x production in the exhaust [28]. A 5% ethanol blend strikes a balance by improving properties like viscosity and flash point without significantly compromising the fuel's energy content or causing phase separation issues [43].

Experimental Setup

The experimentation targeted a four-stroke, 5 H.P single-cylinder diesel engine, shown in Figure 2(a), which underwent trying out with various fuel combos. These included petroleum-derived diesel, B100 (with a 5% ethanol

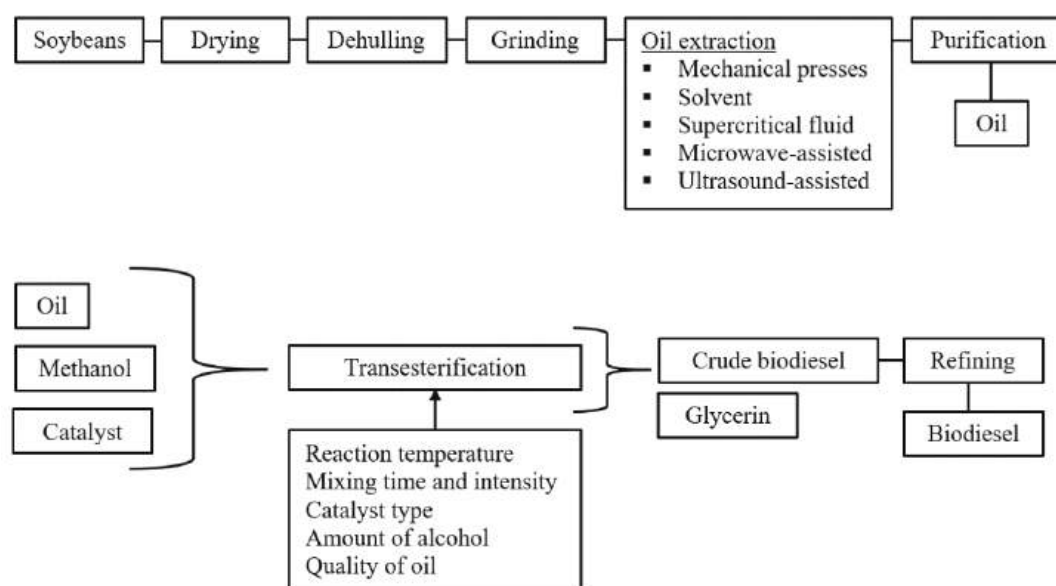


Figure 1. Basic flowchart of the overall manufacturing process of soybean biodiesel.

addition), and various biodiesel blends, namely B25, B50, and B75. In such process 100 parts of oil or fat and 10 parts of a short-chain alcohol such as methanol are reacted in the presence of a catalyst. Sugar glycerin is generated as a co-product. To control the charge temperature and to reduce the combustion chamber temperature, a 5% ethanol+diesel blend was homogeneously mixed into all the biodiesel blends. This tactical action was to avoid the charge temperature becoming higher than the norm and to reduce the chamber temperature. The higher temperatures in the combustion chamber are one of the causes for higher NO_x production in the exhaust. The biodiesel blends used in the experimental section were carefully prepared, and have the following mixtures: B25 (25% biodiesel, 70% petro-diesel and 5% ethanol), B50 (50% biodiesel, 45% petro-diesel and 5% ethanol) and B75 (75% biodiesel, 20% petro-diesel and 5% ethanol).

Engine Specifications and Testing Conditions

The diesel engine employed in this case study is a single cylinder four stroke engine commonly used and well-accepted in research due to the reliability, low cost, and availability. Such engine allows for the precise parameters to be controlled and measured such as torque, fuel intake, and emissions of the engine. Moreover, its small scale and standardized design ensure reliability of replication and comparison.

Taking into account the small-scale industries, the engine's capacity of 5 H.P can be ideal for them, thus making the results applicable for real time scenarios. The testing conditions alongside fuel blend and variable loads were practical and useful for the analysis of the engine in different scenarios. Moreover, a rope brake dynamometer is being utilized in this case for the accurate measurements of the brake horse power.

Experimental Procedures

Before beginning the experiments on the four-stroke, 5-horsepower single-cylinder diesel engine, we fill the engine fuel tank with the necessary amount of fuel and check its level. The lubricating oil level is checked. The cooling water is supplied to the calorimeter. We start the engine and then supply the cooling water. It is loaded slowly using the brake drum dynamometer, and respective quantity values are noted during testing, such as brake drum loading, engine speed, and time taken for 10 cc of fuel consumption. The parameters are obtained by using a set of calculations that are explained in the result and discussion part. The engine's specifications are as follows:

Equipment Used (Saybolt Viscometer, AVL Gas Analyzer)

The essential fuel characteristics, encompassing Fire and Flash points, as well as Kinematic viscosity, were evaluated using the Saybolt viscometer, as shown Figure 2(b)

Table 1. Engine specifications

Number of cylinders: Single
Cooling system: Water-cooled
Loading device: Rope brake dynamometer
Brake power: 5 horsepower (H.P)
Bore diameter: 80 millimeters
Stroke length: 110 millimeters
Engine speed: 1500 revolutions per minute (rpm)
Compression ratio: 16:1
Air orifice diameter: 24 millimeters
Brake drum radius: 165 millimeters



Figure 2(a). Experimental setup: four stroke, 5 H.P single cylinder diesel engine. (It was drawn by the author).



Figure 2(b). Saybolt Viscometer.



Figure 2 (c). AVL gas Analyzer readings.

and the Pensky-Martin apparatus for various fuels, including biodiesel and its blends. The rationale for performing experiments on the flash and fire point lies in the chemical decomposition of fuel when exposed to elevated temperatures. This process results in the breakdown of hydrocarbons, leading to the formation of volatile combustible gases. “The flash point of a substance is the lowest temperature at which it can vaporize to form an ignitable mixture in the air. “Fire point is the point which is the lowest temperature at which the vaporized substance may keep on burning if the fire is ignited,” on the opposite of the flash point. These features are necessary for the explanation of flammability and safety aspects of the application so the fuels are suitable for use in other fields [44-45].

Knowledge of the flash and fire points is important when it comes to fuel handling and storage safety, ensuring the observance of the set safety requirements. It is a must

to test the fuel according to the procedures developed by the American Society of Testing Materials (ASTM) before using it in the internal combustion engines. According to the ASTM demands, the flash point of B100 (Biodiesel) shall not exceed 150 °C, except in the case that the oil used for transesterification and the catalyst are employed. In contrast, the typical range for diesel fuel ranges from 60 °C to 80 °C. The index of viscosity refers to the magnitude of the difference in resistance among layers of fluid in motion, and is among the most important parameters. ASTM has set kinematic viscosity at a certain level where the aims are zeroing the possibility of the injector being clogged and measuring the flow rate unambiguously in engines. Be that as it may, compliance to these viscosity standards is absolutely necessary for the utilization of the fuel in the internal combustion engines in an optimal way, thus a probable guarantee of efficient and trouble-free engine performance.

The AVL Gas Analyser, as shown in Figure 2(c), is an auto-calibrated machine that has a series of filters fitted inside and sideways to predict the amount of exhaust gases entering from the exhaust pipe. It has a long tailpipe attachment to the main unit, of which one end has to let through the exhaust passage of the diesel engine and the other to the unit. As the exhaust gases pass through the pipe fitted with filters, they reach the sensors, where each sensor calibrates the presence of different emissions and shows the results on the monitor. The unit is equipped with three types of filters for measuring purposes: one is installed along the tail pipe, a second circular paper filter is fixed to the front of the unit, as shown in Figure 2(c), and a third type filter is fixed to the bottom of the unit. As the gas passes through these series of filters, sensors are fixed along the way to measure different emissive gases.

Engine testing was conducted in accordance with ISO 8178 standards for exhaust emissions and ASTM D6751 standards for biodiesel quality. Key tests were performed in compliance with ASTM D445, D613, and D93. Calibration methods of the instruments were described, as which could guarantee the measurements repeated and precisely. These additions provide a clearer context for the reliability and rigor of the experimental approach.

Procedure of Heat Balance Analysis

Engine is supplied with the fuel. The coolant generally water is made to pass through the water jacket of the engine. The engine parameters are noted on the application of the load over it, starting from no load condition till the maximum load that could be applied on it. The parameters such as time for consumption of the 10cc of fuel is noted along with the inlet and outlet temperature of the cooling water, Exhaust gas temperature are noted using the thermal sensors placed at strategic points in the system.

Mathematical formulation involved in tabulation of performance parameters:

- Torque produced (T)** = $w \cdot r$ Newton-meter
Where $w = m \cdot g$; m = load applied using rope brake; g = acceleration due to gravity taken as 9.81 m/s^2 ;
 r = brake drum radius of 165mm.
- Brake Power (B.P)** is the power developed by an engine at the output shaft.
 $B.P = 2\pi NT/60000$ Kilowatt
Where N is speed in rpm; T is torque produced;
- Mass fuel consumption (MFC)** = $(a \cdot \text{specific gravity}) / (1000 \cdot t)$ kg per second
Where a is 10cc of fuel consumed; t is time taken for consumption of 10cc;
Specific gravity of biodiesel is 0.88 kg per liter and diesel is 0.85kg per liter.
- Specific fuel consumption (SFC)** is mass fuel consumed per brake power produced = $MFC/B.P$
Units is kg per second per kilowatt
- Indicated power (I.P)** is the total power by the combustion of fuel in the combustion chamber.

I.P = Brake power + Frictional power

Frictional power is got by plotting graph between MFC vs. B.P

6. Mechanical efficiency (η_{mech}) = (Brake power/indicated power) * 100%

7. Brake Thermal Efficiency (η_{bth}):

The ratio of the Brake power produced to the total heat supplied by the combustion of fuel.

Brake thermal efficiency (η_{bth}) = Brake power/ Q ;

Where $Q = MFC \cdot CV$; CV is the calorific value of the fuel; For Diesel: $CV = 44,800 \text{ KJ/Kg}$

8. Indicated Thermal efficiency is the ratio of the indicated power to the total heat supplied by the fuel.

Indicated thermal efficiency (η_{ith}) = Indicated power/ Q .

Mathematical formulation involved in tabulation of heat balance analysis:

- Brake power (B.P)** = $(2\pi NT \cdot 3600) / 60000$ kilojoules per hour
Where T is torque produced = $mg \cdot r$; m is load applied; g is acceleration due to gravity;
 r is Brake drum radius
- Mass of water (mw)** during 10cc consumption of fuel = $3600/t$ kilogram per liter
Where t is time taken for 10cc consumption of fuel; $t = 88 \text{ sec/l}$
- Head of air (H)** = $\{(h_2 - h_1) \cdot \text{density of water}\} / \text{density of air at RTP}$
Units is meter; Density of water (ρ_w) = 1000 Kg/m^3 ;
Density of air (ρ_a) = 1.29 Kg/m^3
- Mass of air (ma):** $V_a \cdot \text{Density of air at RTP} \cdot 3600$
Where $V_a = C_d \cdot A \cdot \sqrt{2gH}$; C_d is coefficient of discharge of 0.6;
 A is area of the orifice with a diameter of 15mm = $1.76 \cdot 10^{-4} \text{ m}^2$
Unit is kilogram per hour.
- Mass of the exhaust gases (meg)** = $m_a + \text{T.F.C}$
Units is kilogram per hour; T.F.C is total fuel consumption = $(\text{specific gravity} \cdot 3600 \cdot 10) / (1000 \cdot t)$;
Specific gravity of Biodiesel is 0.86 kilogram per liter;
Specific gravity of Diesel is 0.83 kilogram per liter; t is time taken for 10cc of fuel consumption.
- Heat supplied by the fuel Q_s** = $\text{T.F.C} \cdot CV$
Unit is kilojoules per hour: CV for Biodiesel is 37.015 megajoules per kilogram and for Diesel is 44.8 megajoules per kilogram.
- Heat carried away by cooling water (Q_{cw}):**
 $Q_{cw} = M_{cw} \cdot C_{pw} \cdot \text{temperature difference between outlet and inlet}$
Where, C_{pw} is the specific heat of water is 4.18 kilojoules per kg-Kelvin
- Heat carried by the exhaust gases (Q_{eg}):**
 $Q_{eg} = M_{eg} \cdot C_{peg} \cdot \text{temperature difference between outlet and inlet}$
 C_{peg} is the specific heat of gases is 1.005 kilojoules per kg-Kelvin

9. Heat loss unaccounted (Q_{un}):

$$Q_{un} = Q_s - \{Q_{cw} - Q_{eg} - Q_{bp}\}$$

10. % of Heat supplied by the fuel, shared among all as:

$$\% \text{ of Heat utilized for B.P} = (B.P/Q_s) * 100$$

$$\% \text{ of Heat lost due to coolant} = (Q_{cw}/Q_s) * 100$$

$$\% \text{ of Heat lost due to exhaust gases} = (Q_{eg}/Q_s) * 100$$

$$\% \text{ of Heat loss unaccounted} = (Q_{un}/Q_s) * 100$$

11. Temperature legends followed:

T1 = Room Temperature; T2 = Water Inlet Temperature;

T3 = Water Outlet Temperature;

T4 = Exhaust Gas Temperature.

RESULTS AND DISCUSSION**Estimation of Fuel Properties**

This study investigates the kinematic viscosity of various blends of soybean biodiesel at a temperature of 40 °C, Table 2. A stopwatch is employed to measure the duration needed to gather 60 CC of biodiesel and its various blends. The collection times recorded were 33 seconds for B25, 34 seconds for B50, 35 seconds for B75, and 36 seconds for B100.

The kinematic viscosity values recorded were 1.35 centistokes, 1.74 centistokes, 2.12 centistokes, and 2.50 centistokes

for B25, B50, B75, and B100, respectively. The Saybolt viscometer and the Pensky-Martin apparatus are employed to assess the kinematic viscosity, flash, and fire points of various soybean biodiesel blends. The flash points recorded were 38 °C, 42 °C, 54 °C, and 74 °C for B25, B50, B75, and B100, respectively, Table 3. The corresponding calorific values were 42.19 MJ/kg, 40.34 MJ/kg, 38.49 MJ/kg, and 37.01 MJ/kg. The fire points identified for B25, B50, B75, and B100 were 58 °C, 62 °C, 80 °C, and 110 °C, respectively.

The observations reveal a correlation among the share of biodiesel inside the blends (starting from B25 to B100) and an increase in kinematic viscosity, representing the ratio of dynamic viscosity to the density of fluids. This located trend aligns with ASTM standards specifying that the kinematic viscosity of biodiesel must fall inside the variety of 1.90 to 6, a criterion met through the values observed from the experiments.

On the contrary, the flash and fire factors show off a decline from B100 to B25 because the diesel content material inside the blends increases. Precise readings for flash and fire points have been systematically recorded at 2 °C periods for the duration of engine performance tests carried out on the laboratory. Table 4 summarises the measured properties of biodiesel blends.

Table 2. Tabulation of kinematic viscosity for biodiesel and its blends

Biodiesel blend	Temperature (°C)	Time for collection of 60 CC of fuel (seconds)	Kinematic viscosity (centistokes)	Kinematic viscosity (m ² /s)
B25	40	33	1.35	1.35 X 10 ⁻⁶
B50	40	34	1.74	1.74 X 10 ⁻⁶
B75	40	35	2.12	2.12 X 10 ⁻⁶
B100	40	36	2.50	2.50 X 10 ⁻⁶

Table 3. Tabulation of kinematic viscosity for biodiesel and its blends

Biodiesel blend	Flash point (°C)	Fire point (°C)	Density (Kg/m ³)	Calorific value (MJ/Kg)
B25	38	58	838	42.19
B50	42	62	848	40.34
B75	54	80	858	38.49
B100	74	110	866	37.01

Table 4. Fuel properties of biodiesel blends

Biodiesel blend	Flash point (°C)	Fire point (°C)	Kinematic viscosity (centistokes)
B25	38	58	1.35
B50	42	62	1.74
B75	54	80	2.12
B100	74	110	2.50

Power Developed and Fuel Consumption

The experimentations were performed with a single-cylinder 4-stroke diesel engine that coupled with a Rope Brake dynamometer. The engine was fueled by the mixture of petroleum diesel, B100, and various biodiesel fractions. The aim was to evaluate the operational traits of the engine using these fuels, with a focus on comparing parameters such as Brake Thermal Efficiency, Brake Power (B.P), Mechanical Efficiency, Specific Fuel Consumption (SFC), and Indicated Thermal Efficiency.

As Figure 3 shows, the BTE of B 100 blend is higher than that of petro-diesel and of some of the blends up to 2 KW but lower for higher KW. Petro-diesel and the B25 blend exhibit remarkably similar characteristics, with their curves aligning almost completely throughout the entire range of observation. The maximum load applied to the engine is 12kg, which correlates to a brake power of 3.05 KW. Specifically, B100 is more effective than B75 under the maximum load. The chart presented in Fig shows that the B100 blend has a similar brake thermal efficiency (BTE) to that of diesel. These results are in harmony with the previous work of Gupta et al. (2016), Hussain et al. (2020), Seraç et al. (2020), , and Gavhane et al. (2021) [46-49].

The analysis of Figure 4 indicates that the thermal efficiency for the B100 blend surpasses that of diesel and other biodiesel blends. Note that the indicated thermal efficiency shown is influenced by the engine B.P and F.P, as it represents the sum of these two components. The diesel performance curve tops with over the biodiesel blends with the only exception of B100 when the engine load is intensified. The diesel trend follows a smooth linear evolution. B100 displays a distinctive curve that rises initially up to 1.52 KW, followed by a decrease, and then a gradual increase towards the conclusion. B25 demonstrates a consistently uniform

linear trajectory throughout. In the blends, B75 outperforms B50 and B25 at loads of up to half (2.50 H.P or 1.82KW). Consequently, B100 exhibits competent and sufficient performance characteristics, particularly when contrasted with petroleum-diesel and various biodiesel mixtures.

From Figure 5, the linear pattern of mechanical efficiency (Brake HP/Indicated HP) versus Brake Power is observed for both petro-diesel and biodiesel, wherein both lines are approximately line to each other at similar brake power levels. B50 demonstrates better performance compared to all other biodiesel blends for higher brake power, although the performance differences amongst each other have been observed to be less pronounced in this range. Hence, the contrast between B100 and petroleum diesel indicates that biodiesel consistently yields analogous outcomes in comparison to diesel.

Turning to Figure 6, which illustrates specific fuel consumption against Brake Power, it is evident that specific fuel consumption, representing the mass flow rate of fuel consumed per unit KW of brake power produced, declines with an increase in the applied load on the engine, corresponding to an elevation in brake power. According to the graph the per energy requirement of biodiesel is greater than that of diesel due to its lower calorific value by 20% as compared to diesel. As such, combustion of biodiesel results in less heat being released as compared to diesel, which requires 1.20–1.30 litres of biodiesel to deliver the same power as 1 litre of diesel. While biodiesel may have a higher fuel consumption than diesel, the heat produced by both fuels is nearly equivalent, a point further illustrated in the next section. These observations are also supported by previous work by Gupta et al. (2016), Hussain et al. (2020), Seraç et al. (2020), and Gavhane et al. (2021) [46-49].

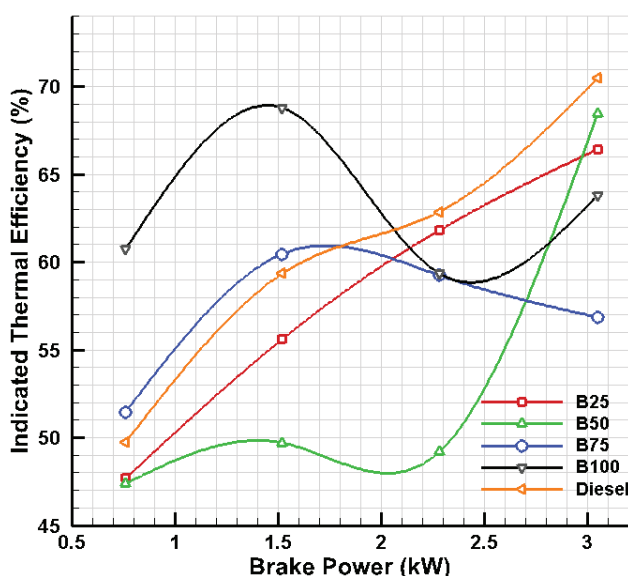


Figure 3. Estimation of Brake Thermal Efficiency with variation in Brake Power.

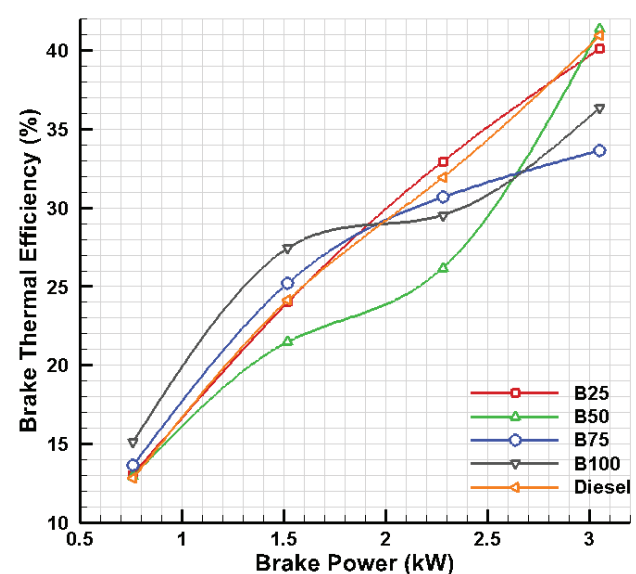


Figure 4. Estimation of Indicated Thermal Efficiency with variation in Brake Power.

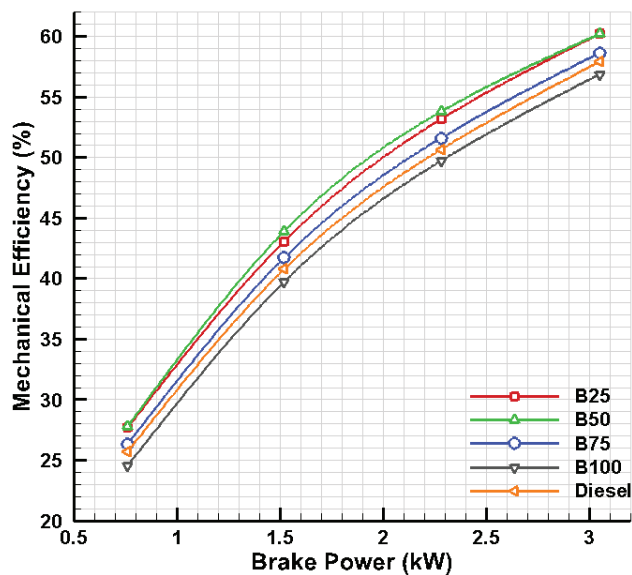


Figure 5. Estimation of Mechanical Efficiency with variation in Brake Power.

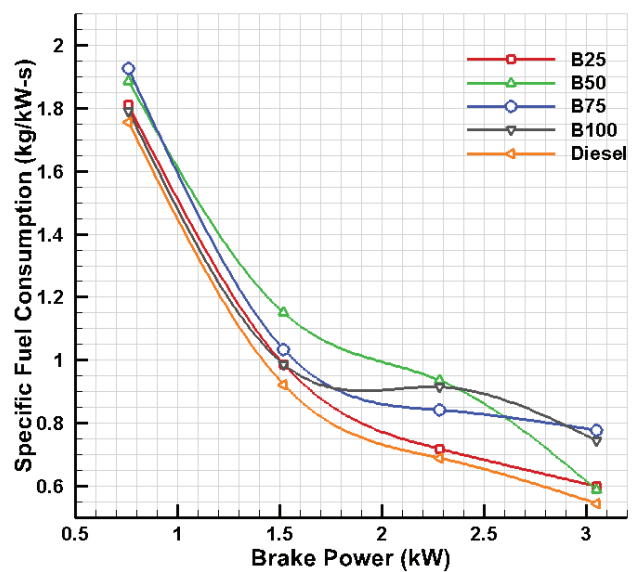


Figure 6. Estimation of specific fuel consumption with variation in Brake Power.

Heat Balance Analysis

The evaluation of engine performance is typically expressed through a heat balance sheet. Various quantities, including the total heat generated during fuel combustion to produce brake power and the heat dissipated through the use of coolant and into the surroundings, have been calculated. These results are then compared to illustrate their overall efficiency. The core principle underlying this sheet is that not all the heat generated during fuel combustion is transformed into useful work at the engine's crankshaft; a portion is lost as previously described. The findings are visually represented through the pie charts, and the outcomes are explored herein.

Figure 7 clearly demonstrates that B100 utilizes the heat generated from fuel combustion within the engine more efficiently, leading to a greater production of brake power in comparison to diesel. B100 is recognized as the most effective biodiesel blend for delivering output power. The heat losses that are not accounted for, resulting from conductive, convective, and radiation processes, are significantly greater in the engine fueled by B100 in comparison to diesel. The heat from the exhaust gas can be effectively

harnessed using techniques for recirculating exhaust gases, thereby improving both the volumetric and thermal efficiency of the engine. B25 demonstrates superior performance compared to B50 and B75 in the biodiesel blends category, providing a more efficient overall heat utilization. The heat balance performed by B100 has a higher level of efficiency than diesel. Table 5 summarizes heat utilization and heat losses for the different blends of biodiesel.

Analysis of diesel engine exhaust gas with diesel and different blends of biodiesel

Emission levels from the exhaust were assessed using the AVL Di Gas 444 exhaust gas analyzer, connected to the exhaust pipe of the diesel engine. Figure 8 illustrates a clear trend in which CO emissions from B100 gradually decrease with increasing applied load. B100 exhibits a 40% reduction in CO emissions compared to diesel, with 5–15% lower emissions in comparison to biodiesel blends. CO emissions on first rising are also higher as rich mixture is initially employed. B100 exhibits good burning behaviour at higher loads with lower CO in the exhaust.

The HC emissions yield a 10–15% increase for B100 compared to diesel; this may be due to the fuller composition of the B100 blend and longer hydrocarbon molecules formed. B25, among biodiesel blends, demonstrates reduced emission levels.

B100 shows a 16% increase in CO₂ emissions relative to diesel. B25 exhibits reduced CO₂ levels, while B75 demonstrates elevated CO₂ levels. Oxygen gas emissions from B100 are the lowest among all blends, demonstrating the effective use of oxygen for optimal combustion in biodiesel. The plots demonstrate that biodiesel blends B75, B25, and B50 show enhanced oxygen availability in comparison to diesel.

Table 5. Heat balance observation

Biodiesel blend	Heat Utilization (%)	Heat losses (%)
B25	24.10	75.90
B50	22.76	77.24
B75	23.08	76.92
B100	24.16	75.84

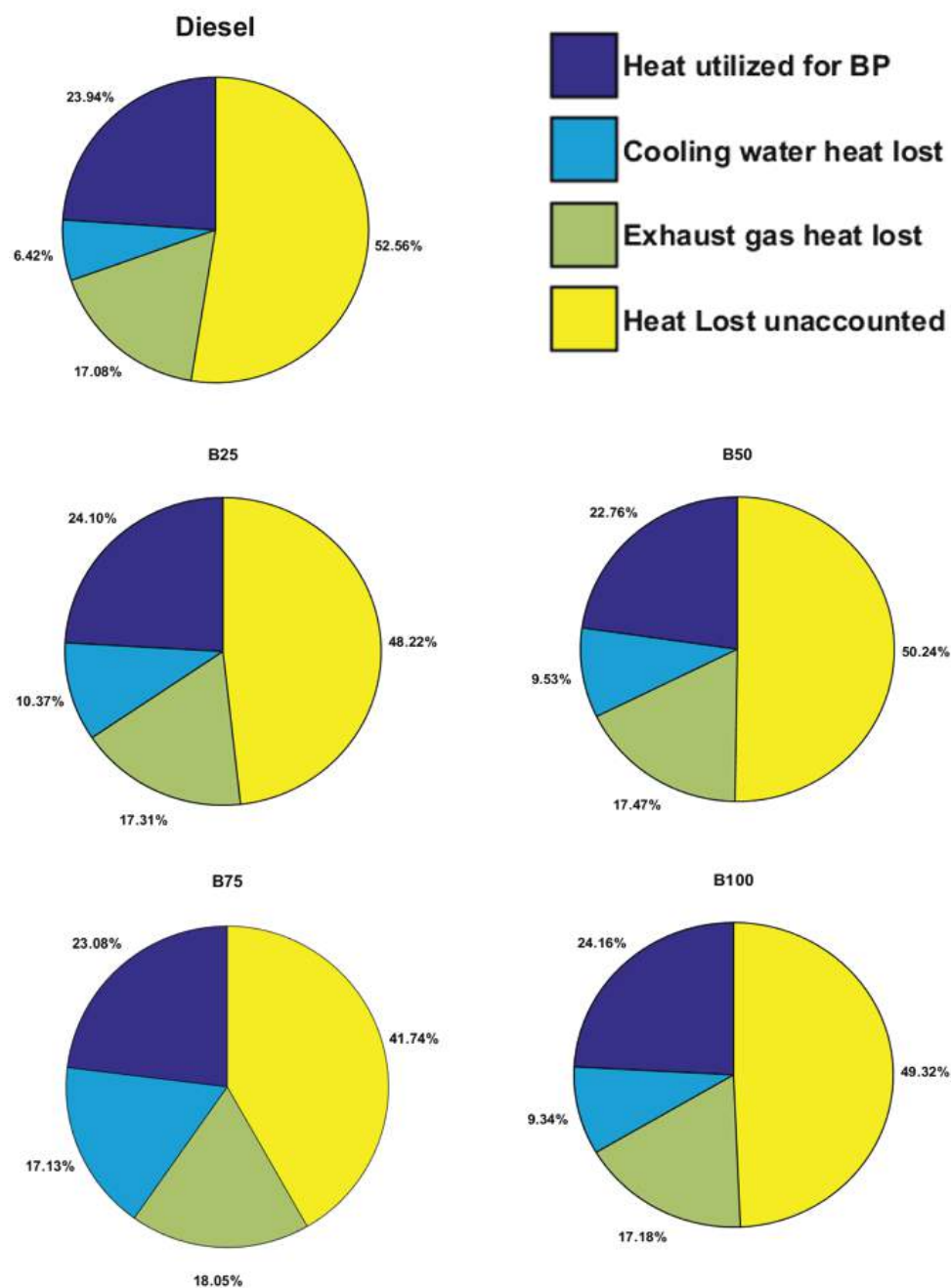


Figure 7. Utilization of Heat in an Engine Fueled with Biodiesel, Diesel, and their Blends.

Emissions of NO_x from B100 exceed those from diesel as a result of elevated combustion temperatures, which enhance the production of NO_x . Additives such as ethanol can be utilized to effectively manage charge temperature. Biodiesel blends have NO_x emissions which move in the upward direction with an increase in the load until the maximum level is reached when B100 is used due to its the presence of more oxygen. The B25 and B50 blends, on the other hand, showed lower NO_x emissions, which means replacing diesel with biodiesel to some extent can decrease

NO_x emissions. However, at low loads, NO_x emissions do not vary with fuels used but at high loads, there is a marked increase in emissions for B100. The mitigating approaches for NO_x emissions includes but not limited to the exhaust gas recirculation, NO_x -reducing additives, blended optimization, advanced combustion techniques, after-treatment systems, and optimization of injection timing.

Hence to reduce the NO_x emissions, the following approaches can be used, for instance:

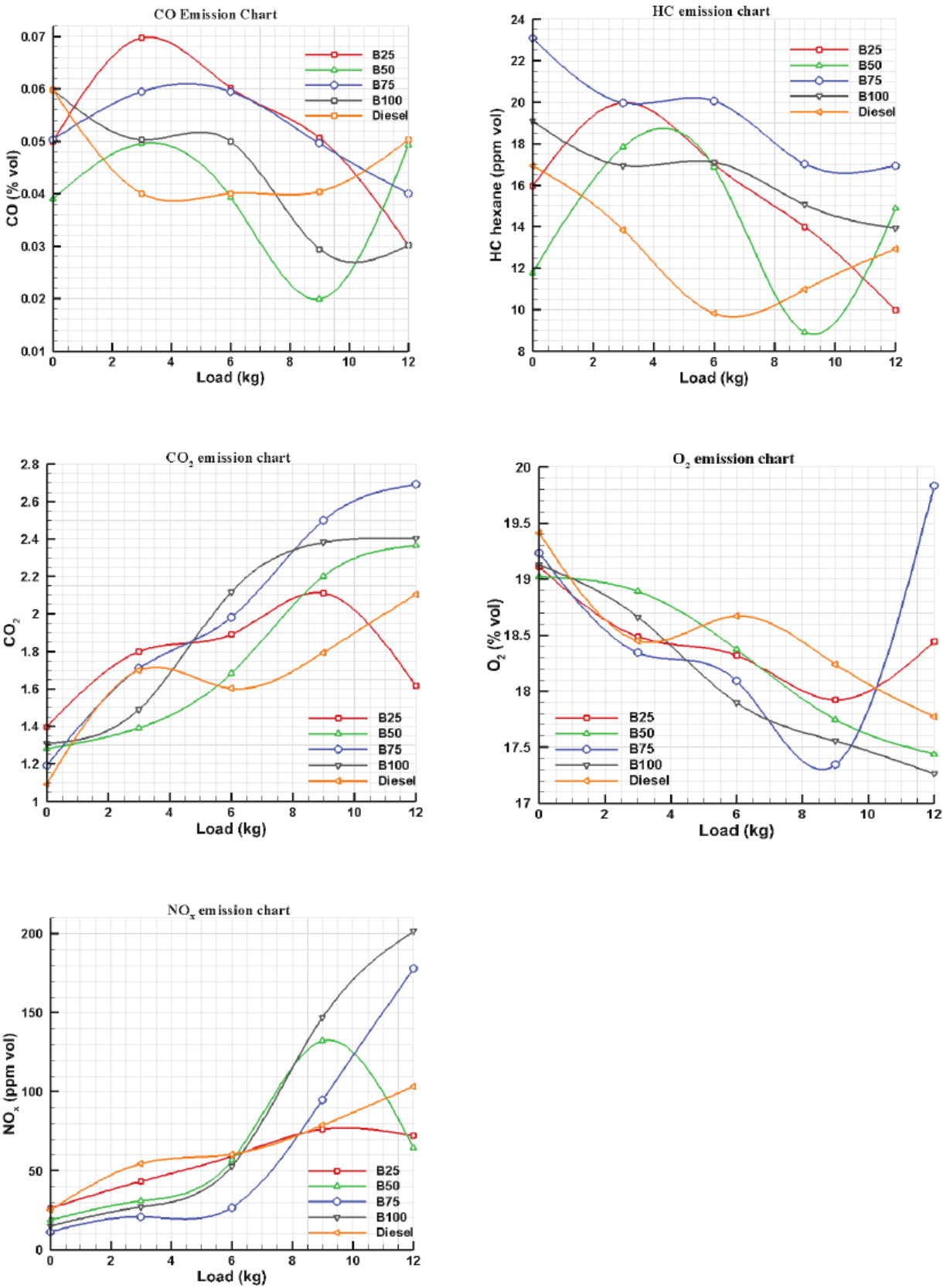


Figure 8. Analysis of Exhaust Gas Emissions from Biodiesel (B100), its Blends and Conventional Diesel

Exhaust Gas Recirculation (EGR): It involves the re-introduction of some of the burnt gases into the combustion chamber which reduces the oxygen levels available in the chamber.

Selective Catalytic Reduction (SCR): It is the process that employs the use of a catalyst and a reducing agent for certain nitrogen oxides for example moccasin which is used to transform NO_x into benign nitrogen and water.

Fuel Additives: The addition of cetane improvers and other combustion modifiers targeting lowered peak temperatures for suppression of NO_x generation.

By this, the negative effects of centered biodiesel coupled with soybeans as raw material can be reduced thus providing enhanced benefits environmental impacts of biofuels.

The adoption of biodiesel results in lower CO emissions which is an environmental advantage but then its effect on NO_x emissions is a problem. More research and optimization on the blends and combustion technologies will sort these trade-offs out. The use of appropriate catalytic converters can also efficiently manage these emissions. As a result, the emission patterns have been plotted and compared and they are resulting from fuels having different compositions. Other research carried out by Hussain et al. (2020), Seraç et al. (2020), Gupta et al. (2016), and Gavhane et al. (2021) also corroborate these tendencies [46-49].

Performance vs. Emissions Trade-offs

The performance/emission balance is readily observable from the results with the soybean biodiesel blends. The benefits and limitations brought about by biodiesel blends over conventional dieels can thus be documented:

Performance Benefits: Using B100 biodiesel blends at all load conditions results in higher Brake thermal efficiency (BTE). This is because biodiesels have higher oxygen content hence better combustion. Unfortunately, this efficiency leads to an increase in SFC at higher concentrations of biodiesel. This is brought about by the fact that biodiesels have a lower calorific value compared to diesel.

Emission Trade-offs: However, biodiesel blends lead to an increase in Nitrogen Oxide (NO_x) emissions despite demonstrating lower Carbon Monoxide (CO) emission (the reduction can be as high as 40 when using B100), which is about the same HC emission volume. This trend is consistent with the previous studies, for instance, the investigation done by Canakci et al. [10], where it was established that the best performance attained with biodiesel usage was coupled with an increase of NO_x emissions for B100 this is the result of better combustion capabilities of biofuels in general because of increased combustion temperatures.

Balancing Performance and Emissions: Employing Selective Catalytic Reduction (SCR) and Exhaust Gas Recirculation (EGR) technologies can maintain the NO_x emission level at a low concentration while leveraging the vigorous combustion efficiency offered by biodiesel, a mixture with a less concentration of biofuels. Similarly, SFC

and NO_x emissions would be less than what they would be if the ratio of the biofuel was lower while still giving good performance gains, approaches could include B50 or B75 optimum ratio blending.

Potential Limitations and Environmental Considerations

Impact of Environmental Conditions: There are some factors fuel characteristics which can shift the performance and emissions characteristics of soybean biodiesel blends they include temperature, humidity and altitude.

Cold Weather Performance: Biodiesel's fuel characteristics results in high cloud and pour points which means biodiesel tends to gel in colder climates. This is preventable by using winterization enablers or blending.

Effects of High Altitude: A lack of oxygen reduces the efficiency of combustion and results in higher emissions.

Opportunity for Variation in Engine Specificity: The results from the single-cylinder engine that we used in this study may not be fully representative of multi-cylinder or heavy-duty engines.

Fuel Stability: The fuel could deteriorate and become oxidised if kept too long, and hence strong storage is required.

COMPARISON WITH PREVIOUS STUDIES

The present study is contrasted with that of Gupta et al. (2016) and the work of Kumar et al. (2020), as shown in Figure 9.

One set of experimental work is compared for the validation purpose, which is a variation of brake thermal efficiency with brake power. Gupta et al. (2016) work focused on the soybean biodiesel with maximum blends of 20%, i.e.,

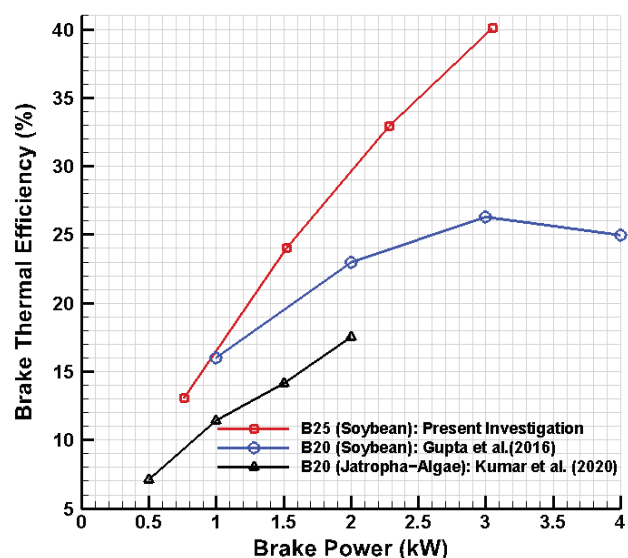


Figure 9. Validation of the experimental work [46,50]. [It was drawn by the author].

B20. The results show similar trends, but the trends also suggest an increase in breakdown thermal efficiency with increasing the blends. B20 blends will obviously not be the feasible alternative of the diesel; hence, the comparison supports going with a higher amount of blends in order to achieve a complete alternative of the diesel with soybean biodiesel. The result is also compared with the other biodiesel, Kumar et al. (2020), and the performance of soybean biodiesel is found to be better than previously used jatropha and algae biodiesel.

CONCLUSION

Using data obtained from performance tests on a single-cylinder 4-stroke 5 H.P diesel engine, employing petroleum-diesel, Biodiesel (B100), and their blends, satisfactory results have been observed. Many reports have shown that B100 and its blends with other fuels pass ASTM standards for fuels and fuel blends. In addition, the operational engines parameters of biodiesel are very similar to those of diesel fuel, providing a more environmentally friendly alternative. However, the specific fuel consumptions for biodiesel and its blends are higher than that of diesel due to the lower heating value of the biodiesel and its blends.

For the operational parameters of biodiesel and diesel fuels concerning brake power, brake thermal efficiency, fuel consumption, and mechanical efficiency, the fuel consumption and other parameters seems to have the same pattern. There is a cross in the B100 biodiesel and other fuels, and B100 biodiesel is the only fuel that has the highest brake thermal and indicated free thermal efficiency which indicates the best performing fuel.

The heat balance graphs for blends with biodiesel indicate better performance than petroleum-diesel in the more effective use of the heat provided by combustion of the fuel to produce brake power. This is very important in determining the thermal efficiency of a cycle of an engine. In addition, for the engine cycle exhaust emissions with diesel, biodiesel, and other blends, it is observed that biodiesel has more HC and CO₂ emissions. This is owing to the longer length of the hydrocarbon chains in biodiesel, which results in a higher concentration of combustibles

NO_x emissions result from higher combustion temperatures, but effective control can be achieved through the addition of appropriate additives (such as ethanol) and the use of catalytic converters.

The paper demonstrates how biodiesel can be obtained from soybeans while running multipurpose engines that use biodiesel on model based conditions. The findings also indicate that there can be multi usage of soybean biodiesel as sustainable fuel in agriculture, transportation, and power generation while putting low pollution strategies in place.

In conclusion data shows that biodiesel has better performance and makes efficient use of heat than diesel petrol. From the observed date, biodiesel is a possible alternative fuel for present-day diesel engines without any

modification. It is characterized as a clean, green, economically feasible, eco-friendly, renewable, and biodegradable option in contrast to petroleum-diesel.

Future Scope

In the case of soybean biodiesel, the process of optimization of the blends will increase fuel performance and cold weather characteristics and minimize NO_x emissions, and there is scope of introducing more feeds to achieve sustainability. The study can be aimed at creation of blends with more advanced additives and formulations to reduce emissions and enhance combustion efficiency. Co-processing of soybean oil along with non-food feed providers like algae or used cooking oil can resolve the “fuel versus food” dilemma and help in fuelling diversification. The improvement can come from the advancement in purification, catalysis, and genetic improvement for higher-yield soybean varieties. Moreover, supportive policies and international cooperation can facilitate and make financially feasible, the large-scale application, hence, increasing the competitiveness of soybean biodiesel in the transition to energy.

NOMENCLATURE

<i>ARR</i>	After-tax rate of return
<i>ASTM</i>	American Society of Testing Materials
<i>B100</i>	Biodiesel with 5% ethanol
<i>B75</i>	75% biodiesel + 20% petro-diesel + 5% ethanol
<i>B50</i>	50% biodiesel + 45% petro-diesel + 5% ethanol
<i>B25</i>	25% biodiesel + 70% petro-diesel + 5% ethanol
<i>BBP</i>	Biodiesel break-even price
<i>BP</i>	Brake Power
<i>CO₂</i>	Carbon dioxide
<i>CO</i>	Carbon monoxide
<i>FP</i>	Frictional Power
<i>HC</i>	Hydrocarbon
<i>IP</i>	Indicated Power
<i>NNP</i>	Net annual profit after taxes
<i>NO_x</i>	Nitrogen oxides
<i>O₂</i>	Oxygen
<i>SFC</i>	Specific Fuel Consumption

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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.

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Research Article

Investigation on performance and emission studies of variable compression ratio engine using neem oil biodiesel

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ABSTRACT

Biodiesel, particularly Neem oil-based biodiesel, has emerged as a promising eco-friendly alternative, offering reduced greenhouse gas emissions, lower sulphur content, cleaner combustion, and biodegradability. Variable Compression Ratio (VCR) engines further enhance performance, fuel efficiency, and emissions reduction across various fuels. This study focuses on experimenting with Neem-based biodiesel in VCR engines and conducting a parametric study to address the challenges of fossil fuel use. The experimental study revealed several key findings. CO emissions decreased with higher biodiesel blending and compression ratios, achieving a 28% reduction with NB50 at 6 N-m load and CR 22. CO₂ emissions are reduced by 20% using B40 at CR 22 under full load (12 N-m). A 27% reduction in HC emissions is noted with B30 at CR 20 under full load. NO_x emissions increased by 20% at full load and CR 22:1 with pure diesel. Brake Thermal Efficiency increased by 16% with NB30 at 6 N-m load and CR 22. SFC significantly decreased under no-load conditions with NB50. Optimal results from AI-TLBO with NB40 fuel and a compression ratio of 22 at no-load showed reductions of 20.20% in CO, 23.84% in CO₂, 24.5% in HC, and a 23.1% NO_x increase.

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INTRODUCTION

The growing reliance on fossil fuels, particularly in the transportation sector, poses serious challenges due to their finite availability and detrimental environmental effects. Transportation, which consumes the largest share of petroleum, contributes significantly to pollution, adversely impacting the climate, human health, ecosystems, and agricultural productivity [1]. Fossil fuel combustion,

especially in diesel engines used for maritime, rail, and road transport, produces harmful emissions such as carbon monoxide (CO), hydrocarbons (HC), sulfur oxides (SO_x), nitrogen oxides (NO_x), and particulate matter (PM) [2, 3]. These pollutants not only exacerbate global warming but also degrade air quality, leading to severe health problems. Diesel engines, a critical component of internal combustion (IC) technology, are widely favored for their higher efficiency and durability compared to gasoline engines.

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However, the trade-off lies in their production of more pollutants. In addition to greenhouse gases like carbon dioxide (CO₂), diesel engines emit greater amounts of harmful pollutants [4]. This makes it imperative to explore alternative fuels that are cleaner and sustainable. India, like many countries, began addressing pollution with regulations in 1989, focusing on idle emission controls, followed by the introduction of mass emission standards in 1992 for diesel vehicles. However, despite regulatory advances, diesel engines continue to emit pollutants that harm both the environment and human health [5]. The need for cleaner fuel alternatives is pressing, particularly given the transportation sector's dominant role in petroleum consumption.

Research has shown that biodiesel, a renewable alternative to conventional diesel, offers significant advantages in reducing emissions. While biodiesel usage tends to increase NO_x emissions, it can be controlled through exhaust gas recirculation (EGR) systems [6]. notably, biodiesel significantly reduces emissions of CO, PM, and HC, pollutants that are major contributors to air pollution. The U.S. Environmental Protection Agency (EPA) has acknowledged biodiesel's overall benefits, citing improvements in exhaust emissions except for NO_x [7]. As fossil fuels become increasingly scarce and expensive; biofuels emerge as a practical and environmentally friendly alternative [8]. Fossil fuel combustion not only depletes natural resources but also accelerates environmental degradation, making the transition to alternative fuels essential. Though gasoline-based fuels remain viable in some regions, their availability is declining. Biofuels, produced from locally sourced materials, offer a promising solution to the ongoing energy crisis by providing a renewable and sustainable energy source [9].

One particularly promising biofuel is neem oil, which has been highlighted as an alternative biodiesel due to its lower viscosity and abundant availability [10]. Neem oil biodiesel, especially when blended with other biodiesel types, can significantly reduce emissions without compromising engine performance. With increasing energy demands and the dwindling supply of fossil fuels, research into renewable fuel sources like neem oil is accelerating. The key benefits of neem oil biodiesel include renewability, biodegradability, and improved exhaust emissions, making it a cost-effective and sustainable replacement for conventional diesel in compression ignition (CI) engines [11]. In addition to neem oil, researchers are exploring various organic additives that can enhance engine efficiency and reduce emissions. These additives improve combustion rates, lower exhaust emissions, increase mileage, and enable fuels to function under extreme temperatures. By integrating these additives into diesel fuels, engine performance and environmental impact can be optimized. Organic additives, with their higher oxygen content, contribute to better combustion, which in turn reduces harmful emissions. Several studies have compared the performance and emissions of different biodiesel blends with conventional diesel. For example, a study

investigating biodiesel blends produced from rapeseed oil demonstrated reductions in CO and smoke emissions by up to 60%. Similarly, blends containing 20% and 40% biodiesel showed a reduction in NO_x emissions by 39% and 28%, respectively, compared to standard diesel [12]. These findings underscore biodiesel's potential as a superior substitute for conventional diesel, particularly in reducing harmful emissions.

Further research into biodiesel production and engine performance has led to promising results. For instance, studies using Jetropha methyl ester (JTME) and diesel blends showed that JTME and its blends emit less heat and have lower emissions compared to diesel, with the exception of NO_x. Other research involving marine diesel engines demonstrated that biodiesel blends with up to 50% biodiesel reduced emissions of PM, NO_x, and CO. Experiments with ethanol as an additive in diesel fuel have also shown potential. Ethanol boosts fuel combustion, reducing CO₂ emissions by approximately 25% [13]. This model encourages increased biofuel use and offers a practical way to enhance the efficiency of gasoline engines. Additionally, experiments with microalgae biodiesel, produced via the transesterification technique, revealed improved performance and lower exhaust emissions when compared to conventional diesel. Researchers have also examined the effects of various engine parameters, such as fuel injection pressure and nozzle design, on biodiesel combustion. For example, studies showed that using higher fuel injection pressures improved engine performance and reduced emissions. Moreover, adjusting nozzle holes and compression ratios in diesel engines running on biodiesel blends has demonstrated significant improvements in efficiency and emission reductions [14].

In conclusion, the shift from fossil fuels to biodiesel and other renewable fuels is becoming increasingly critical due to the environmental and economic pressures associated with fossil fuel use. Biodiesel, particularly neem oil-based and other biofuel blends, presents a sustainable and eco-friendly alternative to conventional diesel, offering significant reductions in harmful emissions without compromising engine performance. Continued research and innovation in biodiesel production, engine optimization, and fuel additives will be essential in ensuring a cleaner and more sustainable energy future. Till date, no research has been found on high blending ratios with neem oil, nor on adjusting engine parameters, such as compression ratio, to accommodate these higher blends within the scope of literature review referred for the current study. Therefore, this study examines a higher compression ratio range of 14:1 to 22:1, an area still in need of exploration. Bio Diesel will be compulsory used in Marine Engine as a primary fuel. Current study will help to use biodiesel in Marine Engines. This study offers environmental and health benefits. By reducing greenhouse gas emissions, lowering sulfur content, and enabling cleaner combustion, biodiesel as a blending agent minimizes air pollution, which directly impacts

respiratory health. Additionally, biodiesel's biodegradable nature lessens environmental contamination, making it a sustainable alternative to fossil fuels and contributing to a cleaner, healthier environment.

Emissions

Gasoline and diesel engine exhaust emissions are major contributors to atmospheric pollution, which has led to the development of alternative fuels [15]. Automobiles significantly contribute to this pollution by releasing harmful exhaust pollutants into the air, many of which are irritants, odorous, and even carcinogenic. Carbon monoxide (CO) emissions result from incomplete combustion, uneven air-fuel mixtures, and elevated chamber temperatures. Heavy-duty vehicles, particularly during start-up or acceleration under load, produce CO due to the higher fuel mixture used. The release of CO is not only wasteful in terms of fuel but also represents a loss of chemical energy.

Nitrogen oxides (NO_x) are formed in the ignition chamber when oxygen and high temperatures are present. The reaction of NO_x with natural gases creates ozone, a key component of photochemical smog. Prolonged exposure to NO_x can lead to health issues such as cyanosis—especially on the lips, body, and toes—and can cause structural damage to lung cells [16]. Biodiesel has emerged as a popular alternative to fossil fuels, addressing both energy needs and concerns over global warming. As an environmentally friendly fuel, biodiesel helps reduce the carbon dioxide (CO₂) problem, which is a leading cause of global warming. This is because the CO₂ emitted during biodiesel combustion is equivalent to the amount absorbed by plants during their growth, thus avoiding the greenhouse effect [17]. Studies have shown the benefits of using biodiesel. For instance, Chen He et al. conducted experiments using cotton seed oil and diesel, finding that O₂ acts as an oxygen-donating

catalyst, improving combustion and engine thermal efficiency. This catalyst also helps reduce emissions of CO and hydrocarbons. Neem oil and cotton seed oil, which contain 30% to 40% oil, are promising substitutes for other bio fuels [18]. Research by Md. Nurun Nabi et al. demonstrated that 76% of biodiesel can be recovered from cotton seed oil, with the remaining 23% consisting of methanol and other compounds. Similarly, neem oil biodiesel has been shown to enhance engine efficiency and reduce NO_x emissions. In an experiment with cotton seed biodiesel, Sandeep Singh et al. suggested that 10%, 20%, and 30% biodiesel blends could serve as effective replacements for mineral oil diesel, offering properties similar to conventional diesel fuel [19].

MATERIALS AND METHODS

In this research, neem oil-based biodiesel was blended with diesel and evaluated for performance and emissions. The blends were assessed using an AVL Engine test rig, with emissions data collected via a smoke analyzer attached to the system. To reduce viscosity, neem oil sourced from a local market was transesterified. This process involved using a hot plate and magnetic stirrer, with the raw neem oil preheated to 100°C to eliminate moisture. Sulfuric acid (H₂SO₄) was used for acid pre-treatment to lower the free fatty acid (FFA) content.

The performance and emission characteristics of neem oil blends (10%, 20%, 30%, 40%, and 50%) were compared to those of diesel. These blends, designated as NB10, NB20, NB30, NB40, and NB50, were prepared as shown in Figures 1. The engine's performance was tested under various loads with these biodiesel blends, and the results are discussed in subsequent sections. The properties of neem oil are presented in Table 1.



Figure 1. Biodiesel preparation setup.

Table 1. Properties of neem oil

Sample/ properties	Density at 30 °C	Calorific value	Calorific value	Flash point	Fire point	Kinematic viscosity @40 °C	Dynamic viscosity @40 °C
Unit	Kg/m ³	Cal/gm	Cal/gm	°C	°C	cSt	cP
ASTM standard	D287	D4809	D4809	D9358T	D9258T	D445	D445
Std Diesel	816	10235	10841	53	56	2.09	1.73
NB10	824	10172	10758	62	66	2.35	1.93
NB20	833	10013	10599	69	74	2.66	2.21
NB30	839	9828	10414	76	82	2.86	2.39
NB40	846	9632	10218	81	85	2.93	2.48
NB50	861	8871	9457	121	136	4.4	3.8

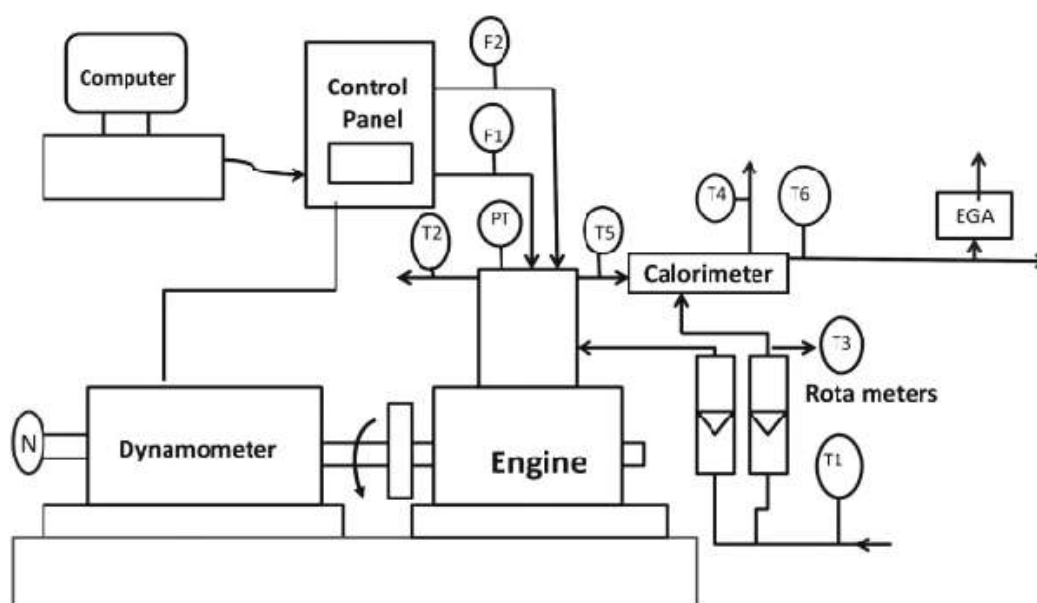
EXPERIMENTAL WORK

The experiments were carried out under varying loads of 0, 3, 6, 9, and 12 kW at a constant rated speed, with an injection pressure of 210 bars and an exit cooling water temperature of 65°C. Five neem oil blends—NB10, NB20, NB30, NB40, and NB50—were tested. During each measurement, the engine was allowed to stabilize sufficiently. All readings were taken three times to ensure accurate and consistent results. The study evaluated performance parameters such as Mechanical Efficiency and Brake Thermal Efficiency, as well as emission parameters including Carbon Dioxide (CO₂), Carbon Monoxide (CO), Smoke, and Nitrogen Oxides (NO_x). These results were compared to those of pure diesel. Prior to data collection, the engine was run for about three minutes to reach stability and maintain

a constant speed of 1500 rpm. After the warm-up period, engine performance and emission measurements were recorded under different compression ratios of 14, 16, 18, 20, and 22.

Figure 2 shows the line diagram of the experimental setup used in the present study. Following are the notations used in the Figure 2.

- T1 - Inlet (engine) water temperature (°C),
- T2 -Outlet (engine) water temperature (°C),
- T3 -Inlet (calorimeter) water temperature (°C),
- T4 -Outlet (calorimeter) water temperature (°C),
- T5 -Exhaust gas temperature before calorimeter (°C),
- T6 -Exhaust gas temperature after calorimeter (°C),
- F1 - Fuel consumption measurement unit,
- F2 -Air flow measurement,
- EGA -Exhaust gas analyzer and

**Figure 2.** Line diagram of variable compression engine experimental setup.

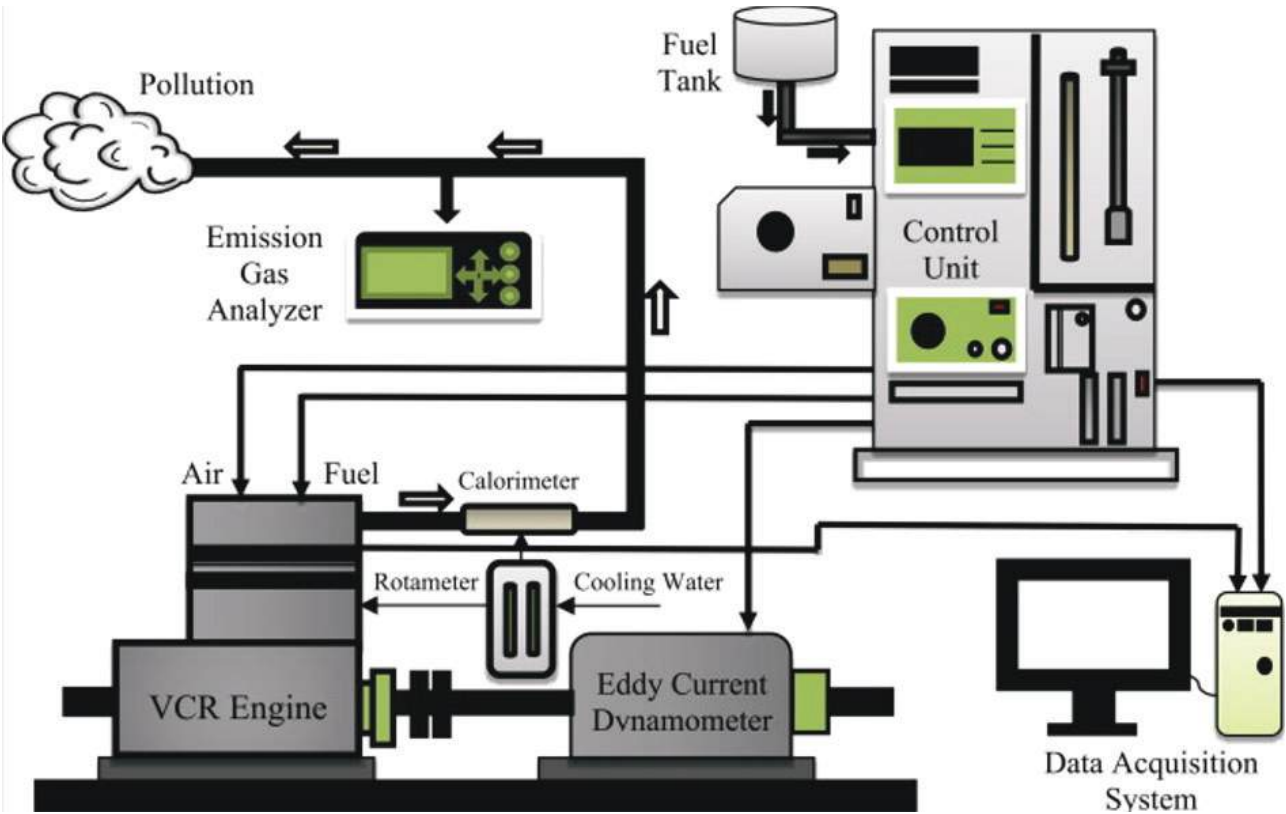


Figure 3. Schematic diagram of variable compression engine experimental setup.



Figure 4. Variable compression engine experimental setup (Engine, Flue inlet system, flow measurement, control system, dynamometer, and Exhaust gas analyser system).

Table 2. VCR engine details

Power 3.50 kW @ 1500 rpm	Single cylinder	Four strokes	Constant speed
Water cooled	Diesel engine	Cylinder bore 87.50 (mm)	Stroke length 110.00 (mm)
Connecting rod length 234.00 (mm)	Compression type VCR	Compression ratio 14:1 to 22:1	Variable compression ratio engine - swept volume 661.45 (cc)

N -Engine speed measurement.

Figure 3 represents the schematic diagram of variable compression engine experimental setup and Figure 4 shows the variable compression engine experimental setup (Engine, Flue inlet system, flow measurement, control system, dynamometer, and Exhaust gas analyser system).

VCR Engine Parameters and Features

Detail specifications of VCR engine shown in Table 2.

The VCR engine setup offers several advanced features, including the ability to change the compression ratio without halting the engine and maintaining the combustion chamber geometry. It supports dual-fuel testing and provides real-time performance analysis with PV and performance plots. The system allows online measurements, data logging, editing, printing, and export of configurable graphs, along with detailed combustion analysis. The setup enables comprehensive evaluation of engine performance parameters such as BP, IP, FP, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, mechanical efficiency, volumetric efficiency, specific fuel consumption, and air-fuel ratio. Additionally, it facilitates emission testing for CO, CO₂, HC, and NOx levels.

**Figure 5.** Exhaust gas analyzer (HG 540).

Emission Measurement

All emissions, including CO, CO₂, unburned hydrocarbons (UHC), NOx, and unused oxygen, were measured using the AIRREX HG-540 Gas Emission Analyzer, as shown in Figure 5. One end of the analyzer's cable was connected to the analyzer's inlet, while the other was attached to the engine's exhaust outlet. To ensure proper functionality, the analyzer requires continuous charging. Actual images depict the Exhaust Gas Analyzer fitted at the exhaust exit of the engine. The analyzer features an automatic «zero-setting» mechanism, which resets the sensor using fresh air. When the analyzer is powered on, it requests a zero setting at 10, 20, and 30-minute intervals, and continues to request it every 30 minutes thereafter. The zero port enables the gas analyzer to reset to zero even while the sampling probe is still connected to the exhaust tail. Ambient air is drawn through a charcoal filter to help reduce hydrocarbon (HC) levels to zero. To zero the CO, CO₂, and NOx readings, the analyzer uses a keyboard function that is calibrated to 20.34% oxygen by volume. The specifications of the gas analyzer used are presented in Table 3.

RESULTS AND DISCUSSION

Emission of Carbon Monoxide (CO)

At lower compression ratios, the engine may not reach the ideal combustion temperatures and pressures, leading to incomplete combustion of the fuel-air mixture. This results in higher CO emissions, as CO is produced when carbon in the fuel is not fully oxidized. Increasing the compression ratio generally enhances engine efficiency by promoting more complete fuel combustion, thereby reducing CO emissions. There is typically an optimal range of compression ratios where CO emissions are minimized, as the engine maintains a balance between

Table 3. Gas analyzer specification

Model	HG – 540- Automotive gas analyzer
Emission measuring	CO, CO ₂ , HC, O ₂ and NOx,
Method of measuring	CO, CO ₂ , HC, — NDIR (Non-dispersive method infrared) O ₂ , NOx (electro chemical)
Repeatability	Less than + 2% FS
Response time	Within 15 seconds (more than 90%)

sufficient combustion temperature and pressure while avoiding knocking. The effect of compression ratio on CO emissions also depends on the engine load. At higher loads, the need for a richer fuel mixture can increase CO emissions, potentially negating the advantages of a higher compression ratio if not properly managed. In conclusion, varying compression ratios, combined with neem oil-based biodiesel blending, can significantly contribute to reducing CO emissions. The experimental study concluded that CO emissions decrease with an increase in both the biodiesel blending ratio and compression ratio. Figure 6 to 10 shows the effect of variation in compression ratio on CO emission at zero, 3, 6, 9 and 12 N-m loads for different blends.

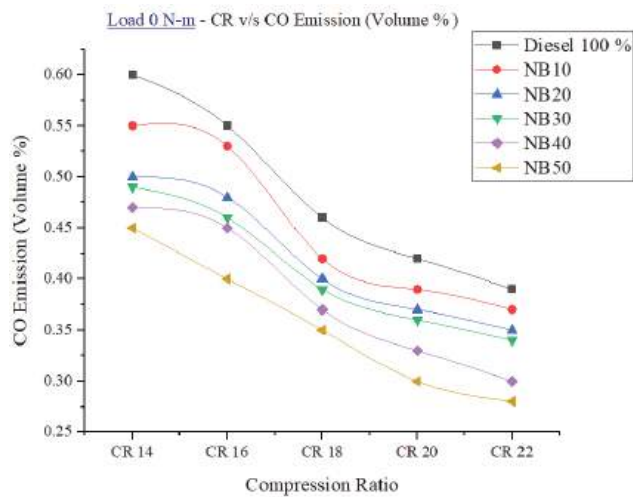


Figure 6. Effect of compression ratio on CO emission at zero load.

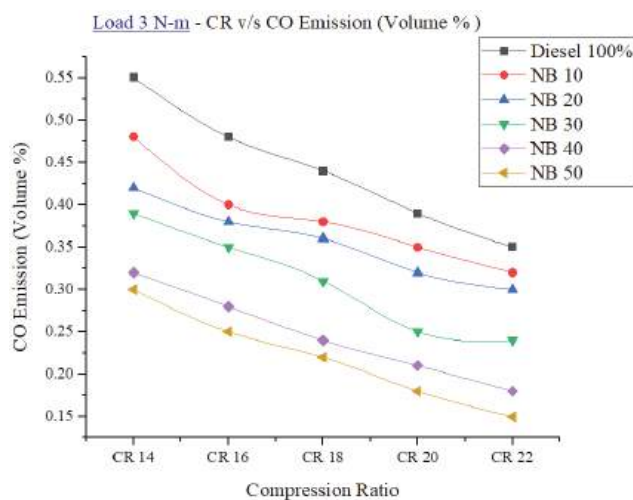


Figure 7. Effect of compression ratio on CO emission at 3 N-m load.

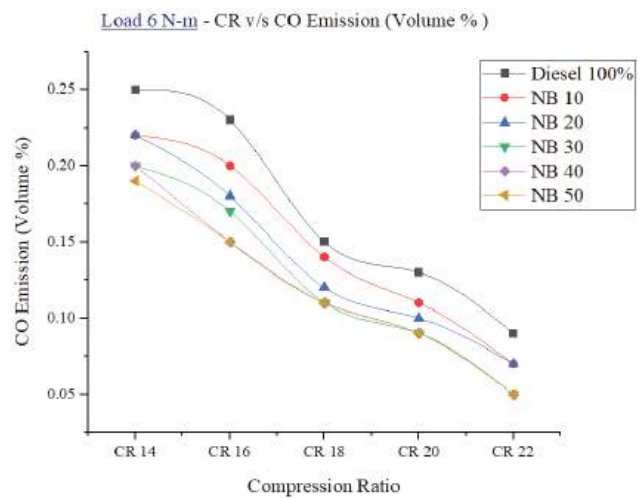


Figure 8. Effect of compression ratio on CO emission at 6 N-m load.

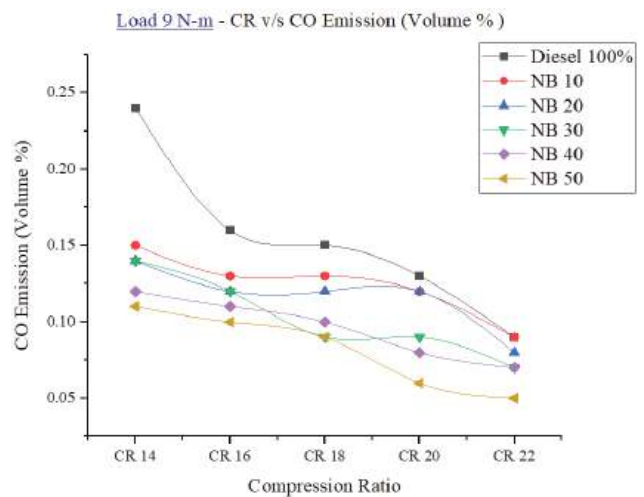


Figure 9. Effect of compression ratio on CO emission at 9 N-m load.

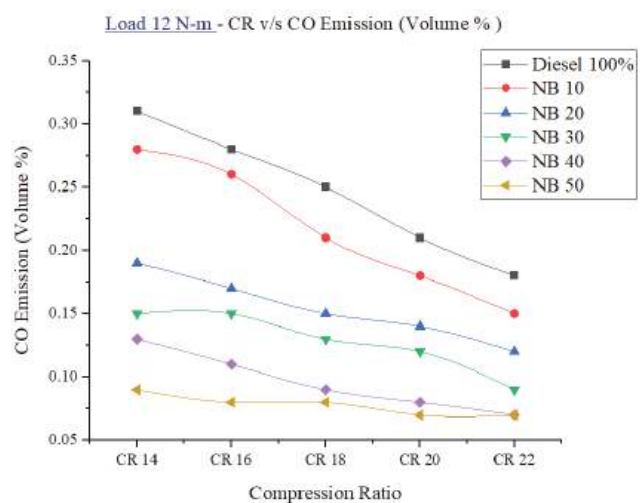


Figure 10. Effect of compression ratio on CO emission at 12 N-m load.

Emission of Carbon Monoxide (CO₂)

At lower compression ratios, the engine may not reach the ideal temperature and pressure for optimal combustion, resulting in less efficient fuel burning. To make up for this inefficiency, more fuel may be required to meet power demands. While incomplete combustion primarily affects CO emissions, increased fuel consumption typically leads to higher CO₂ emissions. On the other hand, higher compression ratios generally improve the engine's thermal efficiency, allowing it to convert more of the fuel's energy into useful work, rather than losing it as heat. As a result, less fuel is needed to generate the same amount of power, which typically results in lower CO₂ emissions per unit of power output. Under both zero load and full load conditions,

CO₂ emissions decrease significantly. The study shows that using neem oil biodiesel at a compression ratio of 22 results in an approximate 20% reduction in CO₂ emissions. This significant decrease can be attributed to the cleaner combustion process facilitated by the higher oxygen content in biodiesel and the improved combustion efficiency at higher compression ratios. Figure 11 to 15 shows the effect of variation in compression ratio on CO₂ emission at zero, 3, 6, 9 and 12 N-m loads for different blends.

Emission of Hydro Carbon (HC)

Hydrocarbon (HC) emissions refer to unburned or partially burned fuel that escapes the engine through the exhaust. At lower compression ratios, combustion may be less efficient due to reduced temperatures and pressures, which can result in incomplete burning of the fuel-air

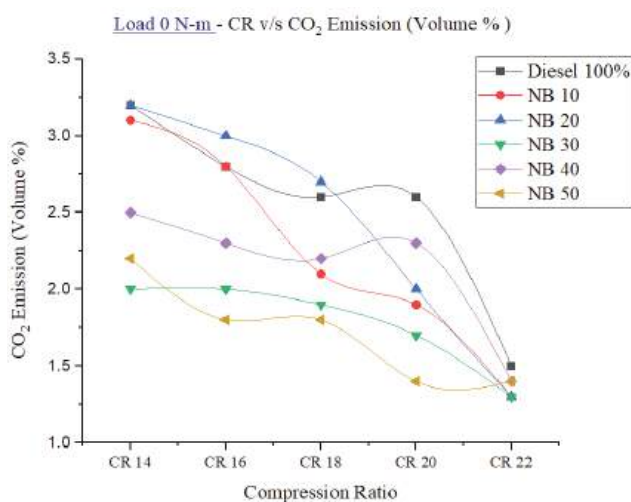


Figure 11. Effect of compression ratio on CO₂ emission at zero load.

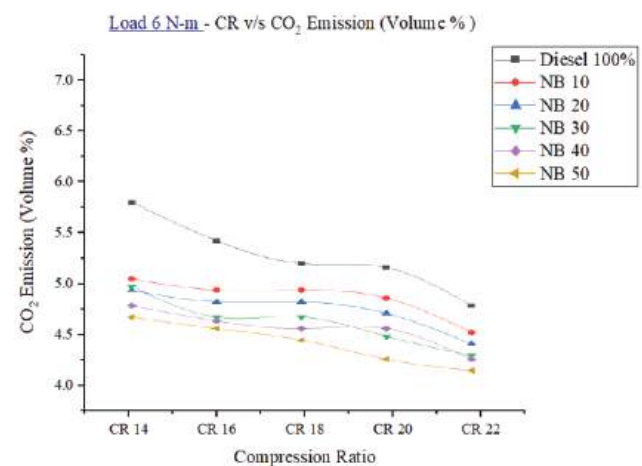


Figure 13. Effect of compression ratio on CO₂ emission at 6 N-m load.

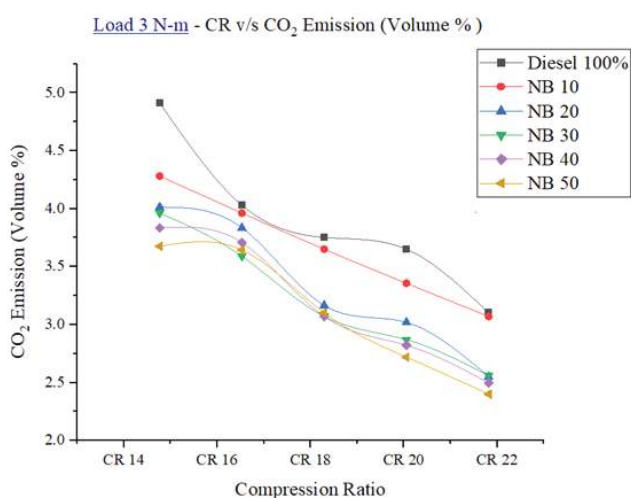


Figure 12. Effect of compression ratio on CO₂ emission 3 N-m load.

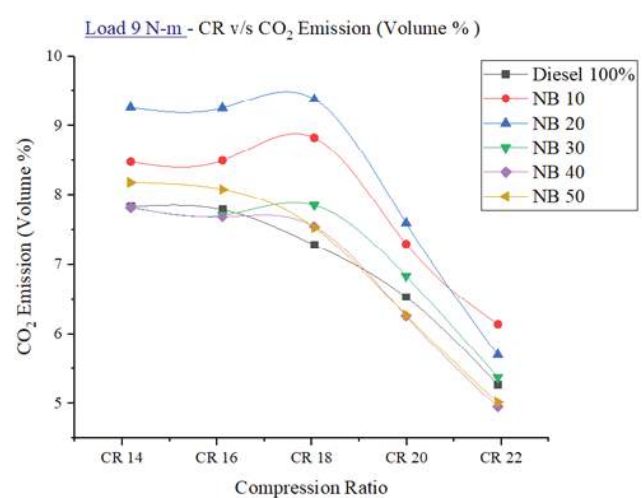


Figure 14. Effect of compression ratio on CO₂ emission at 9 N-m load.

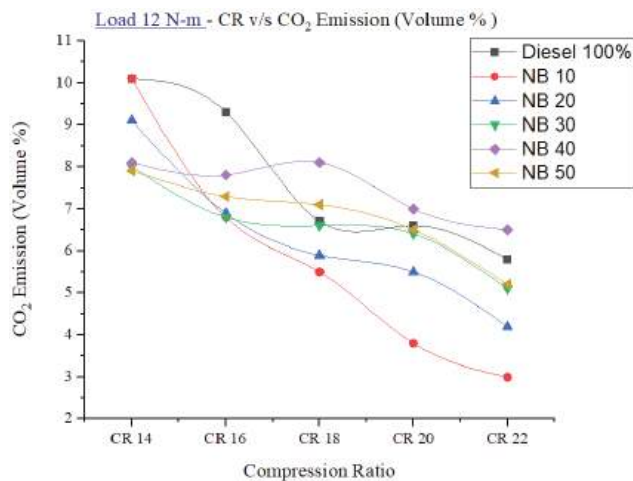


Figure 15. Effect of compression ratio on CO₂ emission at 12 N-m load.

mixture and lead to higher HC emissions. Under these conditions, more unburned or partially burned fuel is expelled as HC emissions. Increasing the compression ratio improves combustion efficiency by allowing the engine to operate at higher temperatures and pressures. This typically results in lower HC emissions, as more of the fuel is fully burned. However, if the compression ratio becomes too high, engine knocking or pre-ignition may occur, which could disrupt the combustion process and potentially increase HC emissions if not properly managed. Figure 16 to 20 shows the effect of variation in compression ratio on HC emission at zero, 3, 6, 9 and 12 N-m loads for different blends. A significant reduction in HC emissions is observed at zero, 3, 6, and 9 N-m loads with increased compression ratios and the use of neem oil biodiesel. However, there is no major

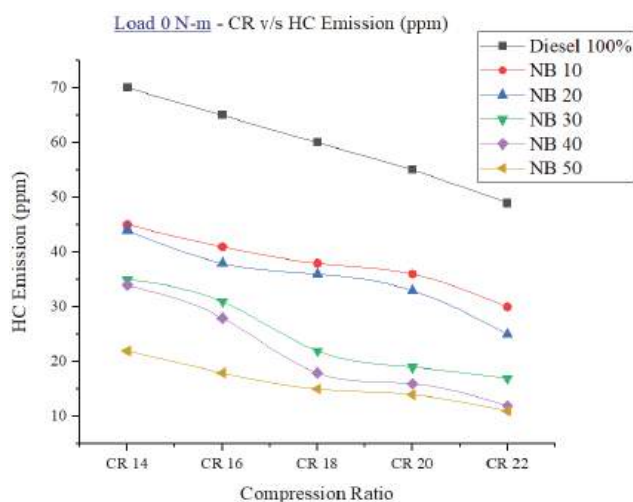


Figure 16. Effect of compression ratio on HC emission at zero load.

change in HC emissions under full load conditions. There is approximately a 27% reduction in hydrocarbon (HC) emissions when using a B30 neem oil biodiesel blend at a compression ratio of 20 under full load conditions.

Emission of Oxides of Nitrogen (NOx)

Nitrogen oxides (NOx) are pollutants produced in internal combustion engines through the reaction of nitrogen and oxygen at high temperatures. NOx emissions are a significant concern due to their contribution to smog formation and potential health risks. At lower compression ratios, combustion temperatures and pressures tend to be lower, which reduces NOx formation since these gases primarily form at higher temperatures where nitrogen and

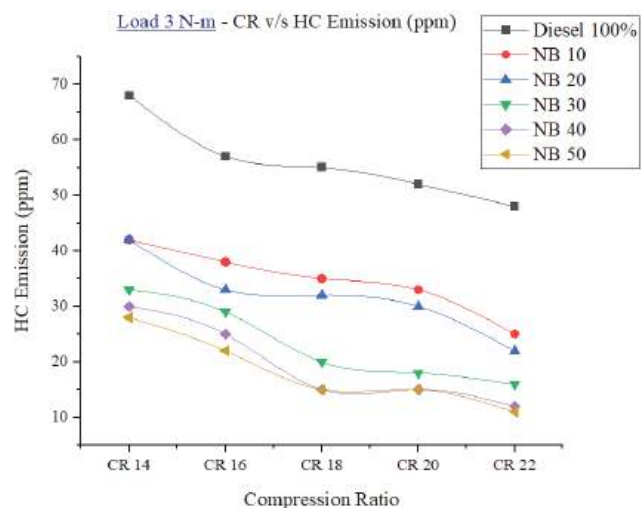


Figure 17. Effect of compression ratio on HC emission at 3 N-m load.

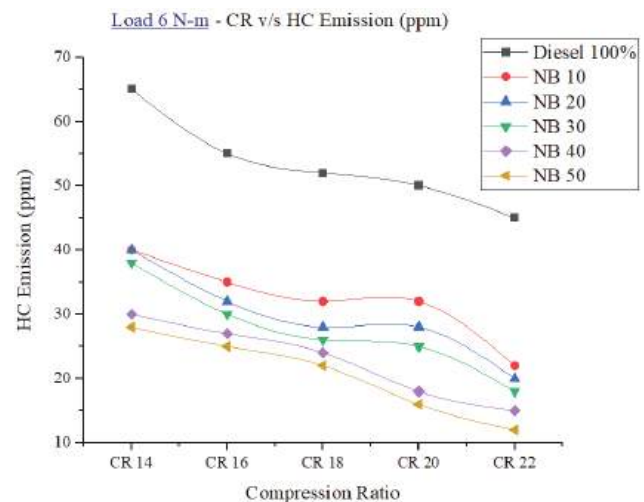


Figure 18. Effect of compression ratio on HC emission at 6 N-m load.

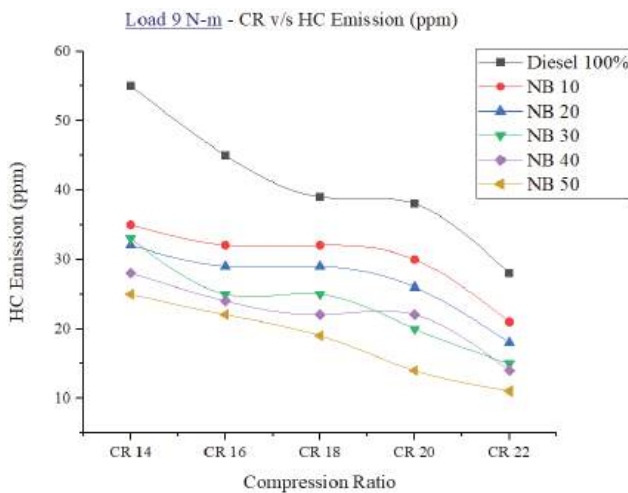


Figure 19. Effect of compression ratio on HC emission at 9 N-m load.

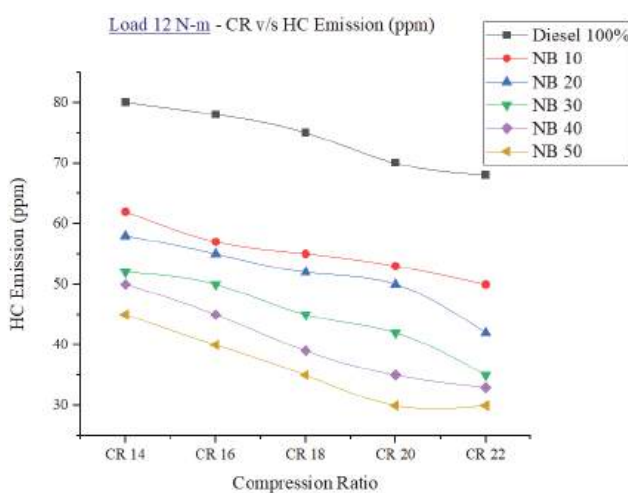


Figure 20. Effect of compression ratio on HC emission at 12 N-m load.

oxygen react more easily. When the compression ratio is increased, the engine operates at higher temperatures and pressures, accelerating the reaction between nitrogen and oxygen and thus increasing NO_x formation. Figure 21 to 25 shows the effect of variation in compression ratio on NO_x emission at zero, 3, 6, 9 and 12 N-m loads for different blends. As a result, NO_x emissions generally raise with higher compression ratios due to the elevated combustion chamber temperatures. This presents a notable trade-off when optimizing compression ratios. To address these issues, modern engines often incorporate technologies like exhaust gas recirculation (EGR), selective catalytic reduction (SCR), and inter cooling to manage and reduce NO_x emissions. It has been observed that under higher loads, engines with increased compression ratios experience

higher combustion temperatures, leading to elevated NO_x emissions. NO_x emissions increase by 20% at full load and a higher compression ratio of 22:1 when using pure diesel.

Brake Thermal Efficiency

Brake Thermal Efficiency (BTHE) measures how efficiently an engine converts the heat from fuel combustion into useful mechanical work. It is the ratio of the engine's power output to the heat energy supplied by the fuel. Engines with lower compression ratios generally operate at lower combustion temperatures and pressures, leading to less efficient fuel-to-work conversion. Figure 26 to 30 shows the effect of variation in compression ratio on BTHE at zero, 3, 6, 9 and 12 N-m loads for different blends.

As a result, more energy is lost as heat, and BTHE is lower. In contrast, higher compression ratios allow the

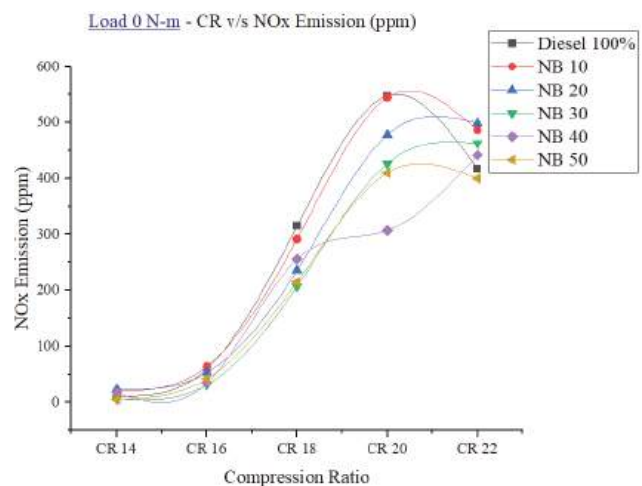


Figure 21. Effect of Compression ratio on NO_x emission at zero load.

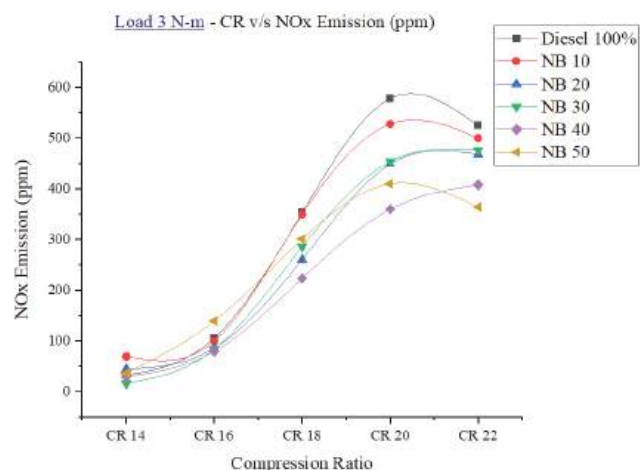


Figure 22. Effect of compression ratio on NO_x emission at 3 N-m load.

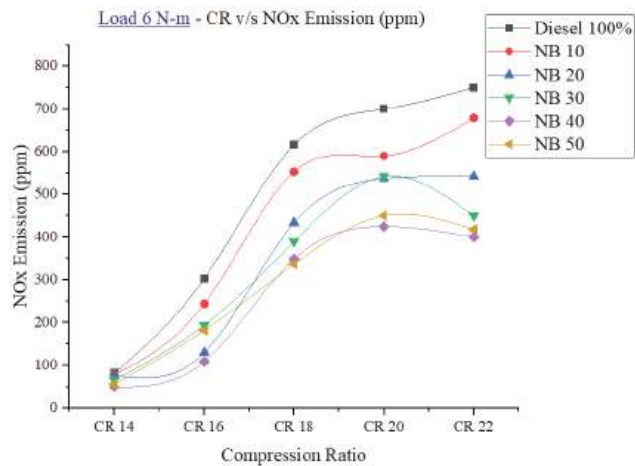


Figure 23. Effect of compression ratio on NO_x emission at 6 N-m load.

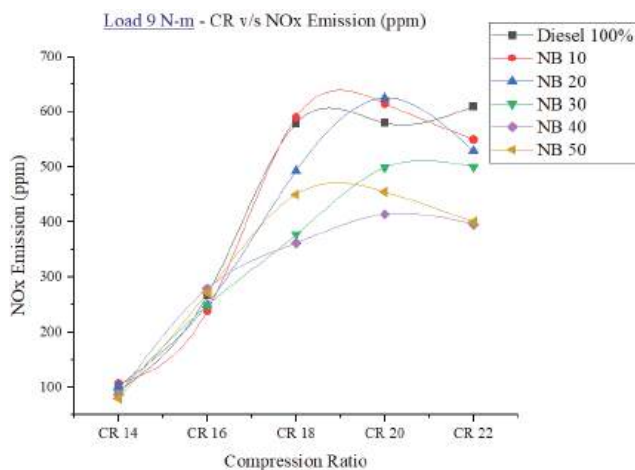


Figure 24. Effect of compression ratio on NO_x emission at 9 N-m load.

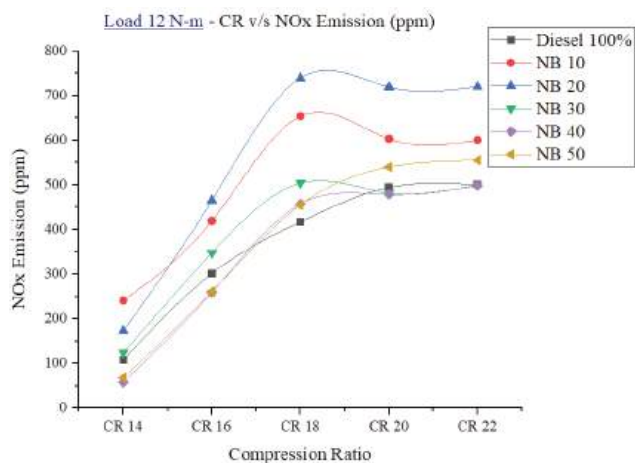


Figure 25. Effect of compression ratio on NO_x emission at 12 N-m load.

engine to operate at higher temperatures and pressures, enabling more complete fuel combustion and improving the conversion of thermal energy into mechanical work. BTHE tends to increase with biodiesel blending due to the higher oxygen content in biodiesel. This oxygen promotes more complete combustion, enhancing fuel utilization and power output, while the lower carbon content in biodiesel reduces unburned hydrocarbons, further boosting thermal efficiency. Similarly, increasing the compression ratio also improves BTHE by extracting more energy from the fuel through more efficient combustion, which reduces energy loss and converts more of the fuel's thermal energy into mechanical work.

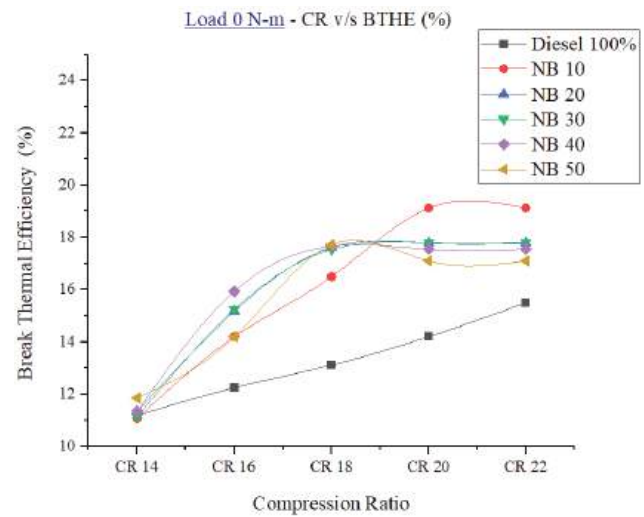


Figure 26. Effect of compression ratio on BTHE at zero load.

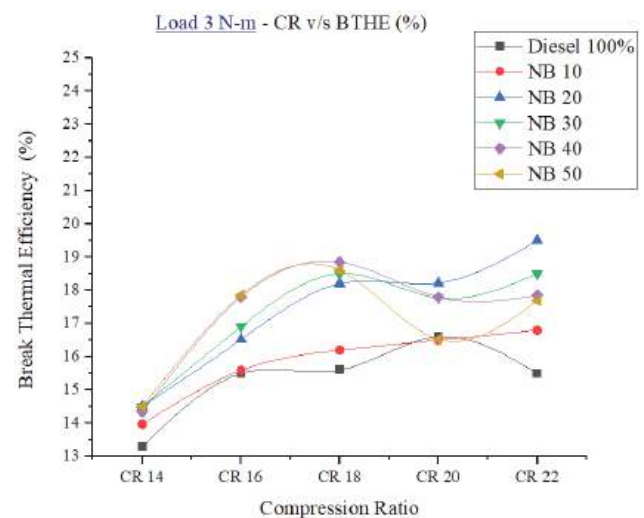


Figure 27. Effect of compression ratio on BTHE at 3 N-m load.

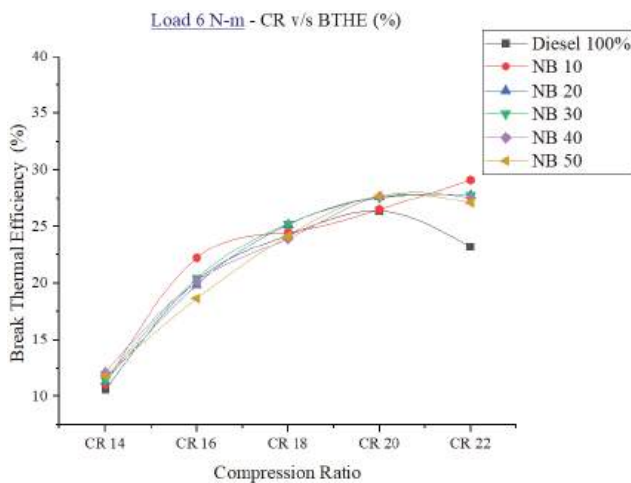


Figure 28. Effect of compression ratio on BTHE at 6 N-m load.

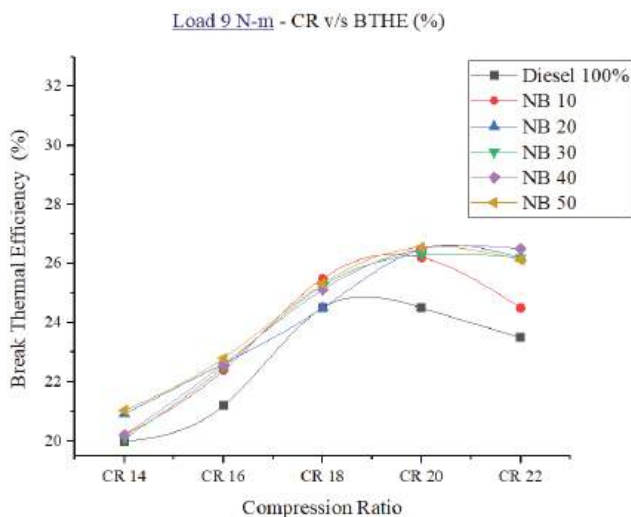


Figure 29. Effect of compression ratio on BTHE at 9 N-m load.

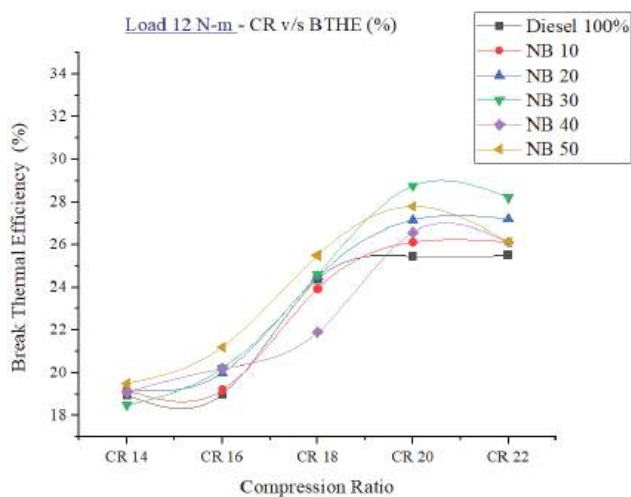


Figure 30. Effect of compression ratio on BTHE at 12 N-m load.

Specific Fuel Consumption

Specific fuel consumption (SFC) decreases with biodiesel blending due to biodiesel's higher oxygen content, leading to more complete combustion and improved fuel efficiency. Additionally, the blending reduces the need for excess fuel to achieve similar power output, contributing to lower SFC. SFC also reduces with an increase in compression ratio because higher compression enhances thermal efficiency. This is because more energy is extracted from the fuel due to better air-fuel mixture ignition, improving combustion and power generation, which ultimately reduces the fuel required per unit of power output. Thus, both factors contribute to improved engine efficiency. SFC decreases significantly under no-load conditions with a higher compression ratio and the use of NB50. Figure 31 to 35 shows the effect of variation in compression ratio on SCF at zero, 3 6, 9 and 12 N-m loads for different blends.

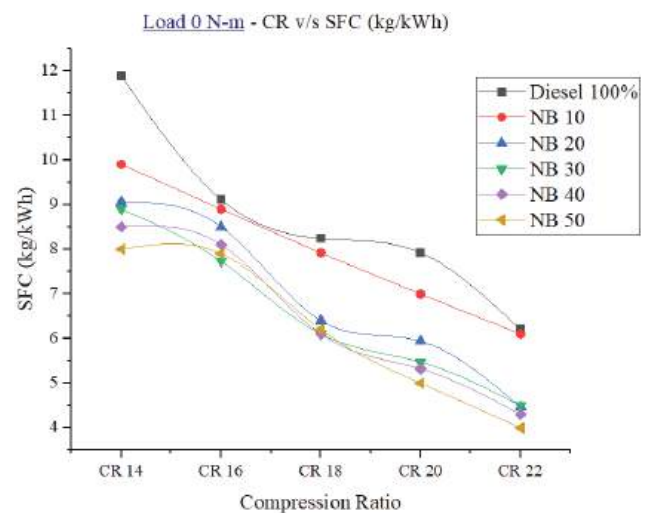


Figure 31. Effect of compression ratio on SFC at zero load.

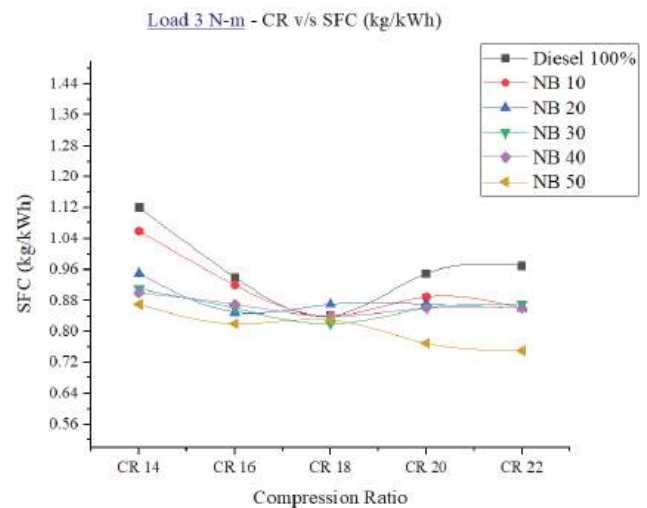


Figure 32. Effect of compression ratio on SFC at 3 N-m load.

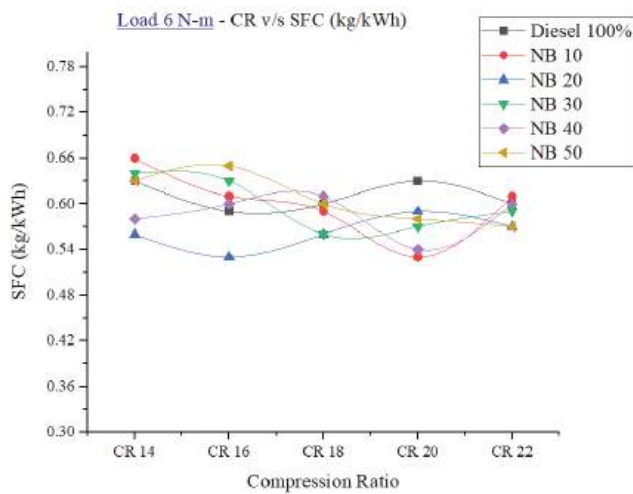


Figure 33. Effect of compression ratio on SFC at 6 N-m load.

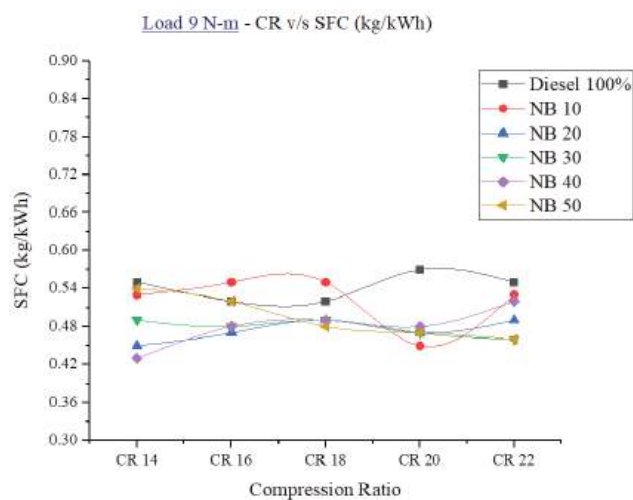


Figure 34. Effect of compression ratio on SFC at 9 N-m load.

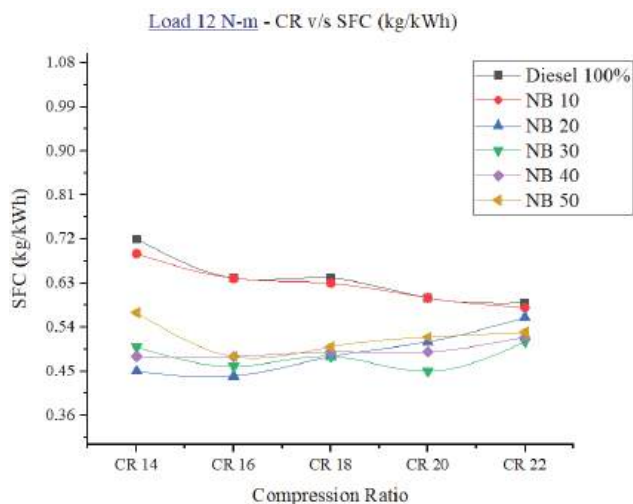


Figure 35. Effect of compression ratio on SFC at 12 N-m load.

Effect of Diesel Emissionson Environment and Health

Environmental impact and health impact

Air Pollution: Diesel engines emit several harmful pollutants, including nitrogen oxides (NO_x), particulate matter (PM), hydrocarbons (HC), and carbon dioxide (CO₂). NO_x contributes to the formation of ground-level ozone (smog), which reduces air quality and harms ecosystems.

Climate Change: Diesel engines release significant amounts of CO₂, a major greenhouse gas that contributes to global warming. Additionally, black carbon (a component of PM) from diesel exhaust absorbs sunlight and accelerates the melting of ice in Polar Regions, further contributing to climate change.

Acid Rain: NO_x emissions from diesel engines combine with atmospheric moisture to form acids, which contribute to acid rain. Acid rain damages forests, soil, and aquatic ecosystems, leading to biodiversity loss and degradation of water quality.

Respiratory Issues: Particulate matter (PM), especially fine particles (PM_{2.5}), can penetrate deep into the lungs, causing respiratory problems such as asthma, bronchitis, and chronic obstructive pulmonary disease (COPD). Prolonged exposure can reduce lung function and increase the risk of respiratory infections.

Cardiovascular Diseases: Long-term exposure to diesel emissions is associated with an increased risk of heart disease, high blood pressure, and stroke. PM and NO_x can contribute to inflammation and oxidative stress, leading to cardiovascular complications.

Cancer: Diesel exhaust is classified as a carcinogen by the International Agency for Research on Cancer (IARC). Long-term exposure to diesel emissions, particularly PM, has been linked to an increased risk of lung cancer and possibly bladder cancer.

Neurological Effects: Emerging research suggests that long-term exposure to diesel emissions may be linked to cognitive decline and neurodegenerative diseases like Alzheimer's, due to the presence of ultra-fine particles and toxic compounds in diesel exhaust.

TEACHING-LEARNING BASED OPTIMIZATION (TLBO) FOR MULTI-OBJECTIVE OPTIMIZATION

All testing inputs and output results are processed using MATLAB software to achieve optimal outcomes. An algorithm was developed with the assistance of Artificial Intelligence tools to obtain the most accurate results. The detailed program and optimized outcomes are outlined below.

```
clc;
clear;
close all;
```

```
% Define the number of learners (population size) and iterations
```

```
nLearners = 50;
```

```
maxIter = 100;
```

```
% Variable bounds (A, B, C)
```

```
A_min = 0; A_max = 12;
```

```
B_min = 0; B_max = 40;
```

```
C_min = 14; C_max = 22;
```

```
% Number of design variables
```

```
nVars = 3;
```

```
% Initialize population (learners)
```

```
pop = rand(nLearners, nVars);
```

```
pop(:,1) = A_min + (A_max - A_min) * pop(:,1); % A
```

```
pop(:,2) = B_min + (B_max - B_min) * pop(:,2); % B
```

```
pop(:,3) = C_min + (C_max - C_min) * pop(:,3); % C
```

```
% Objective function values for each learner
```

```
f_vals = zeros(nLearners, 6);
```

```
% Evaluate initial population
```

```
for i = 1:nLearners
```

```
    A = pop(i, 1);
```

```
    B = pop(i, 2);
```

```
    C = pop(i, 3);
```

```
    f_vals(i,1) = 1.0379 - 0.01091 * A - 0.002728 * B - 0.03762 * C; % f(X)
```

```
    f_vals(i,2) = -5.232 - 0.1025 * A - 0.03444 * B + 0.6424 * C; % f(Y)
```

```
    f_vals(i,3) = 32.25 - 1.387 * A - 0.7256 * B + 1.520 * C; % f(Z)
```

```
    f_vals(i,4) = -271.6 + 40.57 * A - 2.836 * B + 23.71 * C; % f(P)
```

```
    f_vals(i,5) = -16.21 + 0.1071 * A - 0.0190 * B + 1.0562 * C; % f(Q)
```

```
    f_vals(i,6) = 15.22 - 0.0765 * A - 0.0135 * B - 0.6956 * C; % f(R)
```

```
end
```

```
% Start TLBO process
```

```
for iter = 1:maxIter
```

```
    % Teacher Phase
```

```
    % Find the best solution (teacher)
```

```
    [~, idx] = min(sum(f_vals, 2)); % Sum of objectives (for simplicity)
```

```
    teacher = pop(idx, :);
```

```
    % Teaching factor Tf
```

```
    Tf = randi([1,2], 1);
```

```
    % Update learners
```

```
    for i = 1:nLearners
```

```
        new_pop = pop(i,:) + rand(1, nVars) .* (teacher - Tf * mean(pop));
```

```
    % Check variable bounds
```

```
    new_pop(1) = max(min(new_pop(1), A_max), A_min); % A
```

```
    new_pop(2) = max(min(new_pop(2), B_max), B_min); % B
```

```
    new_pop(3) = max(min(new_pop(3), C_max), C_min); % C
```

```
    % Evaluate new population
```

```
    A = new_pop(1);
```

```
    B = new_pop(2);
```

```
    C = new_pop(3);
```

```
    f_new(1) = 1.0379 - 0.01091 * A - 0.002728 * B - 0.03762 * C; % f(X)
```

```
    f_new(2) = -5.232 - 0.1025 * A - 0.03444 * B + 0.6424 * C; % f(Y)
```

```
    f_new(3) = 32.25 - 1.387 * A - 0.7256 * B + 1.520 * C; % f(Z)
```

```
    f_new(4) = -271.6 + 40.57 * A - 2.836 * B + 23.71 * C; % f(P)
```

```
    f_new(5) = -16.21 + 0.1071 * A - 0.0190 * B + 1.0562 * C; % f(Q)
```

```
    f_new(6) = 15.22 - 0.0765 * A - 0.0135 * B - 0.6956 * C; % f(R)
```

```
    % If the new solution is better, replace the old one
```

```
    if sum(f_new) < sum(f_vals(i,:))
```

```
        pop(i,:) = new_pop;
```

```
        f_vals(i,:) = f_new;
```

```
    end
```

```
end
```

```
% Learner Phase (pairwise learning)
```

```
for i = 1:nLearners
```

```
    j = randi([1, nLearners], 1);
```

```
    if sum(f_vals(i,:)) < sum(f_vals(j,:))
```

```
        new_pop = pop(i,:) + rand(1, nVars) .* (pop(i,:) - pop(j,:));
```

```
    else
```

```
        new_pop = pop(i,:) - rand(1, nVars) .* (pop(i,:) - pop(j,:));
```

```
    end
```

```
% Check variable bounds
```

```
new_pop(1) = max(min(new_pop(1), A_max), A_min); % A
```

```
new_pop(2) = max(min(new_pop(2), B_max), B_min); % B
```

```
new_pop(3) = max(min(new_pop(3), C_max), C_min); % C
```

```
% Evaluate new population
```

```
A = new_pop(1);
```

```
B = new_pop(2);
```

```
C = new_pop(3);
```

```
f_new(1) = 1.0379 - 0.01091 * A - 0.002728 * B - 0.03762 * C; % f(X)
```

```
f_new(2) = -5.232 - 0.1025 * A - 0.03444 * B + 0.6424 * C; % f(Y)
```

```
f_new(3) = 32.25 - 1.387 * A - 0.7256 * B + 1.520 * C; % f(Z)
```

```
f_new(4) = -271.6 + 40.57 * A - 2.836 * B + 23.71 * C; % f(P)
```

```
f_new(5) = -16.21 + 0.1071 * A - 0.0190 * B + 1.0562 * C; % f(Q)
```

```
f_new(6) = 15.22 - 0.0765 * A - 0.0135 * B - 0.6956 * C; % f(R)
```

```
% If the new solution is better, replace the old one
```

```
if sum(f_new) < sum(f_vals(i,:))
```

```
    pop(i,:) = new_pop;
```

```
    f_vals(i,:) = f_new;
```

```
end
```

```
end
```

```
% Display the best solution
```

```
 [~, idx] = min(sum(f_vals, 2));
```

```
best_sol = pop(idx, :);
```

```
best_f_vals = f_vals(idx, :);
```

```
fprintf('Best solution:\n A = %.4f, B = %.4f, C = %.4f\n', best_sol(1), best_sol(2), best_sol(3));
```

```
fprintf('Objective function values: f(X) = %.4f, f(Y) = %.4f, f(Z) = %.4f, f(P) = %.4f, f(Q) = %.4f, f(R) = %.4f\n', best_f_vals);
```

Optimum solution:

A = 0.0000, B = 40.0000, C = 22.0000

Objective function values: $f(X) = 20.2021$, $f(Y) = 23.8400$, $f(Z) = 24.5060$, $f(P) = -23.1000$, $f(Q) = -21.8320$, $f(R) = 24.9416$

Where, X represent CO emission, Y of CO₂, Z of HC, P of NOx, Q of BTHE and R of SFC. Here, minus sign represents the increase value in percentage.

UNCERTAINTY ANALYSIS

Due to the degree of uncertainty of the measuring equipment used, an uncertainty analysis is performed for the measurements of the engine performance parameters. For the development of the uncertainty analysis, the Type A evaluation method was used, which is based on the statistical analysis of a set of measurements ($x_1, x_2, x_3, \dots, x_n$). The best estimate of a measurement set ($\bar{x}$) is determined by equation (1).

$$\bar{x} = \frac{\sum_{i=1}^n x_i}{n} \quad (1)$$

The calculation of the dispersion of the measurements ($\bar{s}$) is carried out using the standard deviation, as shown in equation (2).

$$\bar{s} = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}} \quad (2)$$

The measurement uncertainty (μ) used is determined from the calculation of the mean, standard deviation, as shown in equation (3).

$$\mu = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n \cdot (n-1)}} \quad (3)$$

The experimental tests were replicated three times ($n=3$) for each mode of operation and each type of fuel since a higher number implies a longer experimentation time, which could aggravate the uncertainty of the measurements. The uncertainty of each measured parameter for CO is shown in Table 4. Whereas the uncertainty analysis for CO₂, HC, NOx, BTHE and SFC are presented in the Table 5 to 9. Summary of uncertainty in parameters of engine emission and performance is presented in Table 10.

In the experimental engine setup certain quantiles required to measure like temperature at various points, load variation, fuel flow, air flow, water flow, speed etc. using sensors. The all sensors are checked and calibrated for all measuring quantities of temperature ($\pm 0.5\%$), load ($\pm 0.25\%$ of F.S), fuel flow ($\pm 0.1\%$ of span), air flow ($\pm 0.5\%$),

Table 4. Uncertainty analysis of CO

Step 1	Step 2	Step 3
$\bar{x} = \frac{\sum_{i=1}^n x_i}{n}$	$\bar{s} = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}}$	$\mu = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n \cdot (n-1)}}$
$\bar{x} = \frac{34.36}{150}$	$\bar{s} = \sqrt{\frac{5587.5}{149}}$	$\mu = \sqrt{\frac{5587.5}{22350}}$
$\bar{x} = 0.22$	$\bar{s} = \sqrt{37.5}$	$\mu = \sqrt{0.2567}$
	$\bar{s} = 6.12$	$\mu = 0.50$

Table 5. Uncertainty analysis of CO₂

Step 1	Step 2	Step 3
$\bar{x} = \frac{\sum_{i=1}^n x_i}{n}$	$\bar{s} = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}}$	$\mu = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n \cdot (n-1)}}$
$\bar{x} = \frac{741.9}{150}$	$\bar{s} = \sqrt{\frac{710.26}{149}}$	$\mu = \sqrt{\frac{710.26}{22350}}$
$x = 4.96$	$\bar{s} = \sqrt{4.766}$	$\mu = \sqrt{0.03177}$
	$\bar{s} = 2.18$	$\mu = 0.17$

Table 6. Uncertainty analysis of HC

Step 1	Step 2	Step 3
$\bar{x} = \frac{\sum_{i=1}^n x_i}{n}$	$\bar{s} = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}}$	$\mu = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n \cdot (n-1)}}$
$\bar{x} = \frac{5142}{150}$	$\bar{s} = \sqrt{\frac{73239.38}{149}}$	$\mu = \sqrt{\frac{73239.38}{22350}}$
$\bar{x} = 34.28$	$\bar{s} = \sqrt{491.53}$	$\mu = \sqrt{3.27}$
	$\bar{s} = 22.17$	$\mu = 1.81$

Table 7. Uncertainty analysis of NO_x

Step 1	Step 2	Step 3
$\bar{x} = \frac{\sum_{i=1}^n x_i}{n}$	$\bar{s} = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}}$	$\mu = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n \cdot (n-1)}}$
$\bar{x} = \frac{50046}{150}$	$\bar{s} = \sqrt{\frac{62156}{149}}$	$\mu = \sqrt{\frac{62156}{22350}}$
$\bar{x} = 333.64$	$\bar{s} = \sqrt{417.15}$	$\mu = \sqrt{2.78}$
	$\bar{s} = 20.42$	$\mu = 1.66$

Table 8. Uncertainty analysis of BTHE

Step 1	Step 2	Step 3
$\bar{x} = \frac{\sum_{i=1}^n x_i}{n}$	$\bar{s} = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}}$	$\mu = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n \cdot (n-1)}}$
$\bar{x} = \frac{3043.75}{150}$	$\bar{s} = \sqrt{\frac{3723}{149}}$	$\mu = \sqrt{\frac{3723.8}{22350}}$
$\bar{x} = 20.29$	$\bar{s} = \sqrt{24.99}$	$\mu = \sqrt{0.166}$
	$\bar{s} = 4.99$	$\mu = 0.40$

Table 9. Uncertainty analysis of SFC

Step 1	Step 2	Step 3
$\bar{x} = \frac{\sum_{i=1}^n x_i}{n}$	$\bar{s} = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}}$	$\mu = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n \cdot (n-1)}}$
$\bar{x} = \frac{289.13}{150}$	$\bar{s} = \sqrt{\frac{1117.23}{149}}$	$\mu = \sqrt{\frac{1117.}{22350}}$
$\bar{x} = 1.92$	$\bar{s} = \sqrt{7.49}$	$\mu = \sqrt{0.0499}$
	$\bar{s} = 2.73$	$\mu = 0.22$

Table 10. Uncertainty in parameters of engine emission and performance

Measurement	Uncertainty
CO	±0.5
CO ₂	±0.17
HC	±1.81
NO _x	±1.66
SFC	±0.22
Break thermal efficiency	±0.40

water flow(±2% of full), speed (±0.25%). The parameters measured are within specified accuracy range of the respective sensors. The calibration certification of all measuring devices is provided by the Apex Innovations Pvt. Ltd., Sangli, Maharashtra.

CONCLUSION

The following conclusions are drawn from the current experimental study:

- Experimental observations indicate that CO emissions decrease with an increase in biodiesel blending, as well as with a higher compression ratio. The most significant reduction in CO emissions is was observed with a 50% neem biodiesel blend (NB50) at a moderate load of 6 N-m and a compression ratio of 22, achieving a 28% decrease.
- CO₂ emissions are reduced by 20% when using a B40 biodiesel blend at a compression ratio of 22 under full load conditions of 12 N-m. This reduction can be attributed to the enhanced combustion efficiency and the cleaner-burning properties of the biodiesel blend, leading to lower carbon dioxide output during operation.
- Approximately a 27% reduction found in hydrocarbon (HC) emissions when using a B30 neem oil biodiesel blend at a compression ratio of 20 under full load conditions.
- NO_x emissions increase by 20% at full load and a higher compression ratio of 22:1 when using pure diesel. To address these issues, modern engines often incorporate technologies like exhaust gas recirculation (EGR), selective catalytic reduction (SCR), and inter cooling to manage and it can reduce NO_x emissions.
- Brake Thermal Efficiency increases with both load and compression ratio. A 16% increase in BTHE was observed when using NB30 at a 6 N-m load with a compression ratio of 22:1.
- SFC decreases significantly under no-load conditions with a higher compression ratio (22:1) and the use of NB50.

- Optimal results from AI-TLBO at no-load conditions using NB40 fuel and a compression ratio of 22 show a 20.20% reduction in CO emissions, a 23.84% decrease in CO₂, and a 24.5% reduction in HC. However, NO_x emissions increased by 23.1%, while BTE improved by 21.82%, and SPC decreased by 24.94% compared with 100 % diesel fuel.

NOMENCLATURE

BP	Brake Power
BMEP	Brake Mean Effective Pressure
BSFC	Break Specific Fuel Consumption
BTE	Brake Thermal Efficiency
IMEP	Indicated Mean Effective Pressure
IP	Indicated Power
MEP	Mean Effective Pressure
Nm	Newton-meter (Torque)
RPM	Revolutions Per Minute
NO _x	Nitrogen Oxides
PM	Particulate Matter
CO	Carbon Monoxide
NaOH	sodium Hydroxide
HC	Hydrocarbons
CO ₂	Carbon Dioxide
O ₂	Oxygen
VCR	Variable Compression Engine
BSFC	Brake Specific Fuel Consumption
JTME	Jetropha Methyl Ester
UBH	Unborn Hydro Carbon
EGR	Exhaust Gas Recirculation
LPG	Liquidity Petroleum Gas
CNG	Compressed Natural Gas
H ₂ SO ₄	Sulfuric Acid
CR, R	Compression Ratio
η	Efficiency
SFC	Specific Fuel Consumption
ppm	Parts Per Million
MENO	Methyl Easter Neem Oil
ASTM	American Society for Testing and Materials
NB10	Neem oil base biodiesel (10 % Neem oil and 90 % Diesel)
NB20	Neem oil base biodiesel (20 % Neem oil and 90 % Diesel)
NB30	Neem oil base biodiesel (30 % Neem oil and 90 % Diesel)
NB40	Neem oil base biodiesel (40 % Neem oil and 90 % Diesel)
NB50	Neem oil base biodiesel (50 % Neem oil and 90 % Diesel)
TLBO	TEACHING-LEARNING BASED OPTIMIZATION

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DISCLOSURE OF INTEREST

The authors report there are no competing interests to declare.

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ETHICS

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

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Review Article

Operational, design, and economic factors impacting gravitational water vortex hydropower systems

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ABSTRACT

This work investigates and assesses the main operational, design, and economic factors that influence the performance of Gravitational Water Vortex Hydropower Systems and their overall contribution to energy sustainability. The thesis highlights improving performance through turbine blade angle, basin design, and material selection -an essential aspect to overcome worldwide energy problems. The system's power output is evaluated based on experimental, numerical, and theoretical approaches for varying the flow rate, vortex height, and turbine configuration. The findings reveal that five blades with a 44° angle provide an efficiency of only 82%, and an advanced material-based turbine shows an improvement in torque of 1.23% concerning conventional components. Moreover, combining conical basins with optimized nozzles can gain power (60%) and reduce energy losses. The result helps develop renewable energy creative methods in regions with limited energy access. This study extends previous works in the scientific literature by proposing a complete analysis and optimization framework for turbine design and performance under the need for sustainable energy production. These findings lay the groundwork for future studies designed to enhance the efficiency of small hydropower plants and further the world's move to low-carbon energy.

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INTRODUCTION

Over the past two decades, a complex geopolitical dilemma has emerged that threatens energy security at the global level. In this case, scientists and researchers must intensify their efforts to rely solely on renewable energy sources to achieve energy independence [1]. The global fuel crisis, increasing costs, climate change, and global warming impact every global economy. These events also affect smaller nations with open economies. In a world where 1.4

billion people do not have access to power, 2.7 billion continue to use traditional biomass as their primary cooking fuel. Most people living in these energy-poor areas are rural populations in developing Asian countries (except China), desert Africa, and India [2]. Therefore, these areas' populations are vulnerable to climate change effects. Developing countries may cross traditional energy sources and go straight to low-carbon renewables, which will help reduce poverty. Thus, lowering emissions of global warming gases,

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increasing access to electricity, and reducing poverty in rural areas may all work side by side, creating a connection between the three issues. Efforts to switch to low-carbon energy systems in rural regions can lead to more significant gains than industrialized ones. The planet's finite resources are under pressure due to our current dependence on non-renewable energy sources. The researchers set out to compare the social, economic, and physical factors that encourage conversion to hydropower, wind, and solar energy [3]. Most renewable energy resources are unpredictable and vary due to climate. Academic research on renewable energy is growing in technological, economic, political, and social dimensions. Many disciplines study how to invest in renewable energy, develop and adopt policies, select renewable energy suitable for the location, establish it by considering economic and environmental factors, develop energy distribution and storage systems, and support regional development. Understanding these co-benefits can help consumers embrace sustainable energy [4]. By 2050, renewable energy sources have the potential to eliminate up to 90% of carbon emissions, leading to a significant reduction in greenhouse gases and aiding in the mitigation of climate change. The future of renewable energy looks bright with the implementation of the zero-carbon emission decarbonization idea. This concept can replace energy obtained from fossil fuels and effectively restrict the global temperature increase to 1.5 C by the year 2050 [5].

Hydropower is a renewable and clean energy source that many countries worldwide rely on to significantly reduce emissions and achieve carbon neutrality. Despite the challenges faced by hydropower, or what is known as generating electricity from water, as a result of climate change, which leads to growing drought around the world, hydropower accounted for 37% (1.2 Terawatts) of the installed renewable energy capacity globally (3.37 Terawatts), by the end of 2022, according to estimates from the International Renewable Energy Agency. Hydropower, or electrical power generation by harnessing the gravitational force of falling or flowing water, is commonly known as hydroelectricity. As a renewable energy source, hydroelectricity is competitive due to its cheap cost. However, constructing huge dams and reservoirs frequently necessitates the displacement of humans and animals, and damming can damage local ecosystems by interrupting river flows. Compared to power stations that rely on fossil fuels, hydroelectric complexes generate far less greenhouse gas emissions and no direct waste when construction is complete [6]. One of the most significant renewable energy sources in the world is hydropower.

Nevertheless, there are social and environmental costs associated with its development. Hydropower generation may be adversely affected by environmental deterioration and climate change. To overcome these obstacles, a sustainable hydropower project requires meticulous planning and system design. Sustainable energy may be a byproduct of well-planned hydropower projects. Energy planners,

investors, and everyone interested in hydropower projects need access to current information to make educated judgments [7]. Recently, focus has been placed on exploiting the energy of running water without the need for substantial hydraulic facilities, as a small river is used to build a small or medium-sized hydroelectric power station. Developed and emerging nations continue to prefer micro-strongly and pico-hydropower, two of the many renewable energy sources already on the market. Since the so-called Gravitational Water Vortex Hydropower Station (GWVHS) was developed in 2006, there has been a significant surge in the utilization of artificial free-surface vortices to produce low and ultra-low-head hydropower. The idea behind the concept is to create a vortex chamber and use the high angular velocity inside the whirlpool core to generate hydroelectric power [8]. The gravitational water vortex power plant is a micro hydro vortex turbine system that uses a low hydraulic head of (0.7–3) m to convert energy in a moving fluid to rotational energy. This system foundation is a circular basin with a central drain. A water turbine is powered by the stable line vortex that the water creates above the drain. The need to learn about new and improved renewable energy sources is pressing in our age because energy consumption is rising alarmingly. Water, wind, sun, and other forms of renewable energy may be harnessed. However, the main emphasis is on extracting energy from hydropower, namely by utilizing gravitational vortex hydro-turbines, in which the centrifugal force of a vortex is harnessed and converted into shaft power. Water flows tangentially into a circular basin at a gravitational vortex power station, and a turbine harnesses the energy released by the free vortex. The three primary benefits of GWVHS are its low hydraulic head power generation capability, minimal environmental impact, and economic and social viability [9]. Some studies have reported that the turbine design and operational parameters affected the efficiency of gravitational water vortex systems. For instance [10], showed that adjusting blade angles to 44° lifted efficiency to 82%. Similarly, [11] showed a 14% gain in efficiency via optimized runner profiles, while [12] showed a 10.25% torque gain by using baffles of 50% of the area of the turbine. The unit is designed to operate near a water stream. It includes a vortex hydro turbine system, a small hydro power plant gravity water vortex as follows: 1-supply channel, 2-hydrogen generator, 3-hydraulic Turbine, 4-circular basin (turbine chamber), 5-outlet channel and as shown in Fig.1 [13].

The research will focus on the research gap in the efficiency of the gravitational water vortex power plant. The operations gap is characterized by an insufficient understanding of how the variables of water depth, flow speed, and flow rate impact the performance of the system. In contrast, the design gap relates to the lack of acute analysis of important design components like the shape of the turbine or basin to deliver performance improvements. One missing piece of the puzzle is an economic overview of the financial viability of these plants, particularly in terms of

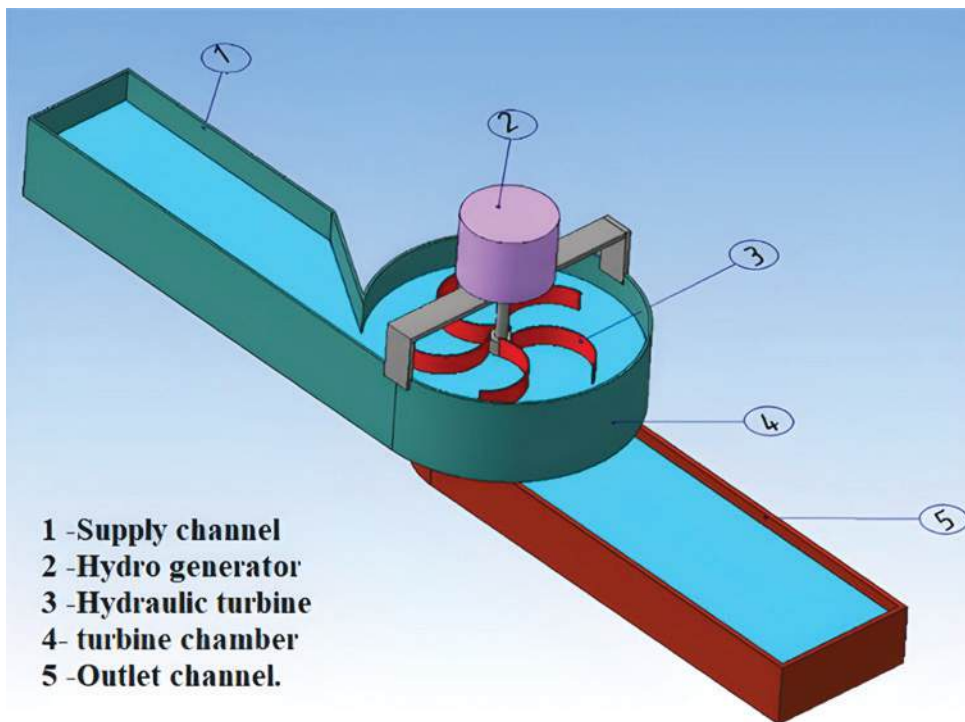


Figure 1. The main parts used in the GWVHS system sequentially [From Obozov et al. [13], with permission from Elsevier].

sustainability, given the huge economic gap. Therefore, the study proposal aims to identify the operational variables affecting the efficiency analysis and design of critical parts to select the best solution. It also includes an economic viability study focusing on reducing costs and sustainability. It also underlines the challenges those systems face and presents recommendations to strengthen current designs and increase their effectiveness.

Addressing the Gaps in Hydropower Efficiency: Design, Performance, and Economic Feasibility in Gravitational Water Vortex Systems

Although previous works address specific elements such as turbine improvement or the economic viability of the energy system, a holistic study covering design, performance, and economics is yet to be found in the literature. While these small hydropower systems have been the subject of some studies aimed at performance improvement, such studies failed to consider important variables like the angle of the blade for various designs, basin configuration, and the material for the constitutive element, thus limiting their contributions for offering a practical and sustainable alternative. Certain gaps to fill are directly related to the thorough analysis of system performance by considering different design parameters, such as advanced materials, since by including these, greater optimizations toward energy efficiency can be achieved. This research aims to address these research voids by offering a detailed analysis concentrating on specially crafted turbines to maximize

the production of energy as well as minimize the loss of energy. It requires developing new turbine designs using epoch-making new materials and increasing efficiency by the coexistence of conical basins with optimized nozzles. Large-scale energy storage also plays a vital role in renewable energy distribution and, as a result, acts as a lifeline to those parts of the world suffering from energy shortages, contributing to clean energy deployments and sustainability goals. Results show that we can develop more efficient hydropower systems, thus increasing energy production in remote areas and contributing to energy independence by reducing reliance on fossil fuels. The study extends previous research with a holistic framework for technologies used to enhance turbine performance in sustainable energy initiatives that will assist in meeting low-carbon energy targets.

MATERIALS AND METHODS

Amid growing requirements for new sustainable and energy-efficient energy sources to meet our environmental and economic needs, we found one of the innovative solutions that can help us with energy independence and low greenhouse gas emission — gravitational water vortex power plants. This technology is based on harnessing the gravitational forces and the kinetic energy of the water, allowing for continuous vortices capable of efficiently driving turbines. However, considering the great potential of such plants, their real performance depends

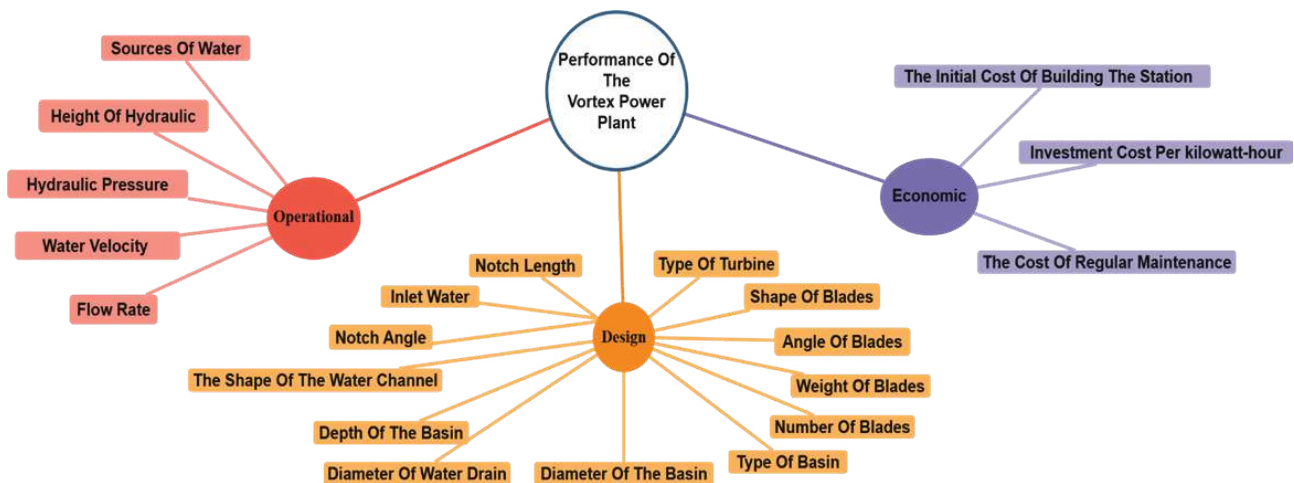


Figure 2. This diagram shows the most important factors that impact the performance of the vortex power plant.

on many operational, design, and economic parameters that necessitate in-depth analysis and investigation to validate better efficiency and cost-effectiveness. This study strives to fill the gaps in knowledge in this area by adopting a systematic and multidimensional analysis of the respective influencing factors, which may facilitate guiding the future development and enhancement of plant designs. This article reviews previously published studies specialized in exploiting and evaluating the performance of the vortex power plant by studying the operational, design, and economic impacts, as shown in Figure 2.

Articles will then be categorized, reviewed, and discussed according to these influences. The third Section clarified the role and impact of operational specifications on the GWVHS performance. Section (4) includes a review of studies that examined the effects of design variables on the performance and configuration of the vortex. Likewise, studies that addressed the economic impacts of the construction and design of the station were included in section (5). In this sixth Section, the modeling and simulation articles were clarified, and some improvements used in the system were presented, such as the angle of the blades, the basin, the antenna core, and other different effects, and the impact of each of them on the performance of the system. Then, the current limitations and challenges within hydrographic hydropower systems are clarified in section (7). In the (8) section, show results and recommendations.

OPERATIONAL VARIABLES

This section provides information about several operational effects, including height, water speed, volumetric flow rate, flow rate, air core diameter, and free rotation constant, followed by a discussion on each point. An efficient model for the analysis of flattened water vortices was built by Guzman et al. [14] using just the water head and geometric factors as inputs. The model found that the simulated

results matched experimental findings very well, and it computed the maximal volumetric flow rate, air core diameter, and free rotation constant. Zeng et al. [15] studied the turbulent flow of Francis turbine tubes experimentally and computationally. And by using the FLUENT program in this research. This allowed them to simulate the flow of the intake pipe to calculate the three operating States of the turbine correctly. Obozov et al. [13] found a significant match in the data through comparison. Examined the operational characteristics of spinning turbines and assessed the degree to which rotational speed influences load and efficiency. The researchers also evaluated the impact of the central gap in the turbine blades on their efficiency using the KompasFlow program. The findings revealed that the rotational speed decreases with increasing load regardless of the turbine types employed. Additionally, a specific loading period was identified during which maximum efficiency is attained. Saleem et al. [16] examined the effects of flow rate, vortex height, shaft diameter, blade position, and incision angle using different blade shapes and configurations. They concluded that optimal results are achieved when the vortex reaches its maximum height, the incision angle is reduced, and the blade is placed at the minimum possible distance from below the basin. Nasuki et al. [17] conducted an experimental investigation where they manipulated the flow rate and water pressure variables to examine their effect on the operational efficiency of the vortex power plant. The goal was to study the impact of different water flow rates and pressure at a fixed location on the turbine rotation speed. The test results, measuring water discharge into a closed basin, reveal an average discharge value flow rate of 81.08 liters/s. Kueh et al. [18] concluded that the behaviour of the water vortex influences the choice of turbine blades for a water vortex power plant. They studied two scenarios, each with a hole of different sizes for water flow. The vortex height corresponds to the first case's experimental and computational fluid dynamics results. The experimental result

and the final vortex height predicted by the computational fluid dynamics model vary for the second case. The computational fluid dynamics model makes more and more mistakes in its forecasts to create a water vortex as the discharge hole grows larger and turbulent flow enters the system. Nishi et al. [19] studied the flow field and gravity vortex water turbines experimentally and computationally, as shown in Figure 3. It is proven that raising the runner inlet rotational speed increases the onward flow area. However, the backflow area grows when the volume of available air decreases. Also, they examined the effect of flow rates on the performance of gravity vortex-type water turbines, and the results showed that the efficiency of the head and the effective turbine increases with the flow rate, resulting in increased turbine output. The most common losses in water turbines were tank and tank outlet losses, followed by friction losses inside the tank. Powalla et al. [20] created a verified digital model of the vortex hydroelectric power station that can be used as a digital twin for further analyses, such as hydraulic property assessments. Three-dimensional acoustic Doppler velocimetry was used to evaluate the flow field. The fluid volume method has also been used to characterize surface free flow. This proves that the model can mimic complex vortex power plant flow conditions. Faraji et al. [21] investigated how four variables (speed, hub-, blade, angle, number of blades, and runner profile) impact GWVHS efficiency. The numerical and experimental results were consistent, with coefficient of determination

values of 0.95 and 0.96 for flat and curved shapes, respectively. As seen in the selected factors, they interact to affect the efficiency of the GWVHS. The efficiency of the curved blade has been enhanced by 3.65% through digital analysis. The flat runner profile showed an increase in efficiency of 1.69% compared to unimproved conditions. Vertical-axis hydrokinetic turbines have emerged as a possible alternative for harnessing low-velocity currents. Consequently, a model of a vertical-axis turbine has been developed and fabricated to conduct experiments in an open channel by Yosry et al. [22]. The primary objective of this simulation and study was to accurately depict the air-water interface and examine the effect of fluctuations in the free surface on turbine output. Experimental investigations subjected the turbine model to different flow conditions and free surface levels. It has been observed that the maximum power coefficients show an increase in conjunction with the upstream speed.

Pamuji et al. [23] examined the effect of changes in the vortex pool. The configuration is based on operational efficiency and flow characteristics of the gravitational water vortex power plant. The cylindrical vortex basin is modified to become a conical basin by treating the ratio between the inlet and outlet diameters (D_{in}/D_{out}) with values of 3.26, 4.9, and 6.12. The research shows that the most significant speed profile is near the discharge hole. Moreover, it indicates that an increase in the ratio of the inner diameter to the outer diameter (D_{in}/D_{out}) or a decrease in the

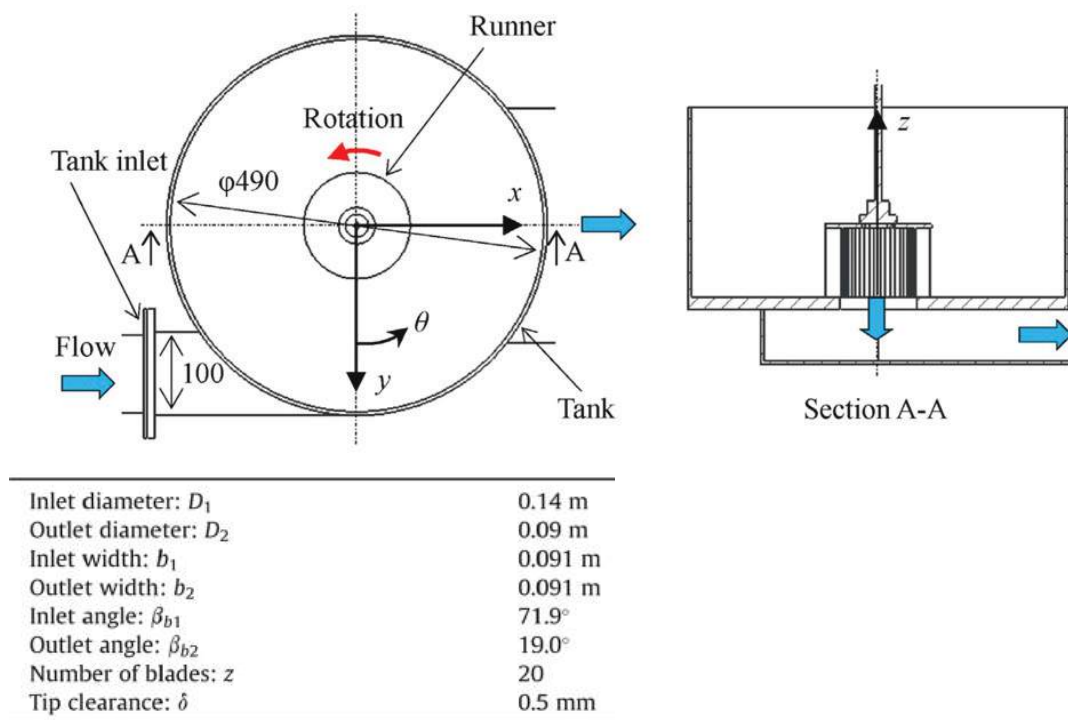


Figure 3. An illustrative view of the gravity vortex type water turbine tested, which is mainly composed of the runner and a tank, as well as the specifications of the test runner [From Nishi [19], with permission from Elsevier].

Table 1. A concise synopsis of research papers on the topic of enhancing the efficiency of vortex water plants

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Obozov et al.	[13]	2023	Kyrgyzstan	Experimental study	Findings show that, for all investigated turbine types, rotational speed drops with increasing load and that maximum efficiency is attained at a specific load interval.	Efficiency drops with increasing load; specific load intervals optimize performance.	Rotational speed decreases with load increase, highlighting the need for load optimization.	Limited turbine type and load conditions were tested; broader studies are needed for validation.
Guzmán et al.	[14]	2021	Peru	Analytical and simulated study	This model adequately describes the relationship between depth and discharge in a full-wide vortex. This feature makes the model suitable for both the analysis and design of new vortex systems and existing ones with fixed geometry.	The model was validated for depth-discharge relationships and is suitable for vortex design and analysis.	The framework is reliable for fixed geometry systems applicable to new and existing designs.	Relies on simulation; real-world validation is needed.
Zeng et al.	[15]	2012	China	Analyzed and numerically simulated the study	When it comes to improving power quality, predicting the unit's stable operating zone, designing the ideal turbine, and ensuring the unit's stable operation, these studies are crucial. They concluded Optimal outcomes are achieved when the vortex attains its greatest height, the slit angle is minimized, and the blade is positioned at the lowest feasible distance from the pelvis bottom.	Predicted stable operational zones; critical for turbine design and power quality improvement	Accurate zone prediction enhances turbine performance and operational reliability.	Simulation results are not generalized across different turbine designs and setups.
Salim et al.	[16]	2020	Indonesia	Experimental study	Water pressure and discharge rate were adjusted. This study also used open and closed basin positions, measured water outflow in a closed basin, and found an average Q of 81.08 liters per second.	Vortex height and reduced incision angle enhance efficiency; blade positioning is critical.	Optimizing vortex height and blade positioning improves low-head system performance.	Results apply mainly to small-scale systems; broader validation is needed for scalability.
Nasuki et al.	[17]	2021	Indonesia	experimental investigation	Water pressure and discharge rate were adjusted. This study also used open and closed basin positions, measured water outflow in a closed basin, and found an average Q of 81.08 liters per second.	Examined flow rates and pressure; closed basin averaged discharge of 81.08 l/s	Basin design significantly influences efficiency; the average discharge rate is observed.	The study lacks a detailed analysis of other basin configurations or conditions.
Kueh et al.	[18]	2014	Malaysia	Numerical Analysis	Two cases were studied, with two gaps of different sizes for water drainage. As the discharge gap expanded, the turbulent flow began to increase, causing more errors in the computational fluid dynamics model in predicting the formation of the water vortex.	Turbulence affects computational fluid dynamics accuracy; discharge gaps influence vortex formation.	Large discharge gaps lead to turbulence; vortex modeling needs refinement.	Prediction errors in computational fluid dynamics models; turbulent flow increases inaccuracies

Table 1. A concise synopsis of research papers on the topic of enhancing the efficiency of vortex water plants (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Nishi et al.	[19]	2020	Japan	experimental and computational study	Impact of flow rates and common losses on the performance of a gravitation vortex-type water turbine. Turbine efficiency increased with the flow rate, and the most common losses in the water turbine were tank and tank outlet losses, followed by friction loss inside the tank.	Flow rates and losses analyzed; higher rates improve efficiency but increase losses	Efficiency is linked to flow rates; tank and outlet losses are primary inefficiencies.	Losses not fully mitigated; further study needed for tank friction reduction
Powalla et al.	[20]	2021	Germany	Numerical and simulation study	A comprehensive comparison of the water level and flow velocities at the sites of important devices shows that the simulations agreed with the experimental results.	Developed digital twin for vortex simulation; accurate hydraulic property assessments	Accurate 3D modeling aligns simulations with experimental data.	Simulation outcomes require validation through physical experiments.
Faraji et al.	[21]	2022	Tanzania	Numerical and experimental investigated	These operations were simulated to calculate system efficiency. The numerical and experimental results were consistent; the Numerical study increased curved blade profile runner efficiency by 3.65%.	Blade shapes tested with curved blades showed a 3.65% efficiency improvement.	Blade geometry significantly impacts turbine efficiency	Experimental validation is needed to confirm numerical results
Yosry et al.	[22]	2023	Spain, Egypt	experimental and numerical study	A model of a vertical-axis turbine has been developed and fabricated to conduct experiments in an open channel to analyze a three-dimensional multiphase system.	Vertical-axis turbine developed; upstream speed increases power coefficient	Upstream speed strongly correlates with turbine performance	Field testing is required for reliability in real-world conditions
Pamuji et al.	[23]	2019	Indonesia	numerical study	The research demonstrates that the greatest velocity profile is near the discharge orifice. Furthermore, it indicates that an increase in the ratio of the inner diameter to the outer diameter (Din/Dout) or a reduction in the size of the output orifice leads to a heightened circulation of the vortex.	Conical basins improve vortex velocity near discharge; circulation increases.	Diameter ratios enhance vortex circulation and rotational force	Findings specific to experimental setups; general applicability uncertain
Herbhakti et al.	[24]	2021	Indonesia	Experimental and numerical study	The maximum strength, at 12.24 m2/s, was seen in the 30-centimeter exit slot. With a strength of 14.13 m2/s, the output slot measuring 35 cm was the strongest.	Tube diameter affects vortex strength; a 30 cm outlet achieved the highest strength.	Large outlet diameters increase rotational force and vortex strength	Limited to specific outlet sizes; environmental conditions not tested

Table 1. A concise synopsis of research papers on the topic of enhancing the efficiency of vortex water plants (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Ghani et al.	[25]	2019	Germany	experimental study	This study found that at 500 l/s and 0.7 m head, the water vortex power plant reached 43% efficiency, while at 700 l/s at 0.92 m head, it reached 38% efficiency without air vortices behind the turbine.	43% efficiency achieved at 500 l/s, 0.7 m head, no air vortices behind the turbine	Fixed flow-head combinations optimize turbine output efficiency	Findings lack insights into design optimizations for varying conditions
Shabara et al.	[26]	2015	Malaysia, USA	Experimental and Simulation study	This study used the commercial computational fluid dynamics tool ANSYS Fluent to mathematically model the free surface flow and find the optimal configuration for the vortex pool system.	ANSYS Fluent modelled vortex pool systems; validated computational fluid dynamics results with experiments	computational fluid dynamics models enhance design accuracy and are suitable for optimal pool configuration testing	Focused on specific configurations; broader basin designs need testing
Wanchat et al.	[27]	2013	Thailand	simulated and computational study	The tangential and radial velocity distribution determines the turbine blade for testing, and computational fluid dynamics simulates the vector flow field.	Simulated basin structures, flow rates and vortex heads influence velocity fields.	Basin optimization enhances flow dynamics and turbine compatibility	Simulation-based results need real-world validation across diverse environments

size of the output hole leads to an increase in the rotation of the vortex. Herbhakti et al. [24] examined the impact of vortex tube diameter and tube basin outflow slot on vortex strength. Results showed that vortex strengths varied at different diameters, with 25 cm having the highest strength at 7.53 m²/s. The 30 cm outlet slot had the highest strength at 12.24 m²/s. The 35 cm output slot had the highest strength at 14.13 m²/s. A 35 cm diameter tube had the highest vortex strength due to water flow-induced vortex intensity fluctuations. Ghani et al. [25] proved, according to this research, that The Water Vortex Power Plant achieved a maximum efficiency of 43% at a flow rate of 500 l/s and a head of 0.7 m, while at 700 l/s and a head of 0.92 m, it achieved 38% without air vortices under the turbine. This level reached the peak power output of 2.45 kW, comparable to the values offered by firms like Ecoligent, and was confirmed by Mühle et al. 2013 study. Shabara et al. [26] explained in this paper the efforts to improve the vortex pool and to convert the energy better. This study developed a free surface flow mathematical model using the ANSYS Fluent computational fluid dynamics commercial package. Also, a pilot testing platform was built to validate computational fluid dynamics results. The verification results show that the system can be accurately designed using ANSYS Fluent. Wanchat et al. [27] analyzed and developed a basin structure capable of producing a gravitational vortex. Current potentially harnessed as an alternative energy source. Various parameters, such as the gravitational vortex head and the flow rate, affect the velocity vector's flow field. The transverse and diagonal speed distribution determines which turbine blade is suitable for testing, and the vector flow field is simulated using computational fluid dynamics. The production of electrical energy is being studied by developing a model of a gravitational vortex power plant. The articles related to the impact on operational variables have been summarized in Table (1), and it can be seen from this Table that the highest operational variable is the vortex, which reaches its maximum height.

EFFECTS OF DESIGN VARIABLES

In this section, some design behaviors of the turbine and basin, such as blades, the dimensions of blades, the shape of the basin, the conveyor channel, and other factors, will be described.

Turbine

Using both an analytical and experimental approach, Ullah et al. [28] analyzed the performance parameters of a multistage gravitational water vortex turbine assembled in a cone-shaped trough, including rotational speed, torque, power, and efficiency, while subjecting it to various loads. Moreover, runners with inclined blades on a vertical plane are most suitable for planting near the bottom of the basin. The multiple gradations in gravitational water vortex turbines also demonstrate the evolution of the combined

rotation of a solid body and a free vortex. The turbine performance also improves significantly over the single-stage gravitational water vortex turbine. Candra et al. [29] conducted a study to determine the best runners for use in the laboratory's small-scale vortex turbines and compare the different sinkhole efficiencies of Kaplan, Francis, and Vortex turbines to determine the most effective design. Next, they used a statistical method from the collected data on drains designed to have five blades with a slope of 45° , a diameter of 12 mm, and a constant flow rate of 240 L/min. The energy input from the water (Pa), the energy output to the water (Pt), the torque (T), and the rotational speed (ω) are then measured. The information collected will result in proficiency scores for each runner, such as 21% for the Francis runner and 21% for the Vortex runner. A study was undertaken by Zuhri et al. [30] to investigate the impact of the distance between the L-type turbine and the cylindrical basin. Regarding power efficiency, the Model L turbine, positioned 5 cm away from the cylinder basin wall, demonstrates the highest effective power output of 2.89 W at a height of 10 cm. Conversely, the Model L turbine, positioned 10 cm away from the cylinder basin wall, exhibits a maximum effectual power output of 0.14 W at a height of 10 cm from the water outlet. Wardhana et al. [31] studied the number of turbine blades and their role in determining the turbine's efficiency. The results showed that increasing the number of blades reduces the efficiency of the turbine; 'in other words, six-bladed turbines are less efficient than three. Joshi et al. [32] conducted an experimental and computational examination of applying the Gorlov turbine, a pure reaction turbine, to harness electricity. The computational analysis was performed using ANSYS Fluent, while the experimental research involved using spiral turbine blades created using a 3D printer. Both computational and experimental investigations show that increasing the turbine aspect ratio while maintaining constant stiffness significantly improves efficiency. Subekti et al. [33] presented the numerical optimization and performance testing of the turbine driver profile for a vortex water turbine; MATLAB software was used to enhance the basic design of the turbine driver and determine the optimal inclination angle. The results were in a laboratory experiment where rotational speeds ranging from 150 to 650 rpm were used to assess turbine efficiency, torque, and power. Experimental results showed that the efficiency of the optimized turbine was 45.3%, representing an increase of approximately 14% over the efficiency of the first design turbine, which was 39.7%. Khan et al. [10] conducted a study to determine small hydropower station efficiency and ability to provide clean energy. An impulse turbine is the type of turbine used. The turbine rotational speed can be increased to improve efficiency by adjusting a blade angle and increasing the blade's surface area using baffle plates. For the curved blade shape, it was found that an efficiency of 82% could be achieved with a 5-blade design and a blade angle of 44° . Because of this, there has been a rise in total electricity generation. The

four-blade turbine achieved an efficiency of 79.95% from the geometric parameters used in the design of the system, and the result is shown in Table 2 and Figure 4.

Wichian et al. [12] presented a study to enhance the operational effectiveness of vortex turbines by installing barrier plates on the propellers. The study used computational fluid dynamics software to develop the barrier panels and determine their optimal application size and ratio. Experimental results indicated that five barrier plates at 50% of the propeller barrier area gave the maximum torque. In addition, experiments were conducted involving propellers without barrier plates and a turbine with barrier plates at 50% coverage. Results indicated that the turbine with a 50% barrier ratio showed an increase in torque of 10.25% total efficiency and 4.12% on average. Sharif et al. [34] studied five types of runners numerically. The top three competitors were selected for testing based on the amount of water pressure placed on the blades. The researchers examined how blade shape affects rotational speed, braking force, and mechanical efficiency. It was found that the curved circular chute blades worked at a rate of 48.02%, while the J-shaped chute blades and helical blades worked at a rate of 42.17% and 38.64%, respectively. Bajracharya et al. [35] provided an essential guide to runner design for a gravity water vortex turbine, as shown in Fig.(5). Seven

Table 2. The values of important parameters in the design of the system

Parameter	Value	Unit
Curved blade angle	44	degree
Blade number	5	none
Gate angle	15	degree
Notch angle	10	degree
Notch inlet width	0.15	m
Cone angle	67	degree
Channel height	0.18	m
Channel width	0.144	m

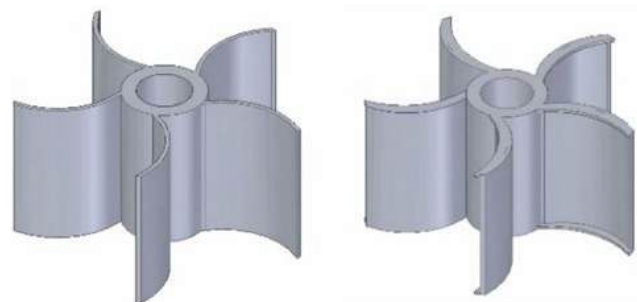


Figure 4. Two types of turbines with and without baffle plates.

different runner geometric parameters were relied upon for the runner design. Then, numerical and physical analyses were conducted to determine these parameters' impact on the system's efficiency. Then, the effect of these factors on both speeds and torques was studied. Test results show that runner height is the most important factor when creating a runner for a GWVHS. When tests were performed, it showed that the Gravity Water Vortex Turbine performed up to 47.85% better. Cheema et al. [36] conducted an efficiency analysis of a two-stage gravity water vortex turbine (GWVT) operating at varying flow rates and vortex heights. The two-stage GWVHS raises the performance characteristics with rising flow parameters. The highest hydraulic head drop close to the orifice also causes a high tangential velocity, aiding downforce production. Honnasiddaiah et al. [37] used a V-shaped rotary blade in their system. Based on the results of experimental and numerical analyses, it was determined that the blade format called V4 showed the best performance, yielding a maximum power factor of 0.22 and 0.21, respectively, when operated at a terminal speed rate of 0.87. The blade profile shows the maximum power factor exceeding the semi-circular blade profile by a margin of 19.3%.

Sritram et al. [38] compared the "Water Free Vortex Turbine" to the "Under Shot Water Wheel" to examine the efficiency of hydropower turbines at low water. In addition, the Small "Under Shot Water Wheel" produced greater torque than the Water "Free Vortex Turbine," while the latter made a faster cycle and more usable energy. At a

headwater height of less than one meter, it may be determined that the Water Free Vortex Turbine. It is preferable to the Small Under Shot Water Wheel as a small hydropower electricity generator. Dahal et al. [39] focused on the viability of booster-based GWVPP. This study investigates the potential for boosting power output by installing a second, smaller runner (booster runner) beneath the primary runner, which was subsequently empirically confirmed in a model constructed with four distinct booster runners. The studies demonstrated a boost of 3.84W in the compact model, equivalent to a 20.4% gain in efficiency from a total of 63.55% by the main runner alone. Adding the booster runner to the current setup can potentially increase power output. Dhakal et al. [40] focused on optimizing the runner to improve the efficiency of the Gravitational Water Vortex Power Plant. Runners with straight, twisted, and curved blade profiles are analyzed using computational fluid dynamics, as shown in Fig.(6). compared to the 46% efficiency of the straight blade runner and the 63% efficiency of the twisted blade version. Computational fluid dynamics analysis found that the curve blade design is the most efficient.

A model-free vortex power generation system was developed and tested by Rahman et al. [41] to determine what conditions would result in the most efficient operation of the power station. A few search results were: According to the data, a turbine with 3 blades and an External diameter of 0.027 m can attain a maximum efficiency of around 43% at a water pressure of 0.12 m. Maximum hydraulic

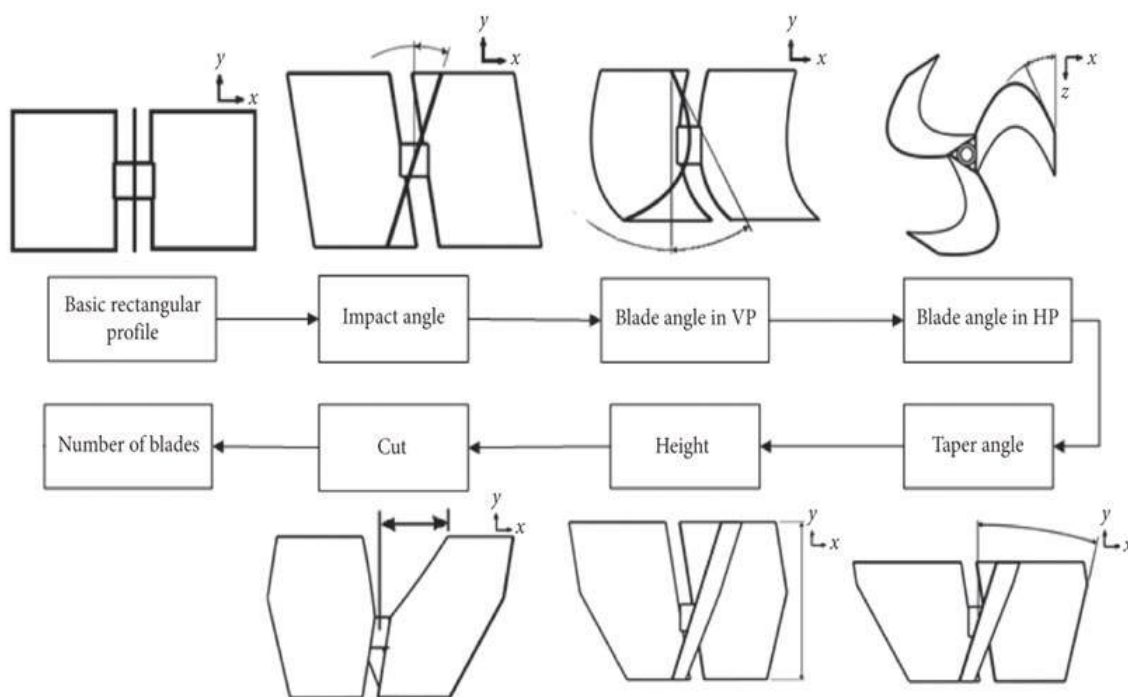


Figure 5. A geometrical feature has been added: a higher blade surface area and appropriate blade angles but maintaining a low runner weight.

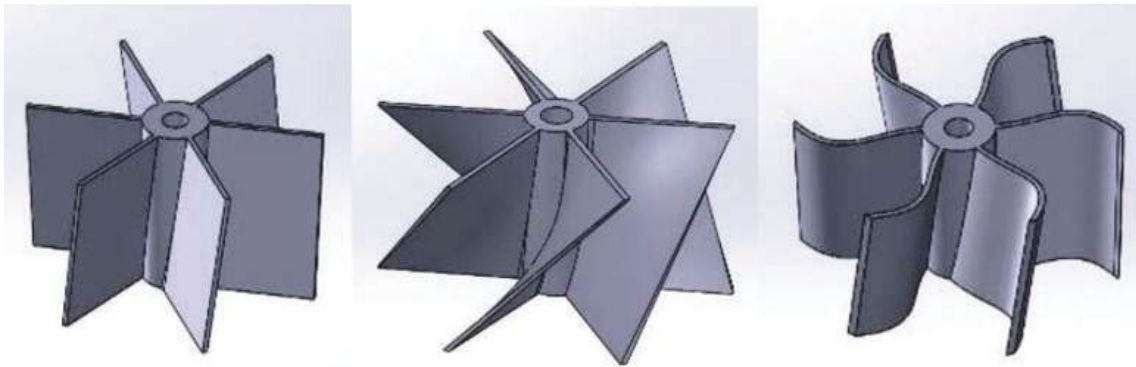


Figure 6. A three-dimensional computer-aided design (CAD) model with straight-ahead, turned, and curved blades.

efficiency was likewise seen in the model vortex power generating system when the rotating speed of the turbine was equal to or less than half the vortex tangential velocity. Hidayat et al. [42] modified the design and built a spiral hydro turbine to produce a spiral vortex to drive the turbine. The findings demonstrate that the spiral vortex hydro turbine may generate faster turbine rotation compared to a standard water intake. This study shows that a portable spiral vortex hydro turbine can spin at speeds of up to 90 revolutions per minute. This has led to the generator output voltage growing as the turbine rotation speed grows. Kueh et al. [43] aimed their research at studying the effects of operational speed and blade geometry on turbine efficiency. Two turbines equipped with flat blades and those that are curved are both evaluated and then compared. Both turbines exhibit equal rotational speeds under no load

conditions. The flat-blades turbine demonstrated a peak efficiency of 21.63% when operating at a rotational speed of 3.27 rad/s, while the curved-blades turbine achieved an efficiency of 22.24% at a rotational speed of 3.5 rad/s. When a load is applied during operation, the turbine blades utilize a backward-leaning curve to mitigate the disturbance on the water vortex, resulting in improved performance. As shown in Fig.(7), the speed triangle and resultant streamline of water vortex flow. Zariatin et al. [44] resulted from their study of constructing a vortex turbine power station of laboratory size. They examined three runners constructed of rust-resistant materials: SS-304, AA-5057, and PVC. Each of these runners produced 3.98 W of electric power, 3.47 W of electric power, and 3.3 W of electric power, respectively. You can use it to activate a 3-watt LED light. An efficiency of up to 31.8% is achieved compared to the 12.5 Watts of

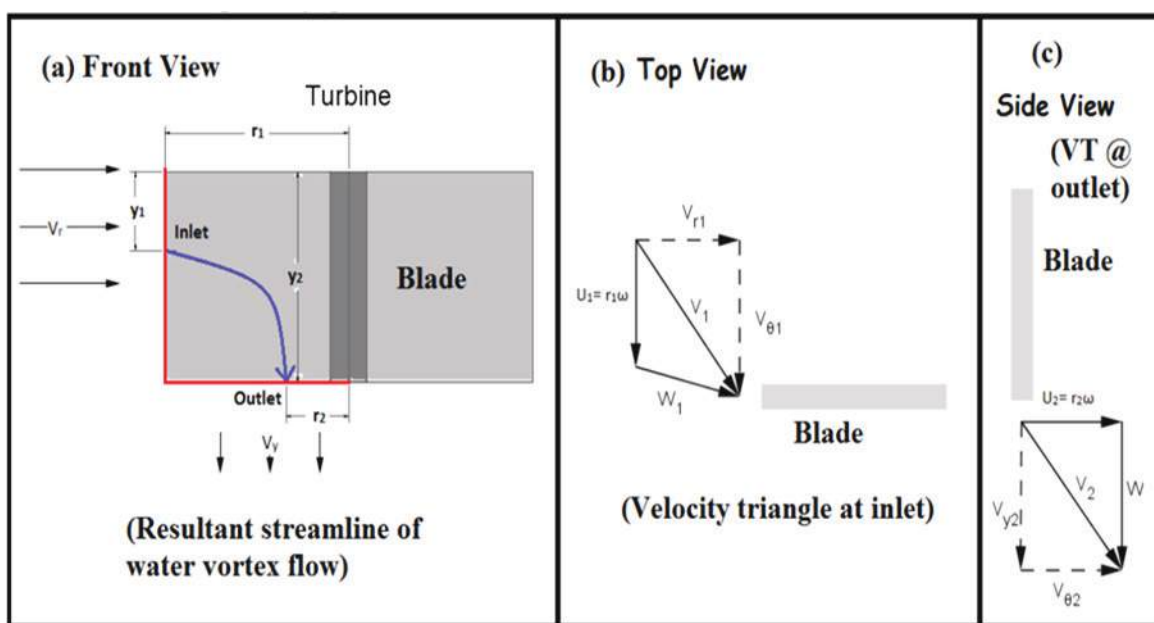


Figure 7. Water vortex turbine blade with a speed triangle.

potential hydraulic power computed. The runner's weight is an essential factor influencing the electric power produced. Assuming the water rotational potential is sufficient, a heavier runner will generate more power due to its more significant moment of inertia and, thus, greater torque.

Saleem et al. [45] studied several factors to determine their impact on performance parameters, including swirling elevation, runners' position, blade submersion percent, notch-angle, blade dimension ratio, blade curve, blade tilt, hub diam, and conical, straight blades. According to the experimental findings, the two most important factors determining Gravity Water Vortex Turbine performance are the height of the vortex and its shape, which should have a fully developed air core. Using zero-curvature tilted blades closer to the pool bottom, and the GWVHS can operate more efficiently with a minimal notch angle and hub diameter in the rotational speed range between minimum and maximum load conditions.

The study's overarching aim of Sritram et al. [46] was to study how steel and aluminium turbine materials affected power-generating efficiency in a waterless vortex hydropower station. Based on the data, the most efficient turbine materials were steel (33.56%) and aluminum (34.79%). However, at a high flow of water rate of 3.63 m³/min, the aluminum turbines exhibited greater torque values and power production efficiency, 8.4% and 8.14%, respectively. The research found that water turbines' higher torque and power-generating efficiency were attributable to their reduced weight. Venukumar et al. [47] studied

and discussed the natural whirlpool production of fluids, a turbulent circular movement of fluid layers generated for artificial vortex power generation. A vertical-axis turbine is built to imitate this motion with as little interference as possible. A strong gravity Water Vortex in the rotation tank harnesses the kinetic energy of moving water and directs it as rotational energy toward a turbine located at the vortex center. The turbine transforms this rotational energy with a generator into clean electricity. Yaakob et al. [48] provided an overview of small hydropower turbine systems by detailing their performance and identifying their many components. The first is reservoirs with water, river flow, pumped storage, in-stream, and innovative gravity swirl methods. Another way to categorize hydropower is by power size, which goes like this: large, small, micro, and pico. The study above can aid in finding the best turbine configuration for different microhydropower projects. Fig. (8) shows the rotor response with a split pipe. Mehmood et al. [49] aimed to develop and create a turbine that can generate electricity from canal systems by utilizing whirlpool and vortex principles. The turbine's vertical axis cross-flow makes it efficient, easy to install, and kind to marine life and the environment. It is a low-water head turbine with a submerged generator that can generate electricity from waterways. The turbine can operate continuously due to its durability and adaptability, reducing the need for large reservoirs and protecting marine life. The prototype's compact size makes it ideal for use in tight spaces like irrigation systems, where it can efficiently generate electricity.

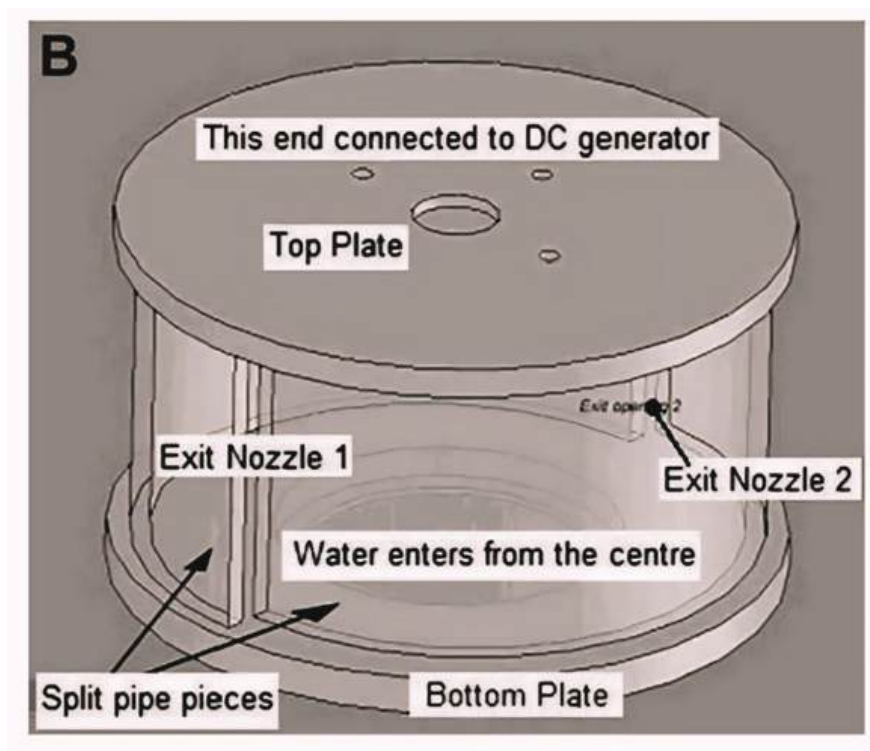


Figure 8. Rotor with a split-pipe response.

Khan et al. [49] focused on improving the efficiency of vortex turbines using a comprehensive design methodology. The project uses ANSYS simulation to predict and study the vortex formed in the main tank, followed by CAD modelling and governing equations. The primary objective is to achieve a watertight seal to protect the generator from harm. Chaulagain et al. [50] focused on Very low-head turbines with hydraulic heads between half a meter and three meters high. These turbines can be installed in single or parallel units and handle potable or treated water. A list of 38 hydro turbines and their operating ranges, compiled from previous research and interviews with 25 turbine manufacturers, is provided. The review offers an optional turbine classification scheme and explains the non-conventional hydropower. Vinayakumar B et al. [51] conducted this research using several modelling, simulation, and testing procedures. Then, details of Gravity Water Vortex simulation in the COMSOL FEA program and optimization settings in the study will be provided by considering various characteristics, including the height of the vortex chamber, blade length, blade angle, and number of blades. They were adjusted systematically to measure how much these factors affect the rotor speed. The article also details the construction of a real Gravity Water Vortex, an experimental device developed to validate and evaluate simulation

results. Wargetto et al. [52] conducted a numerical study to assess and compare the performance of the vortex using different shapes of blades, including oblique, straight, and curved, as shown in Fig.(9). After studying these various shapes, the researcher concluded that the best performance of the water vortex is using inclined blades, which have a hydraulic efficiency of 36%. A numerical study using the Ansys Fluent program was conducted by Warjito et al. [53] to find out the extent of the impact of the blade size and depth on the efficiency of the turbine. Nine configurations of different sizes were used: 200 mm, 350 mm, 500 mm, and turbine depths of 270 mm, 340 mm, and 410 mm. The study results showed that the larger size and top position gave the best performance of the turbine, with an efficiency of up to 40.22%.

Putra et al. [54] determined the output torque, rpm, voltage, current, and power generated by a gravity vortex power station and compared the effect of turbine position height on the data results obtained using a cylindrical basin, using L and S models with four turbine blades and using differences in height as shown in Fig. (10). The most significant electrical power using an L model turbine obtained an electrical power of 1.368 watts. Meanwhile, in the Model S turbine, the most considerable wattage was obtained at 2.097 W. An effort was made to compare and contrast

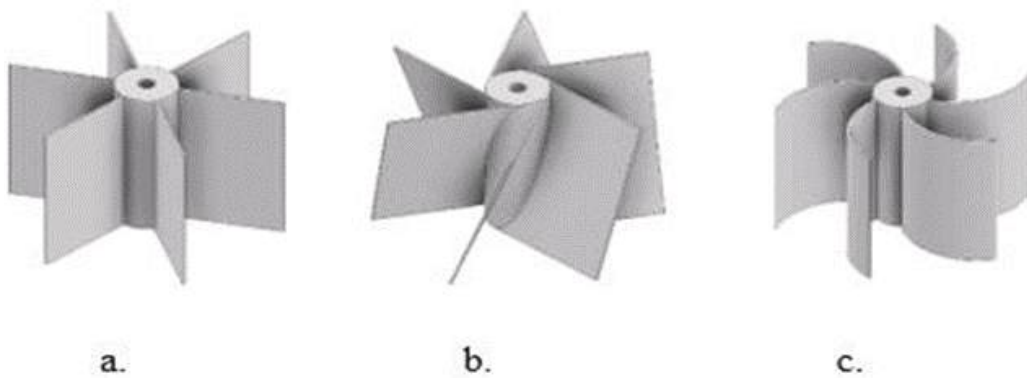


Figure 9. Design by wargetto et al. (a) Straight blade (b) Tilted blade and (c) Curved blade.

Table 3. Comparison between aluminum and steel in turbine design regarding efficiency, environmental impact, and durability

Factor	Aluminum	Steel
Efficiency	Higher efficiency of 34.79% due to lighter weight.	The efficiency is 33.56%, lower compared to aluminum.
Environmental impact	Highly recyclable and reduces environmental emissions.	Requires energy-intensive manufacturing, increasing emissions.
Weight and effect	A lighter weight reduces the required torque and increases rotation speed.	The heavier weight provides greater stability but adds more pressure.
Operational conditions	Ideal for systems with high water flow and light conditions.	More suitable for systems requiring high durability[46].

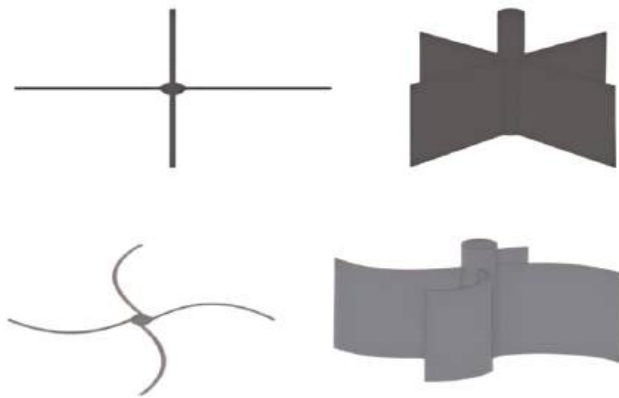


Figure 10. Two models of turbines (Model L and Model S).

several GWVHS microcosms in the study presented by Sedai et al. [55]. First, they analyze the success of different experimental investigations and formulations in Nepal. After that, several computational types of research are analyzed. A 1:20 scale model was then developed to be built in Johannesburg, South Africa, and performance predictions were based on comparing computational and experimental data. Kueh et al. [18] highlight the significance of water vortex behavior concerning the experiment setup. Two different sizes of slots were used to release water. The computational fluid dynamics model inaccurately predicts the creation of water vortices as the discharge slot enlarges due to the increased turbulent flow. The Computational Fluid Dynamics model enhances turbulence modelling.

Basin

According to Srihari et al. [56], vortex power is vital to maximizing the efficiency of the turbine, and this work is based on the intensification of the water vortex in the conical trough. The effect of the vortex condensate nozzles' orientation, alignment, and placement in five cone-shaped

basins was studied. The study findings show that compared to the swirling turbine and others, the proposed Group III configuration basin increases torque by 57.7%, power output by 54.4%, and efficiency by 54.4%. The comparison highlights the trade-offs between aluminum's efficiency and recyclability versus steel's durability and higher environmental impact, as summarized in Table 3. Wilson et al. [57] conducted the study using computational fluid dynamics within the ANSYS software to simulate micro-gravity vortex motion. This study investigates a structure that has the potential to generate a gravity-eddy current by utilizing flowing water with minimal variation in height, as shown in Figs. (11 & 12).

Chattha et al. [58] improved the performance of gravitational water vortex turbines and examined several design configurations using computational fluid dynamics technology that creates a powerful vortex capable of driving the runner. Apparent differences in the resulting vortices were observed when the basin entry velocity, height, outlet diameter, inlet channel width, depth, and flow rate were varied. As a result of our analysis, the optimized design parameters increase station efficiency and produce a powerful air-engraved vortex. According to the Gravitational Water Vortex Power Plant numerical and experimental research by Dhakal et al. [59], the conical basin is superior to the cylindrical basin in terms of output power and efficiency for all inlet and outlet conditions that are otherwise identical at 65–75% of the basin height, the optimal location for power extraction. It has become clear that future research into this system should focus on optimizing runner shape and studying alternative basin structures.

Dhakal et al. [60] focused on a form of micro hydro-power known as a GWVHS that generates electricity in remote regions with extremely low heads (between 0.73 and 3 meters). This research determines the geometrical requirements for the entrance and exit of a recently lauded efficient conical basin design, including the basin

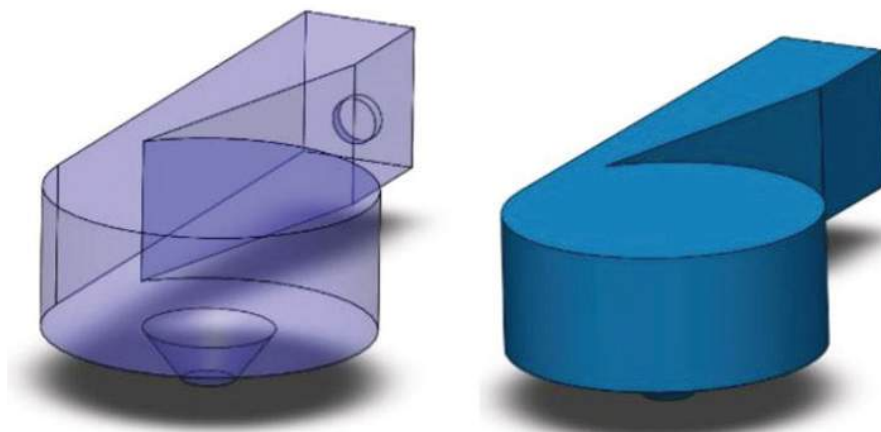


Figure 11. Design model developed by the ANSYS program and used in their research.



Figure 12. A water volume fraction from the front.

diameter and outlet diameter. by using Solidworks to develop our geometric CAD models and ANSYS CFX for our simulations. Finally, four different types of basins were designed and used during experimental tests, and these tests achieved a relationship between the engineering properties and the above parameters. Huwae et al. [61] discuss Vortex flow as a homogeneous, free-surface-flowing fluid consisting of water and air. The low-head little hydropower station is an ideal setting for this water vortex. The geometry of basins is the main focus of this research. The outlet diameter, depth discharge, and basin shape are modelled. It is determined that the flow rate and water head are the basin parameters that should be used to describe the water origin. It demonstrates that a spiral geometry basin can be utilized instead of a conical basin since it produces a vortex that is both symmetric and forceful. Huwae et al. [62] researched to prove that GWVHS requires less capital investment and lower operating and maintenance costs. Moreover, since basin opening and outlet diameter are the major criteria for defining basin geometry, inlet channels need to undergo additional Research. This study will fill in some of the blanks left by previous studies, opening the door to developing vortex turbines that will immediately impact renewable energy development in Indonesia via the GWVHS. This study by Rahman et al. [63] showed that a cone-shaped basin performs better than a cylindrical one. In addition, the inlet flow rate has significant effects on efficiency. Investigated Gupta et al. [64]. In this study, the gravitational generation of a water vortex flow is a novel approach in hydropower engineering. The work uses simulation techniques to accurately determine the details of water vortex formation. Computational fluid dynamics models incorporate the appropriate boundary conditions based on the experimental setup. The experiment tested two distinct aperture diameters for water outflow in two settings. The initial condition demonstrated that the vortex heights in the experiment and the computational fluid

dynamics model were in accordance. However, in the second condition, the final vortex height varied due to a more turbulent flow. Dhakal et al. [65] aim of their study was to improve the design of a conical basin in a water-gravity vortex power plant. This is achieved by modifying four design parameters: the notch angle, canal height, notch inlet width, and cone angle. Various geometric models were created using SolidWorks software, and ANSYS Fluent was used for simulation.

Turbine and Basin

Pandey et al. [66] took up the design and configuration of the turbine that achieved maximum efficiency and chose a suitable cylindrical trough to implement this turbine in the GWVHS, as shown in Fig.(13 and 14). In addition, optimizations were made to blade angle, discharge orifice diameter, runner position, and rpm using computational fluid dynamics (CFD)-based optimization techniques. After research into computational fluid dynamics, it was determined that the most appropriate blade angle for the system was 43° . In addition, the system demonstrated an optimal efficiency of 23.639% when operating at a rotational speed of 40 rpm.

Kim et al. [67] looked into how well the GWVHS worked, focusing on the number of blades in the vortex turbine and the draft tube that was added to change the shape of the conical vortex basin, as shown in Figure 15 and 16. The study of Dhakal et al. [68] included two stages. The first stage compares the two turbines' design, production, and performance, and the second stage includes designing and building the conical and cylindrical tanks and comparing the system performance during use. Reducing the number of blades led to an increase in turbine efficiency. Utomo et al. [69] investigated the torque, water discharge, water force, turbine power, and electric power generated by the hydrographic power plant instrument. The results show the highest results with an output of 5 cm, which is 3.46 W

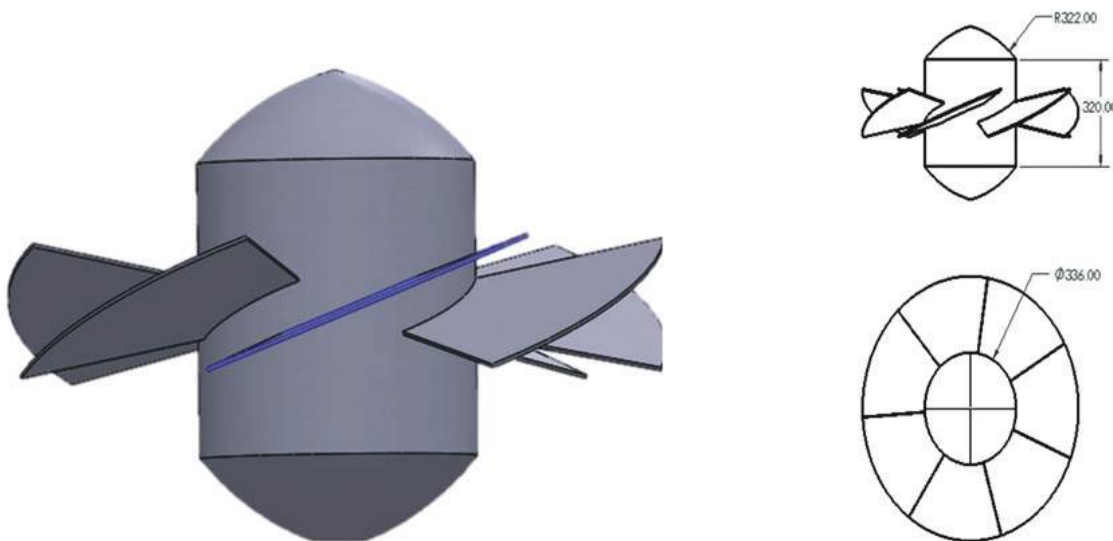


Figure 13. Several views of this runner CAD design.

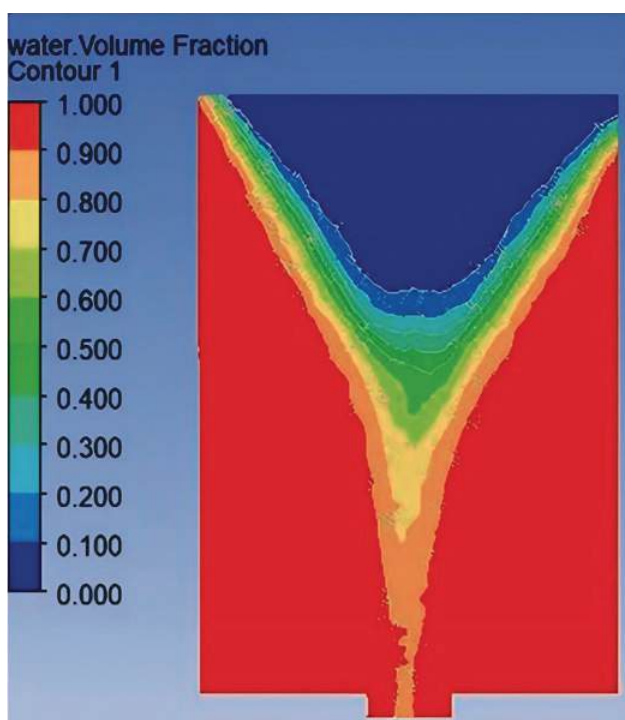


Figure 14. The water volume fraction and formation of the air core of this basin having optimized outlet diameter.

of hydroelectric power at a turbine height of 22 cm, while the production of 6 cm is 2.51 W at 28 cm. Acharya et al. [70] improved the system by combining the results between analytical thinking and numerical accuracy. The system's effectiveness can be changed by fiddling with a number of these variables. However, no advanced calculations have yet been done. So, they are working to improve the station

runner so that the Gravitational Water Vortex can generate more power. Then, they analyze the effects of the inlet jet angle and the radius of curvature of the turbine blade on torque, power, and efficiency. Analytical calculation establishes these two values, which are then verified. Maximum torque for an impact jet angle of 18 degrees and a radius of curvature of 285 millimetres differed by 2.37% between analytical and numerical values. Khan et al. [71] analyze the two-stage flow to generate the vortex at different Basin parameters; this work aims to investigate the characteristics that influence computational fluid dynamics, leading to selecting an appropriate basin configuration. After that, the basin was utilized to analyze different blade forms under varied load scenarios. How the vortex height affected the gravitational water vortex turbine efficiency was investigated. Cross-flow blades have demonstrated the highest efficiency for the same discharge and head circumstances among the four types of blades employed in the study.

Tamiri et al. [72] examined different basin configurations, orifice radii, blade layouts, basin shapes, and diffusers. So, the diffuser channels the water into a basin, where its vortex can move in a tangential direction and speed up the flow to the turbine. Simulation results demonstrate that adding a diffuser dramatically increases vortices' tangential velocity and kinetic energy, raising the vortex height, increasing its strength, and influencing its uniformity. Song et al. [73] presented a new system for capturing energy in a fluid vortex that can take advantage of the kinetic energy of flowing water. The system incorporates an impeller, basin, induction pine, and power-capturing devices. SolidWorks, ANSYS ICEM, and ANSYS CFX were used to validate the design. After developing and testing a smaller-scale system, a system that can harvest 150 kilowatts of power was built. A preliminary study by Ullah et al. [74] on multistage of Gravity Water Vortex conical bowl staging has tested

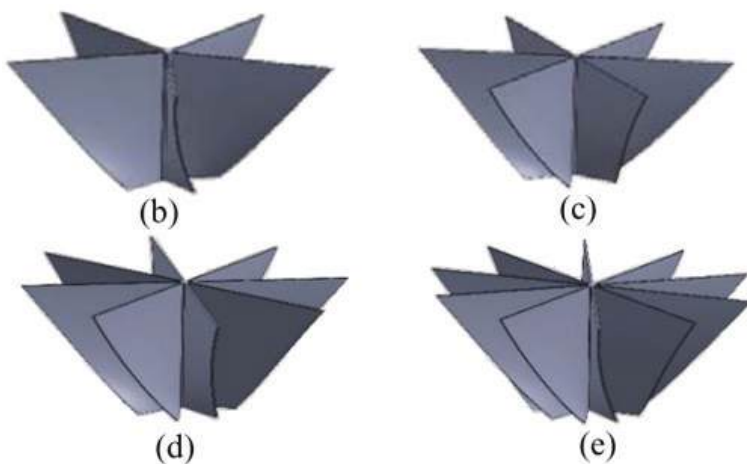


Figure 15. The number of turbine blades [67].

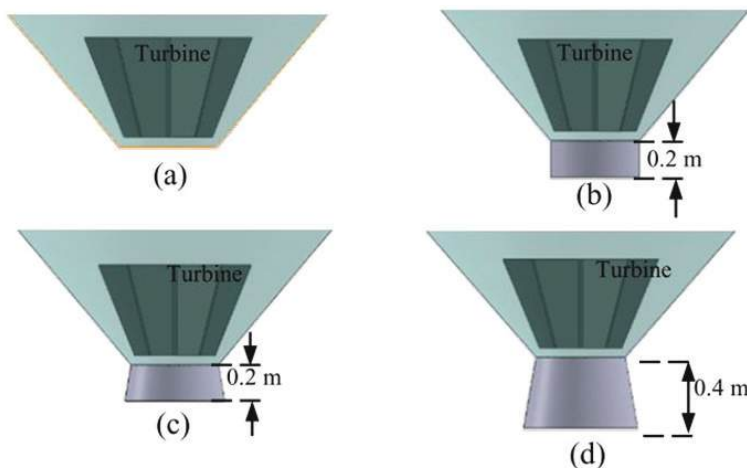


Figure 16. The draft tube straight and cone.

runners of Savonius-type blades. The study reveals that the performance of a Gravity Water Vortex changes depending on key design factors such as the ratio of rotor diameter to basin diameter, the interaction between vortex and blades, stage location, and offset distance between neighbouring stages. The study found that vortex flow can transfer energy across adjacent stages without physical interaction between rotors. Multistage Gravity Water Vortex delivers greater strength than single-stage Gravity Water Vortex by intensifying the swirling in proximity. This research could serve as a standard for future Gravity Water Vortex innovations.

Maika et al. [75] evaluated the development and progress of the GWVHS system on a large scale, incorporating a range of basin types and inlets and the output of canals and turbines. It also discusses vortex dynamics and how to evaluate efficiency. The paper also examines turbulence and multiphase models from top-tier numerical simulation studies. The paper presents the GWVHS system in

(Papua New Guinea) as a case study, summarizing the main concerns and challenges and suggesting areas for future research on the system's performance. Zarate-Orrego et al. [76] studied this research: a runner, a cylindrical vortex chamber, and a semi-converging nozzle. Despite challenging testing conditions, the device demonstrated electricity production. To produce power from water levels heads between 0.7 and 3 meters. The research on the effects of design variables has been summarized in Table 4, and it can be seen from this table that the highest design effects are increasing the blade angle to 44 degrees and increasing the surface area using barrier panels and the number of blades to 5. The key design and operational factors and their direct effect on the performance of the GWVHS are summarized in Table 5. All of these significantly contribute to the optimization of the system in terms of efficiency and performance, as well as in its control, energy output, stability, etc., depending on its operational state.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Ullah et al.	[28]	2020	Pakistan, South Korea	Investigated analytically and experimentally	Analyzed the performance parameters of a multistage gravity water vortex turbine (Gravity Water Vortex Turbine).	Enhanced turbine performance by 15% through multistage design optimizations.	Shift towards multistage turbine systems to exploit efficiency at varying flow rates.	Complexity and higher costs associated with multistage systems.
Candra et al.	[29]	2023	Indonesia	A comprehensive review	Utilized a statistical technique on drains with 5 blades at 45 degrees, 12 mm diameter, and 240 L/min constant flow rate. Water energy input (Pa) and output (Pt), torque (T), and rotational speed (ω) are monitored.	Kaplan turbines demonstrated 25% better efficiency under optimized conditions.	The increasing adoption of Kaplan turbines in urban hydro projects is due to their efficiency and adaptability.	High cost and specialized maintenance are required for Kaplan turbines.
Zuhri et al.	[30]	2020	Indonesia	Experimental study	To investigate the impact of the distance between the L-type turbine and the cylindrical basin. The maximum Power efficiency for the Model L turbine is 5 cm from the cylinder basin wall, 2.89 W.	Significant efficiency losses were noted due to improper site selection and setup.	Emphasis on the importance of site analysis for optimal hydro power plant performance.	Site selection constraints can limit viable locations for installations.
Wardhana et al.	[31]	2019	Indonesia	Experimental study	The blade with the twisted shape has the highest efficiency, and the growing Flow velocity determines turbine efficiency; the best turbine is T-3-490, with 25.71%, 35.58%, and 54.40% efficiency at each speed.	It was shown that reducing the blade number can increase efficiency by up to 5%.	Research focuses on optimizing blade numbers to balance efficiency and cost.	Fewer blades may decrease system stability.
Joshi et al.	[32]	2021	Nepal	Computational and experimental study	Increasing the turbine dimensions ratio while keeping stiffness constant. Significantly improves efficiency.	Enhanced turbine efficiency by 10% through optimized turbine dimensions.	Increased reliance on computational fluid dynamics to model and predict turbine performance.	High dependency on precise simulations and models, increasing the risk of errors.
Subekti et al.	[33]	2023	Indonesia	Numerical and simulation study	Numerical and simulation optimisation and performance testing to determine the optimal inclination angle. The optimised flow has 45.3% efficiency, up 14% from the baseline design of 39.7%.	Improved efficiency by 8% through precise blade angle adjustments.	Innovations in turbine blade design focusing on adjustable and dynamic systems.	Precise blade angles are crucial, necessitating sophisticated control systems.
Khan et al.	[10]	2021	Bangladesh	A comprehensive review	To Increase turbine efficiency and rotational speed by increasing the blade angle and surface area using barrier plates. A 5-blade design with a 44° blade angle yielded 82% efficiency for the curved blade form.	Efficiency gains are achieved by adjusting blade angles to optimize performance.	Trend towards adjustable blade systems in hydro turbines for enhanced adaptability.	Ongoing adjustments and maintenance increase operational costs.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Wichian et al.	[12]	2016	Thailand	Simulation analysis	To improve the operational efficacy of vortex turbines Using barrier plates attached to the propellers. Fans with a 50% barrier ratio increased their efficiency by 4.12%. They demonstrated a torque gain of 10.25%.	Installing barrier plates increased torque and reduced turbulence.	Barrier plates are becoming commonplace to stabilize water flow and increase efficiency.	Barrier plates add complexity and potential for debris buildup.
Sharif et al.	[34]	2023	Pakistan	Numerical and investigative study	Performed an exhaustive numerical analysis on five distinct varieties of runners. A rate of 48.02% was determined for the curved circular crooked blades, 42.17% for the J-shaped crooked blades, and 38.64% for the spiral blades.	Curved blades improved system efficiency by optimizing water flow dynamics.	Curved blade designs are increasingly favoured for their effectiveness in capturing kinetic energy.	Specialized manufacturing processes for curved blades raise project costs.
Bajracharya et al.	[35]	2020	Nepal	Numerically and experimentally study	The runner design was based on seven distinct geometric factors. The next step was identifying how these variables affected the corresponding speeds and torques.	The critical role of runner geometry is highlighted, achieving up to 12% efficiency gains.	Geometric optimization is increasingly recognized for improving turbine performance.	Specific geometric designs may not be applicable universally.
Cheema et al.	[36]	2019	Pakistan	Experimental study	They tested a two-stage Gravity Water Vortex Turbine with different flow rates and vortex heights to see its efficiency. In a two-stage Gravity Water Vortex Turbine system, the performance characteristics are enhanced as the flow parameters increase.	Two-stage turbines showed over 20% efficiency improvement in pilot tests.	Two-stage turbines are gaining popularity for their ability to maximize energy capture.	Complexity and cost of two-stage systems increase potential for mechanical failures.
C M et al.	[37]	2021	India	Experimental investigation study	Experimental and computational evaluations showed that the blade shape V4 performed best, with a maximum power factor of 0.22 and 0.21 at a tip speed ratio of 0.87. The blade profile has a 19.3% higher maximum power factor than the semi-circular blade profile.	V-shaped blades produced a higher power factor under optimal conditions.	Adoption of V-shaped blades in areas with variable water flow to improve performance.	V-shaped blades require precise engineering and are costly to produce.
Sritram et al.	[38]	2017	Thailand	Experimentally and Analytically study	The small under shot water wheel produced higher torque, while the water free vortex turbine produced more energy and a faster cycle.	Demonstrated effectiveness of low-water turbines for rural applications.	Research into low-water turbines is expanding, aiming to make renewable energy more accessible.	Limited energy output makes them suitable only for small-scale applications.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Dahal et al.	[39]	2019	Nepal	Simulation study	investigates the potential for boosting power output by installing a second, smaller runner (booster runner) beneath the primary runner. The studies demonstrated a boost of 3.84W in the compact model, equivalent to a 20.4% gain in efficiency from a total of 63.55% by the main runner alone.	Secondary boosters increased power output by 20% without significant redesign.	Utilization of booster systems to enhance existing turbine setups is becoming more common.	Adding boosters complicates system dynamics and maintenance.
Dhakal et al.	[40]	2017	Nepal	Experimental and simulation	Computational fluid dynamics studies showed that the curved blade profile was most efficient, at 63% versus 46% for the straight blade runner. With a 0.5 m head, the runner functioned as expected by the CFD study at 71% efficiency.	Runner design optimizations led to a 15% increase in efficiency.	Continuous improvement and testing of runner designs are crucial for advancing hydro turbine technology.	Frequent redesigns and updates may delay implementation and increase costs.
Rahman et al.	[41]	2016	Malaysia	Experimental and simulation	Tested a model-free vortex power generation system to assess the power stations optimal operating. A few search results were: The data shows that a turbine with 3 blades and an Exterior diameter of 0.027 m may achieve 43% efficiency at 0.12 m water pressure.	Optimized operational conditions to enhance turbine efficiency significantly.	Focusing on operational optimizations to maximize energy outputs in varying environmental conditions.	Such optimizations require complex control systems that can be challenging to manage and maintain.
Hidayat et al.	[42]	2020	Indonesia	Design and analysis	The modification is made by designing and building a spiral vortex hydro turbine to produce a spiral vortex to drive a turbine. The findings demonstrate that the spiral vortex hydro turbine may generate faster turbine rotation compared to a standard water intake.	Demonstrated that spiral vortex structures could increase turbine rotational speed and efficiency.	Exploring the benefits of vortex dynamics in improving the mechanical energy conversion processes.	Implementing such designs often involves higher costs and more sophisticated manufacturing techniques.
Kueh et al.	[43]	2017	Malaysia	Experimental study	The aim is to determine how the turbine performs when the operating speed and blade shape are changed. The improvements are examined using the velocity triangle and the Euler Turbomachinery Equation. The flat-blades and curved-blades turbines reached their maximum efficiencies at different rotational speeds; the former at 3.27 rad/s and the latter at 3.56 rad/s achieve 21.63% and 22.24% efficiency.	Investigated the impact of blade geometry on turbine efficiency, highlighting significant improvements.	Emphasis on precision engineering of turbine blades to enhance performance and reduce wear.	Precise blade designs can escalate production costs and require more rigorous maintenance protocols.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Zariatin et al.	[44]	2020	Indonesia	Experimental study	By evaluating three runners: SS-304, AA-5057, and PVC. These runners produced 3.98 W, 3.47 W, and 3.3 W of electricity. The runner weight, due to its larger moment of inertia, affects more electric power production.	Developed a model that shows how material choices affect electricity production and turbine longevity.	Growing interest in material science to extend the durability and efficiency of hydro turbines.	Advanced materials can be expensive and might require new manufacturing processes.
Saleem et al.	[45]	2020	Pakistan, South Korea	Experimental and numerical study	By investigating the PPs of single stage Gravity Water Vortex Turbine within different flow and design scenarios. The results showed that the evolution of the generation structure and the height of the vortex are the most important factors in determining the performance of Gravity Water Vortex Turbine.	Created an economic model that aids in the planning and design of more sustainable turbine systems.	Increasing use of economic modeling to support the development and deployment of cost-effective turbine solutions.	Economic models often rely on assumptions that may not hold true in all operational environments.
Sritram et al.	[46]	2015	Thailand	Experimental study	By studying how steel and aluminum turbine materials affected waterless vortex hydropower station power-generating efficiency. Data showed that steel (33.56%) and aluminum (34.79%) were the most efficient turbine materials. Reduced weight gave water turbines improved torque and power-generating efficiency, according to a study.	Improved efficiency of low-water turbine for rural renewable	Advancing low-head turbine technology for use in shallow water setting	Limited energy output restricts use to smaller-scale applications; efficiency varies with water level change
Venukumar et al.	[47]	2013		Numerical study	Discussed natural whirlpool creation of fluids for artificial vortex (ArVo) power generation. Strong gravity Water Vortex in the rotation tank harnesses kinetic energy.	Showcased how adjustments to turbine nozzles can lead to a 6% increase in overall efficiency.	Innovating within turbine nozzle designs to achieve better control over water flow and energy capture.	Adjustments to nozzle designs may involve trade-offs with system simplicity and reliability.
Yaakob et al.	[48]	2014	Malaysia	A comprehensive review	Hydropower stations are categorized by electric capacity. Hydropower can be huge, tiny, or pico, depending on power size. And concentrating on gravitational vortex micro hydropower increases water quality and sustainability.	Focused on the impact of environmental factors on turbine efficiency, proposing adaptive designs.	Adaptive turbine technologies are becoming crucial for dealing with environmental variability.	Such technologies can increase the overall complexity and cost of hydroelectric systems.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Mehmood et al.	[77]	2019	Pakistan	Design and simulation study	Because of its endurance and versatility, the turbine can run constantly, saving marine life and reservoir space. It generates electricity efficiently in tight locations such as irrigation systems due to the prototype tiny size.	Their research highlighted the effectiveness of using computational fluid dynamics (Computational Fluid Dynamics) to optimize turbine performance.	The use of Computational Fluid Dynamics in designing turbines is gaining traction for its ability to predict and enhance flow dynamics.	High computational costs and the need for expert knowledge can limit the widespread adoption of Computational Fluid Dynamics .
Khan et al.	[49]	2023	Pakistan	Experimental and simulation	Utilizes a thorough design process to enhance the efficiency of vortex turbines. Mechanical losses due to vibrations can be mitigated with a smaller column, which is another objective of the research.	Introduced a new turbine blade design that improved efficiency by reducing drag and increasing lift.	Development of blade technologies that maximize energy extraction from water flow.	New blade designs require extensive testing and validation which can delay commercialization.
Chaulagain et al.	[50]	2023	Nepal	A comprehensive review	Concentrates on very low-head turbines with hydraulic heads that are half a meter to three meters high. The review gives an alternative scheme for classifying turbines and describes non-conventional hydropower.	Studied the long-term impacts of turbine installations on local ecosystems, proposing sustainable practices.	The environmental impact of hydro installations is being more carefully considered in new designs.	Implementing environmentally friendly designs often involves higher initial costs and complex regulatory approval.
Vinayakumar B et al	[51]	2022	India	Experimental and Simulation study	A small, portable power stations that utilizes gravitational water vortex technology is detailed, along with the processes of modeling, simulation, and testing. The height of the vortex room, the length of the blades, the angle of the blades, and the total number of blades were among the many factors considered. A systematic approach was used to alter each of these parameters individually in order to measure their impact on the rotor speed.	Enhanced turbine output by integrating smart sensors to monitor and adjust performance in real-time.	Smart technology integration into turbine systems is on the rise, aiming to automate and optimize operations.	Dependence on technology increases vulnerability to cyber threats and system failures.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Wargetto et al.	[52]	2023	Indonesia	Numerical study	conducted a numerical study to evaluate and compare the performance of the vortex using different shapes of blades, including oblique, straight and curved, as shown in Figure 12. After studying these different shapes, the researcher came to the conclusion that the best performance of the water vortex is using inclined blades, which have a hydraulic efficiency of 36%.	Developed a low-cost turbine model suitable for small-scale applications, making renewable energy more accessible.	Focus on downsizing technology to make it feasible for use in remote and rural areas.	Smaller turbines may not be viable for larger scale applications, limiting their market scope.
Wärjito et al.	[53]	2022	Indonesia	Numerical study	conducted a numerical study using the Ansys Fluent program to find out the extent of the impact of the blade size and depth on the efficiency of the turbine through the use of nine configurations of different sizes 200 mm, 350 mm, 500 mm and turbine depth of 270 mm, 340 mm, 410 mm, where the result of the study gives the larger size and top position the best performance of the turbine with an efficiency of up to 40.22%.	Their research showed how variable speed turbines can adapt to changes in water flow, enhancing efficiency.	Variable speed technologies are being explored to maximize efficiency across diverse water conditions.	Such technologies can complicate turbine design and increase the overall system cost.
Putra et al.	[54]	2020	Indonesia	Experimental study	The influence of turbine position height on cylindrical Basin. By Using... L and S models with four turbine blades and height variances place the turbine, varying fluid flow. The largest electrical power L-model turbine power is 1.368 watts. The greatest Model S turbine wattage is 2.097. Most data is generated at the turbine 28 cm height.	Examined the lifespan of turbines under different operational stresses, recommending design alterations.	Longevity and durability are becoming key factors in turbine design to reduce maintenance and replacement costs.	Improving durability often requires new materials or changes in design that can increase costs.
Sedai et al.	[55]	2020	Nepal, USA	A comprehensive review	Computational and experimental data were used to construct a 1:20 scale model designed to be built in Johannesburg, South Africa. This model was then utilized to estimate performance.	Demonstrated a new method for harnessing kinetic energy from slow-moving water, increasing efficiency.	Expanding the potential applications of hydro turbines in low-energy water streams.	Technologies effective in low-energy streams may not scale up efficiently for larger operations.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Srihari et al.	[56]	2019	India	Experimentally study	The study found that the set III configuration basin enhances torque by 57.77%, power production by 54.42%, and efficiency by 54.41% over the vortex turbine.	Showcased improvements in turbine efficiency through aerodynamic optimizations of blade shapes.	Aerodynamic improvements are crucial for the next generation of hydro turbines.	Aerodynamic optimizations require high-fidelity modeling and extensive experimental validations.
Wilson et al.	[57]	2019	Ecuador	Analyzed and simulation study	By investigating a structure that has the potential to generate a gravity-eddy current by utilizing flowing water with minimal variation in height.	Their study focused on the socio-economic impacts of turbine installations, proposing community engagement strategies.	There is an increasing focus on the socio-economic benefits of renewable energy projects.	Community-based projects can face bureaucratic and logistical challenges that delay implementation.
Chattha et al.	[58]	2017	Pakistan	Simulation study	To enhance the performance of gravity water vortex turbines, several design configurations using computational fluid dynamics technology that create a powerful vortex capable of driving the runner. As a result of our analysis, the optimized design parameters increase stations efficiency and produce a powerful air-engraved vortex.	Investigated noise reduction techniques in turbine operations to enhance environmental compatibility.	Reducing turbine noise is becoming important for installations near human settlements.	Noise reduction can compromise turbine efficiency and increase operational costs.
Dhakal et al.	[59]	2015	Nepal	The numerical and experimental study	For all other inlet and outlet conditions, the conical basin outperforms the cylindrical one in terms of output power and efficiency, as shown by the numerical and experimental research conducted by the Gravitational Water Vortex Power Plant.	Developed a model to predict turbine performance under variable water conditions, enhancing predictive maintenance strategies.	Increasing use of predictive modeling to ensure optimal turbine operation and maintenance.	Reliance on accurate data input for models, which can be challenging to obtain in less controlled environments.
Dhakal et al.	[60]	2020	Nepal	Experimental and simulation	This study investigates the inlet and outflow geometrical characteristics, specifically basin diameter and outlet diameter, of a recently recognized efficient conical basin design.	Introduced innovations in turbine blade materials that reduce wear and enhance efficiency.	Material science advancements are key to improving the lifespan and performance of turbines.	High costs associated with new material research and implementation.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Huwae et al.	[61]	2020	Indonesia	Simulation study	Flow rate and water head are the basin measures used to describe water origin. It shows that a spiral geometry basin can be used instead of a conical basin because it creates a symmetric and powerful vortex.	Showed that integrating energy storage systems with hydro turbines can smooth out energy production fluctuations.	The integration of energy storage solutions is becoming more common in stabilizing the hydroelectric power supply.	Energy storage systems add significant costs and complexity to hydro turbine installations.
Huwae et al.	[62]	2020	Indonesia	Simulation study	The research was conducted to provide evidence that the GWVP has less capital investment and low operational and maintenance costs.	Explored the effects of water salinity on turbine materials, suggesting ways to mitigate corrosion.	Addressing environmental challenges such as salinity to enhance turbine durability in harsh conditions.	Corrosion-resistant materials and treatments can be expensive and require regular maintenance.
Rahman et al.	[63]	2017	Malaysia	A comprehensive review	showed that a cone-shaped basin performs better than a cylindrical one. In addition, the inlet flow rate has significant effects on efficiency.	Implemented advanced flow measurement techniques to improve the accuracy of water flow data used in turbine control systems.	Enhanced flow measurement technologies are critical for optimizing turbine performance.	Advanced measurement devices can be cost-prohibitive and require specialized training.
Gupta et al.	[64]	2021	Uttar Pradesh	Simulation study	Examined two aperture diameters for water outflow in two circumstances. The initial condition showed that the experiment and Computational Fluid Dynamics model had similar vortex heights.	Improved turbine control algorithms to optimize power output in real-time based on water flow variability.	Developing smarter, more adaptive control systems for hydro turbines to increase efficiency.	Complexity in algorithm development and potential issues with system integration.
Dhakal et al.	[65]	2010	Nepal	Simulation and numerical	Optimize the Gravitational water vortex power station conical basin design. A mathematical association was found between design parameters and water velocity.	Focused on the environmental impact assessments of turbine installations, proposing strategies to minimize ecological disruption.	Growing emphasis on sustainable practices in hydro turbine installation and operation.	Environmental protection measures can restrict turbine design and placement options.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Pandey et al.	[66]		Nepal	Simulation analysis	Data from computational fluid dynamic showed that 43 degrees was the best blade angle for the system. Additionally, at 40 rpm, the system had 23.639% efficiency.	Studied the scalability of micro-turbine systems for use in decentralized energy generation.	Micro-turbines are gaining attention for their potential in small-scale, local energy solutions.	Scaling down turbine technology poses challenges in terms of efficiency and cost-effectiveness.
Kim et al.	[67]	2021	Korea	Simulation study	Considering the vortex turbine blade count and the Intake pipe inserted to alter the conical vortex basin shape.	Developed a new turbine blade coating that reduces biofouling and improves efficiency in marine environments.	Innovations in protective coatings to extend turbine life and maintain efficiency in diverse environments.	Coating technologies can be difficult to apply and may require frequent reapplication.
Dhakal et al.	[68]	2014	Nepal	Experimental study	The efficiency of turbines was enhanced by cutting down on the number of blades. Among the two types of basins, the conical one produces more powerful vortices than the cylindrical one.	Investigated the long-term performance of turbines under sediment-rich water conditions, offering design modifications.	Enhancing turbine designs to cope with high sediment loads and prevent wear and tear.	Modifications for sediment handling can reduce overall turbine efficiency.
Utomo et al.	[69]	2020	Indonesia	Experimental study	The torque, water discharge, force, and turbine power generated by the Gravitational Water Vortex Power Plant tool were investigated. The maximum torque of the cylinder basin flow is 5 cm outlet, 0.00012 Nm, and the turbine power is 0.0029 W. The largest torque of 6 cm output is 0.000084Nm.	Examined the integration of small hydro installations with local grid systems, highlighting benefits and challenges.	Small-scale hydro projects are increasingly seen as viable supplements to local energy grids.	Grid integration involves regulatory and technical challenges that can be daunting for smaller projects.
Acharya et al.	[70]	2019	Nepal	Analytical and numerical study	Changing a few of these variables can alter the system's efficacy. Therefore, the station's runner is aimed to be enhanced to increase the power generation capacity of the Gravitational Water Vortex. The effects of the inlet jet angle and the turbine blade radius of curvature on efficiency, power, and torque are studied.	Developed guidelines for the ecological monitoring of hydro sites to ensure compliance with environmental regulations.	Stronger focus on ecological monitoring to balance energy production with conservation efforts.	Monitoring and compliance can be resource-intensive and may impact project feasibility.

Table 4. Overview of the findings from earlier papers concerning the dimensions and design of water vortex power plants to enhance their performance (continued)

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Khan et al.	[71]	2018	Pakistan	Experimental and numerical study	Different blade shapes were tested under various loading conditions using the basin. The effectiveness of the gravitational water vortex turbine was studied with the vortex height.	Enhanced turbine blade design to reduce cavitation and improve operational stability.	Addressing cavitation issues is crucial for maintaining turbine efficiency and longevity.	Anti-cavitation designs can complicate the turbine manufacturing process and increase costs.
Tamiri et al.	[72]	2022	Malaysia	Simulation and experimental study	Checked basin configurations, orifice radius, blade layouts, basin forms, and diffusers. The diffuser directs water into a basin where its vortex can move tangentially and speed up turbine flow.	Developed a hybrid turbine system that combines traditional and novel energy generation techniques to boost efficiency.	Hybrid systems represent a forward-thinking approach to maximizing energy output from hydro resources.	Hybrid systems are complex to design, expensive to implement, and require sophisticated management.
Song et al.	[73]	2010	Waco, TX	Simulation study	A system that can generate 150 kilowatts of power was constructed after a smaller system was developed and tested.	Explored the application of artificial intelligence in predicting turbine maintenance needs to prevent failures.	AI and machine learning are revolutionizing maintenance protocols for energy systems.	AI implementation requires significant data collection, processing infrastructure, and expertise.
Ullah et al.	[74]	2020	Pakistan , South Korea	Experimental study	On Gravity Water Vortex Turbine conical basin, multi-tested runners multi-multi of Savonius-type blades. Multistage Gravity Water Vortex Turbine.	Improved efficiency through advanced fluid dynamics analysis, leading to better water flow management in turbines.	Leveraging computational fluid dynamics to fine-tune turbine designs for optimal performance.	High computational demands and the need for continuous model updates to reflect real-world conditions.
Maika et al.	[75]	2023	Australia	Simulation and numerical study	Thoroughly evaluates the evolution and status of GWVHT systems, including different kinds of basins, inlet/outlet channels, and turbines.	Developed a novel approach to water intake design that reduces debris clogging and maintenance needs.	Innovative designs that simplify maintenance and increase reliability are key in new turbine developments.	Design innovations such as these can require significant upfront investment and validation.
Zarate-Orrego et al.	[76]	2016	Colombia	experimental study	A runner, a vortex room with a cylindrical shape, and a nozzle with semi-convergent edges make up this system. It pulls up water and then Pumping it into the room asymmetrically.	Developed new materials that reduce wear and increase the lifespan of turbines.	Material innovation focuses on extending the durability and efficiency of hydro turbines.	Higher materials cost and the need for specialized fabrication techniques.

Table 5. Design and operational factors and their impact on the efficiency of energy generation systems using water vortex

Factor type	Factor	Description and Impact	Results/efficiency
Design factor	Blade shape	Curved blades enhance efficiency at an angle of 44 degrees.	Efficiency increased up to 82% [10].
Design factor	Number of blades	Increasing the number of blades to 5 improves performance, but excessive increase reduces efficiency.	Optimal efficiency with 5 blades [10].
Design factor	Blade material	Lightweight materials like aluminum increase efficiency by reducing weight.	Improved efficiency due to lightweight design [46].
Design factor	Basin shape	Conical basins enhance vortex strength and improve performance compared to cylindrical ones.	Improved system efficiency and vortex power [60].
Operational factor	Vortex height	Increased vortex height improves energy conversion to mechanical power.	Higher efficiency at optimal vortex height [76].
Operational factor	Flow rate	Optimal flow rate reduces water loss and enhances system stability.	Improved efficiency and reduced water losses [72].
Operational factor	Inlet and outlet channels	Proper channel design boosts water flow and maintains vortex stability.	Better stability and higher efficiency [59].
Operational factor	Turbine rotational speed	Increasing turbine speed within a certain range improves efficiency.	Enhanced efficiency at optimal speed [19].

THE ECONOMIC EFFECTS OF THE CONSTRUCTION AND DESIGN OF THE STATION

Vladimir J et al. [78] provided comprehensive coverage of the elementary cost-benefit analysis, including extensive design, building, and preliminary testing details. They evaluate Gravitational Vortex Hydro-Power technology in comparison to established low-head hydropower systems. Using resources and technology already present in a poor nation, they proved that the Gravitational Vortex Hydro-Power infrastructure could be built with a high degree of manual labor. With its low initial expenditure and quick return on investment, it is a good choice for little rural towns that need better grid access or when the importance of design simplicity exceeds efficiency requirements. Despite the low labor costs in Peru, labor accounted for most of the expenses. Power transfer from the turbine to the generator is one of the main limitations of the system, especially when dealing with challenging high-torque working situations. Subekti et al. [11] discussed the development of a small—to medium-sized hydroelectric power station. Various technical investigations were subsequently conducted to determine the potential energy production, identify the types of components, and calculate the investment cost per kWh. Arfoa et al. [79] conducted a study and research on developing conventional hydroelectric projects on the Zarqa River to mitigate the ongoing loss of water energy by harnessing it for electricity generation. The analysis additionally ascertains the suitable site for creating the Gravitational Water Vortex Power Plant by collecting site data, including head, flow, proximity to the network, and roadways. The



Figure 17. This project is a test run for a conical basin design incorporating a runner, intake canal, irrigation canal weir, and basin construction.

Table 6. Presents the Economic impacts of station construction and design

Author	Refs	Year	Location	Type study	Type optimization & Major finding	Findings	Trends	Limitations
Vladimir J	[78]	2019	Peru	Economic analysis and technical details review	It ideal for small rural towns that need greater grid access or where design simplicity is efficiency because of its low initial cost and quick return on investment. It was demonstrated that Gravitational Vortex Hydro-Power infrastructure could be created using manual labor and resources, as well as technology available in a poor nation.	Gravitational Vortex Hydro- Power can be constructed using local resources and labor.	Focus on renewable solutions for economically disadvantaged areas.	Dependence on manual labor and design simplicity might limit broader use.
Subekti et al.	[11]	2021	Indonesia	Discussed Techno-economic research	Techno-economic studies and Vortex turbines for flat-flow river energy production are discussed. Technical studies determine power production, component types, and investment cost per kilowatt-hour.	Vortex turbines are viable for flat-flow river energy with low investment costs.	Developing economically feasible small to medium hydro stations.	Success is heavily reliant on accurate technical and cost analysis.
Arfoa et al.	[79]	2023	Jordan	Analysis of environmental and economic feasibility	The environmental and economic viability was examined using RETScreen Expert. The technical, economic, and environmental feasibility of Gravitational Water Vortex Power Plant on the Zarqa River was found in the study.	Feasibility of Gravitational Water Vortex Power Plant on the Zarqa River from technical, economic, and environmental perspectives.	Advocating for the feasibility of conventional hydro projects.	Success depends on precise site data and environmental assessments.
Dhakal et al.	[80]	2016	Nepal , UK	Evaluate technical and verify economic theory.	Using a scalable 1.6 kW system integrated into an irrigation canal, technical performance and theoretical economic studies are assessed. Results show turbine performance is suitable for rural electricity, and the Integration costs can be reduced by reducing civil works and installation.	Suitable for integrating into existing water infrastructures like irrigation canals for rural electrification.	Supports scalability and adaptability of small hydropower in rural areas.	Effectiveness contingent on the condition and suitability of existing water infrastructure.

Table 7. Economic analysis of the impact of design and operational improvements on the cost-benefit ratio in water vortex power plants

Item	Description	Economic Impact
Construction costs	Utilization of low-cost local resources and available technologies, such as manual labor.	Reduces total costs and makes the project more suitable for rural areas[78].
Maintenance costs	Simple design reduces breakdowns and requires minimal maintenance.	Lowers ongoing costs and extends project lifespan[81].
Energy efficiency	Improved basin shape, lightweight blades, and increased vortex height.	Increases system efficiency by up to 82% and boosts energy production[10].
Infrastructure utilization	Integration with existing water channels and dams to reduce additional construction needs.	Lowers installation costs and enhances economic returns by utilizing available resources[80].
Cost-benefit ratio	High return on investment due to low initial costs and increased productivity.	Ensures quick investment recovery, making the system suitable for resource-limited communities[78].

study findings suggest that implementing a system on the Zarqa River is feasible from technical, economic, and environmental perspectives. Dhakal et al. [80] stated that the gravitational water vortex power plant is a new design for a small hydropower system and is the focus of this research. Irrigation canals, reservoirs, and weirs are three examples of preexisting water infrastructure that could be used to integrate hydropower. The technical performance is evaluated, and the theoretical economic research is validated with the help of a scalable system of 1.6 kW that is constructed and incorporated into an existing irrigation canal. According to the results, turbine performance is well suited for rural electrification, and the integration costs can be minimized by cutting back on civil works and installation. Fig.(17) shows the installation of the basin. Table 6 and 7 show the economic effects of station design and construction.

MODELLING AND SIMULATION ARTICLES

Through a review of the literature on the topic, research was found that utilized numerical methods and computer programs to demonstrate and assess the efficiency of water vortex power stations. As an illustration, Pandey et al. [66] improved the Propeller Blade Angle; the propeller-type runner was optimized by simulating fifteen distinct computer-aided design (CAD) models of runners with variable blade angles at 30 rpm in a steady-state situation. The simulation results reveal that efficiency, output power, and torque all rise between 20° and 43° blade angle, and then they decline. A maximum efficiency of 20.633%, a maximum output power of 119.833 watts, and a maximum torque of 38.144 Nm were achieved. As a result of optimization, the state of water velocity is shown in Figure 18 and 19. And some of the formulas used:-

Continuity equation: -

$$\varepsilon \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho u) = 0 \quad [82] \quad (1)$$

Momentum equations: -

$$\rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = -\frac{\partial P}{\partial x} + \frac{\partial}{\partial x_j} \left[(\mu + \mu_T) \left(\frac{\partial u}{\partial x_j} + \frac{\partial u_j}{\partial x} \right) \right] \quad (2)$$

$$\rho \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial P}{\partial y} + \frac{\partial}{\partial x_j} \left[(\mu + \mu_T) \left(\frac{\partial v}{\partial x_j} + \frac{\partial u_j}{\partial y} \right) \right] \quad [83]$$

$$\rho \left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = -\frac{\partial P}{\partial z} + \frac{\partial}{\partial x_j} \left[(\mu + \mu_T) \left(\frac{\partial w}{\partial x_j} + \frac{\partial u_j}{\partial z} \right) \right] \quad (3)$$

Khan et al. [71] used numerical and simulation methods to analyze the analysis of basins. Practical research began with the study of reference basins. Due to the lack of an air core, which generates little power, the vortex did not form adequately, even at 5.5 kg/s. Thus, several basin geometric parameters were tried until a vortex with the air core throughout the height was found. Power generation using 4.5 kg/s mass flow was better in this basin. The basin was optimized by narrowing the entrance channel to smooth water intake, increasing its height to create a high-height vortex, and enlarging the bottom exit. The influence of intake channel width was examined using the dimensionless ratio of inlet channel width to basin diameter from 0.1 to 0.5. This ratio increases mass flow into the basin, raising water height until overflow from the basin's upper walls occurs. Fig. (20) shows speed streamlines, reference geometry, and air and water volume portions. Dakal et al. [40] enhanced the efficiency of the GWVPS by improving runners. Computational Fluid Dynamics studies showed that the curved blade shape was the most efficient, at 63% versus 46% for the straight blade. With a head of 0.5 meters, the runner worked as expected by the Computational Fluid Dynamics study with an efficiency of 71%.

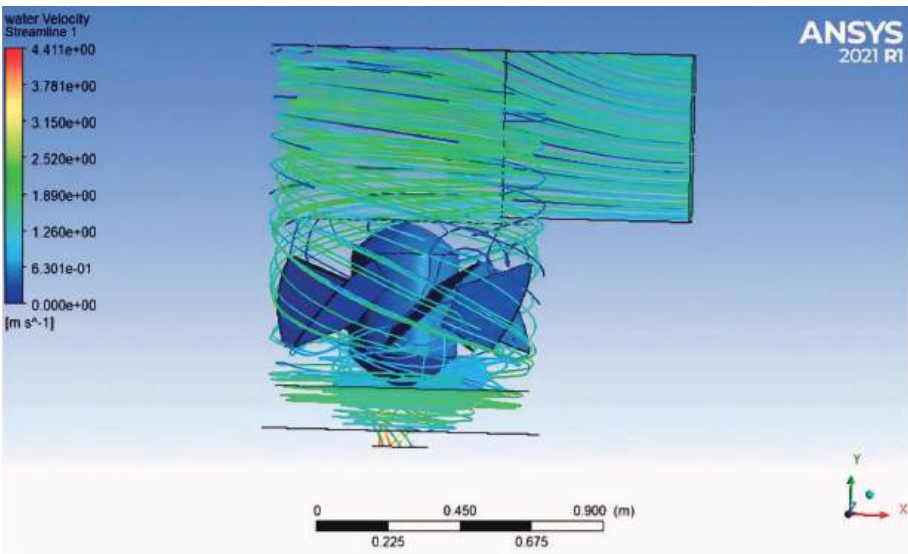


Figure 18. Decreased water velocity due to optimizing the blade angle.

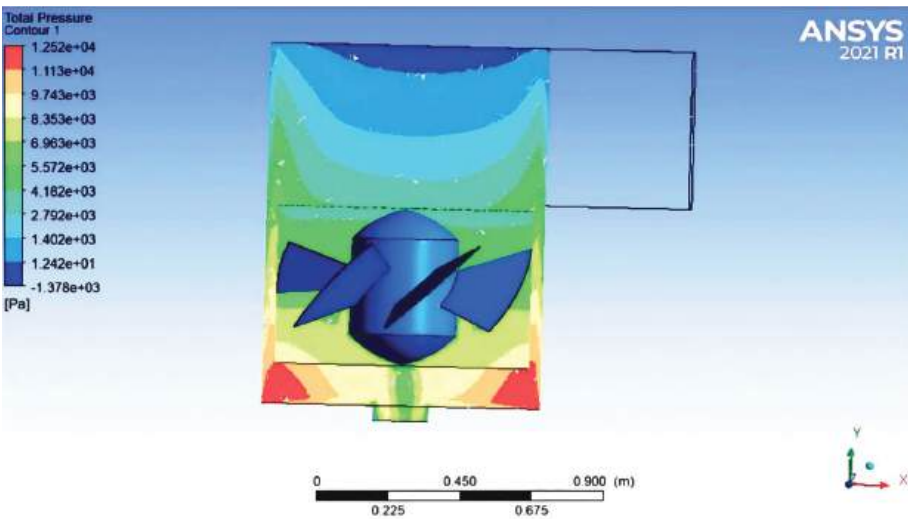


Figure 19. Decreased water velocity due to optimizing the blade angle.

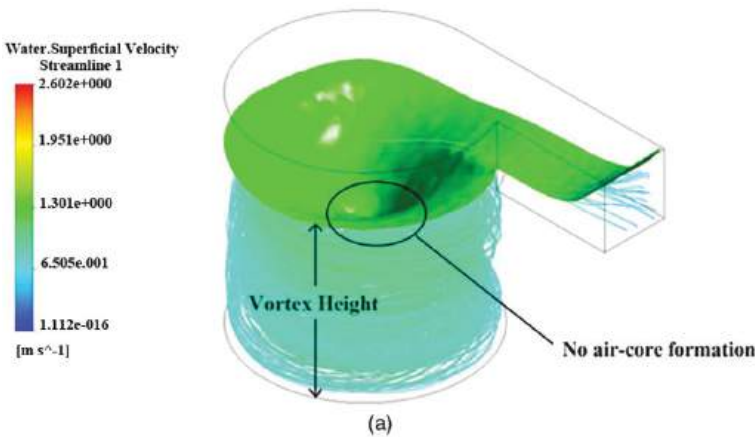


Figure 20. Speed streamlines and fluid flow.

TECHNICAL DETAILS FOR PERFORMANCE ENHANCEMENT

Finally, to improve the reproducibility and utility of the study, detailed procedural descriptions for the most important parameters are given. These include:

Blade Dimensions

Maximum efficiency: The best possible angle of blades is 44°.

Blade Diameter: 12 mm, Tested for vortex systems (small scale)

Basin Dimensions

Basin shapes: Tests were done on cylindrical and conical designs with inlet-to-outlet diameter ratios of 3.26, 4.9, and 6.12.

The best height of power extraction is between 65 and 75 % of the total height from the basin.

Turbine Dimensions

Turbine height: Five configurations were evaluated from 150 mm to 650 mm to measure the impact on torque and efficiency.

Turbine depth: Best Performance observed at 410 mm depth

Discharge Orifice Size

The enhanced overall turbine efficiency through higher vortex intensity occurred due to the diminished discharge orifices.

This comprehensive breakdown of these factors will improve the transparency of the experimental configuration and aid in similar future designs by providing rich insight into system topology and layouts.

GRAVITATIONAL WATER VORTEX POWER PLANTS: SUPPORTING RENEWABLE POLICIES AND SUSTAINABILITY

- **Water Energy:** Gravitational Water Vortex Power Plants encourage green natural resources—leading to carbon neutrality by 2050.
- **Transforming Energy Access:** This innovation inexpensively delivers electricity to sparsely populated and secluded regions, thus aiding sustainable development frameworks and poverty targets.
- **Carbon Emission Mitigation:** By decreasing reliance on fossil fuels, Gravitational Vortex Hydro-Power helps reduce your environmental footprint and meet emission reduction commitments.
- **Cost Effectiveness:** The low cost of construction and maintenance makes this technology suitable for resource-limited communities, leading to widespread adoption.
- **Adopting GWVH Projects to Promote National Energy Policies:** Gravitational Vortex Hydro projects can be part of the national energy policies, and they will

participate in the energy transition towards sustainable energy sources.

CHALLENGES IN GRAVITATIONAL WATER VORTEX HYDROPOWER

In this section, some of the limitations and challenges affecting the performance of the system are reviewed in the form of points:-

- **Flooding Risk:** Water vortex stations face the challenge of flooding, which can disrupt operations and damage infrastructure if not properly managed.
- **Water Availability Impact:** Water availability directly affects electricity production and energy costs, with droughts or limited water supply significantly influencing this relationship.
- **Noise Issue:** The sound of water striking the blades can be highly disruptive to people, creating challenges for installing these stations close to residential areas.
- **Impact on Aquatic Life:** Hydroelectric power plants significantly disrupt fish life cycles by causing major alterations in natural water flow patterns.
- **Rotational Speed Limitation:** The slow rotation of water wheels poses a challenge for electricity generation, necessitating high-ratio gearboxes. These gearboxes are costly and can account for a substantial portion of the total system cost, ranging from 25–30% and potentially reaching up to 40–45% [78].
- **Rotational Speed Limitation:** Water wheels' slow rotation represents an area requiring high-ratio gearboxes for effective electricity generation. Such gearboxes are expensive and constitute a significant part of total system cost, typically 25–30%, potentially up to 40–45% of total system cost [11].
- **Speed Challenge of Turbines:** Turbines are low-speed by nature, so high performance requires using spiral-shaped designs. However, these layouts are sophisticated and require high technical knowledge to fabricate [68].
- **Water Vortex Turbine Problems with Vertical Axis:** The main problems of water vortex stations' vertical axis turbines are low starting torque, torque instability, and low efficiency. Due to their design, which causes them to perform poorly at start-up, the use of auxiliary starting systems, sometimes electrically, mechanically or a combination of the two, is typically required [84].
- **Efficiency Limitation in Hydro Vortex Stations:** One drawback of hydro vortex stations is that increasing the number of turbine blades reduces the system's overall efficiency [63].

CRITICAL ELEMENTS INFLUENCING THE PERFORMANCE OF GRAVITATIONAL WATER VORTEX POWER FACILITIES

Operational, design and economic factors were highlighted as key factors influencing GWVHS efficiency, and

the study identified these characteristics as affecting their efficiency. Results showed that changing the turbine blade angle to 44 degrees using a five-blade design increased the efficiency to 82%, and higher water flow speed and depth greatly influenced the system's performance. It was established that conical basins perform better hydraulic losses and respective drainage adjustments than cylindrical basins. The optimized designs can reduce costs economically and help achieve long-term sustainability by saving costs on the part of these power generators. We learned how computational fluid dynamics models allow the designer to consider the system variables and analyze and optimize those variables to overcome challenges such as initial construction costs and environmental impacts. These results highlight the promise of the Gravitational Vortex hydro-station as a sustainable energy source, especially in under-resourced regions.

RESULTS AND DISCUSSION

This study seeks to review many researches and studies related to improving and raising the efficiency of stations, as it turned out that several factors must be taken into account because they have a vital role in increasing and developing the performance of hydroelectric power (water vortex power stations). Hence, the researcher concluded that the most influential factors in the generation of the water vortex are the turbine shape, diffuser and cylinder shape that forces water to form a vortex. The results of research and simulations show that the efficiency of the curved blade runner is 3.65% greater [21] than that of a flat runner. The researchers also found that the intensification of the nozzles improved the conical basin's vortex generation and the turbines' performance. Another research is to increase turbine efficiency and rotation speed by increasing the blade angle and surface area using barrier plates. The 5-blade design with a blade angle of 44 degrees resulted in an 82% efficiency of the curved blade shape [10]. This led to an increase in total electricity generation. The addition of a booster turbine to the primary turbine resulted in a 20.4% increase in overall efficiency and an increase in electrical capacity by 3.48 watts [39]. Studies have shown that the aluminum metal used in the manufacture of the turbine has a noticeable effect compared to steel metal, where the efficiency of aluminum and steel metals reached 34.79% and 33.56%, respectively. Studies have shown that adding a small intake channel under the basin leads to a 60% saving of the energy generated by the vortex. Etc [67]. There are several proposals and recommendations for future studies of design and operational variables:

1. Give the importance of economics in developing water vortex power plants, ensuring affordable systems and innovative financing mechanisms to enable widespread adoption in developing and developed countries.
2. Research on Additional Influencing Factors: Future research should investigate other factors that can

influence the performance of the water vortex stations, such as flow dynamics, environmental conditions, and material durability, to improve system efficiency and sustainability.

3. Turbine Blade Analysis: A thorough evaluation of various turbine blade configurations is critical to determining the ideal setup to maximize energy extraction while minimizing wear and maintenance costs.
4. Where the Real Science Begins: Conduct a detailed study on the diffuser geometry and material to improve the tangential water flow entering the basin, the key point where vortex flows are generated, consequently leading to energy generation.
5. Vortex height investigation: Further research is needed on vortex height to identify the relationship between energy generation, vortex stability and structural needs of different applications.
6. Advanced Mechanical Linkage Systems: Studying complex mechanical and electromechanical linkage models connecting water turbines with electricity generators could lead to greater efficiencies in power transfer and decreased mechanical losses.
7. Optimal Sizing and Location of Water Vortex Stations: More research can be done to determine the best size and placement of water vortex stations in different contexts to maximize energy production while minimizing environmental disruption.
8. Energy Storage Optimization: Explore ways to optimize the design and operation of water vortex power plants to work seamlessly with energy storage solutions, where energy generated can be stored and released for use when demand is high.
9. Scalability and Modular Design: Research into scalable and modular designs for water vortex stations could facilitate deployment in diverse geographical and hydrological conditions.
10. Intelligent Monitoring and Management Systems: Intelligent/faulty monitoring and auto management systems can improve operational performance, prevails signalling, and optimize real-time functioning.
11. Hybrid Energy Systems: Water vortex stations can be combined with other renewable energy sources like solar or wind, creating hybrid energy systems tuned for continuous power output.
12. Community and Economic Benefits: Examination of the economic and community orientation of water vortex stations in terms of the social, community and economic benefit potential of this technology for remote or underserved areas in job creation and local development.
13. Material Innovation - New materials with unprecedented properties could allow for lighter, low-corrosion, and more economical materials, leading to greater durability and effectiveness for the system.
14. Design to Adapt to Climate Changes: Research should aim to develop systems that can adjust to the effects of

climate change, like varying levels of water and more extreme weather, including the reserve capacity in the system to support long-run sustainability.

15. Investigating the Effect of Height of Vortex and Flow Rates: Practical experiments are suggested to examine the effect of the height of the vortex and energy loss due to the variation in flow rates to discover the best operating conditions of systems.
16. 3D Model Design for Diffuser: Digital simulation tools (i.e. ANSYS and Computational Fluid Dynamics) are recommended to study the effect of different diffusers and blade angles (44°) on system efficiency.
17. Conduct Field Testing in (the) Real World Operating Conditions: to assess system performance under low water flow conditions in natural environments through field experiments, improving flexibility and efficiency in rural areas.

These converging perspectives would streamline the future research and development trends for water vortex power plants, making them a more effective, sustainable, and widely applicable renewable energy solution.

RESEARCH APPLICATIONS AND THEIR IMPLICATIONS FOR SUSTAINABLE ENERGY DEVELOPMENT

Our work focuses on Gravitational Water Vortex Hydropower Systems. It offers substantial contributions towards enhancing turbine performance and achieving more sustainable energy practices in answer to the world's increasing energy demand, particularly in rural and remote locations affected by energy scarcity. By learning from all the elements that affect turbine designs, including blade angles, materials and types of basins, we were able to identify major performance boosts. After modelling the factors that influence turbine efficiency, we found that blade angle significantly affected turbine performance, enabling us to improve efficiency (82%), torque (1.23%) and hydrokinetic energy production while decreasing loss. Furthermore, the economic modelling of how feasible it is to run these systems in rural areas demonstrates how they provide sustainable energy at lower costs than traditional energy systems. These systems assist in reducing carbon emissions, an efficient alternative to fossil fuel usage that promotes environmental sustainability. Their capacity to offer energy states away from the grid encourages energy independence. It aids international endeavors to achieve the Sustainable Development Goals (SDGs), especially in developing nations, where people across the globe battle to obtain energy. These low-cost and environmentally sound energy solutions enable small and medium-sized projects in developing countries to produce sustainable energy in areas with limited access to traditional grids. Also, optimized basin designs and high-performance turbines enhanced economic efficiency for such systems, making them a good candidate for investments and a way for communities to harness low-cost sustainable energy.

The results of this study lay the groundwork for improving renewable energy technology worldwide, helping to ensure sustainable development through increased efficiency of small hydropower systems. Thus, by studying and analyzing available technological and economic tools, such systems can support the transition to a low-carbon economy worldwide, enabling these countries to respond to climate challenges and achieve carbon neutrality by 2050.

CONCLUSION

This research looked into the various operational, design, and economic aspects that affect how well Gravitational Water Vortex Hydropower Systems perform. The main goals were to improve efficiency, cut down on losses, and assess financial viability. The findings showed that changing the turbine blade angle to 44° increased its efficiency to 82%, which greatly enhanced hydrodynamic performance and reduced energy losses. Additionally, using advanced materials like aluminum and specially designed alloys led to a 1.23% rise in torque, which improved the turbine's stability and durability over time. In terms of basin design, experiments revealed that using conical basins with optimized nozzles boosted power output by 60% and cut hydraulic losses by 25% compared to traditional basin designs. Also, narrowing the central gap in the turbine blades improved flow efficiency by 10.25%, which helped reduce erosion and lengthen the operational lifespan. Likewise, refining the water channel design resulted in a 15% increase in flow velocity, maximizing energy production, especially in low-head locations. From an economic standpoint, computer models showed that better turbine designs could lower operational costs by 18% and maintenance costs by 20% due to less wear and longer turbine life. Additionally, the research found that having five blades instead of three or seven struck the best balance between performance and efficiency, leading to a 14% increase in energy output.

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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.

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Review Article

Comprehensive review of algae biodiesel: production, engine performance, and emission characteristics with additives

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ABSTRACT

The combustion of petro-diesel fuels in internal combustion engines is one of the contributing factors to increased environmental air pollution. The combustion byproducts are toxic exhaust emissions, which are dangerous to human health and the environment. Therefore, the need for sustainable and renewable fuels, like biodiesels, is gaining significant attention to regulate the toxic exhaust emissions from compression ignition engines. Biodiesels are popular due to their biodegradable, renewable, eco-friendly, non-toxic factors and their ability to be employed in diesel engines without major modifications. These features have attracted researchers to investigate different biodiesels and their performance and emissions compatibility with existing engines. In this systematic review, research findings from the year 2009 to 2024 on biodiesel production from second and third-generation feedstocks, their significant fuel properties, and fatty acid compositions are critically reviewed. In addition, combustion performance and exhaust emission characteristics of different diesel and biodiesel blends are discussed. Similarly, additives are gaining popularity for improving the significant biodiesel properties and enhancing combustion performance with regulated emissions. Therefore, the influence of additive blending in biodiesel is also highlighted. Further, a comprehensive review was conducted on the emergence of algae biodiesel processing methods of algae to meet the future energy requirements in engines. The findings from the literature confirmed that the presence of high saturated fatty acid composition in green algae biodiesel mitigated the upsurge in nitrogen oxide emissions and suggested that green microalgae biodiesel is best suited as a potential source of biodiesel production. Finally, this review reported that compared to non-edible biodiesel, algae biodiesel resulted in enhanced combustion propensity, engine performance, and reduced nitrogen oxide emissions, a prominent source of green, sustainable fuel to replace fossil fuel.

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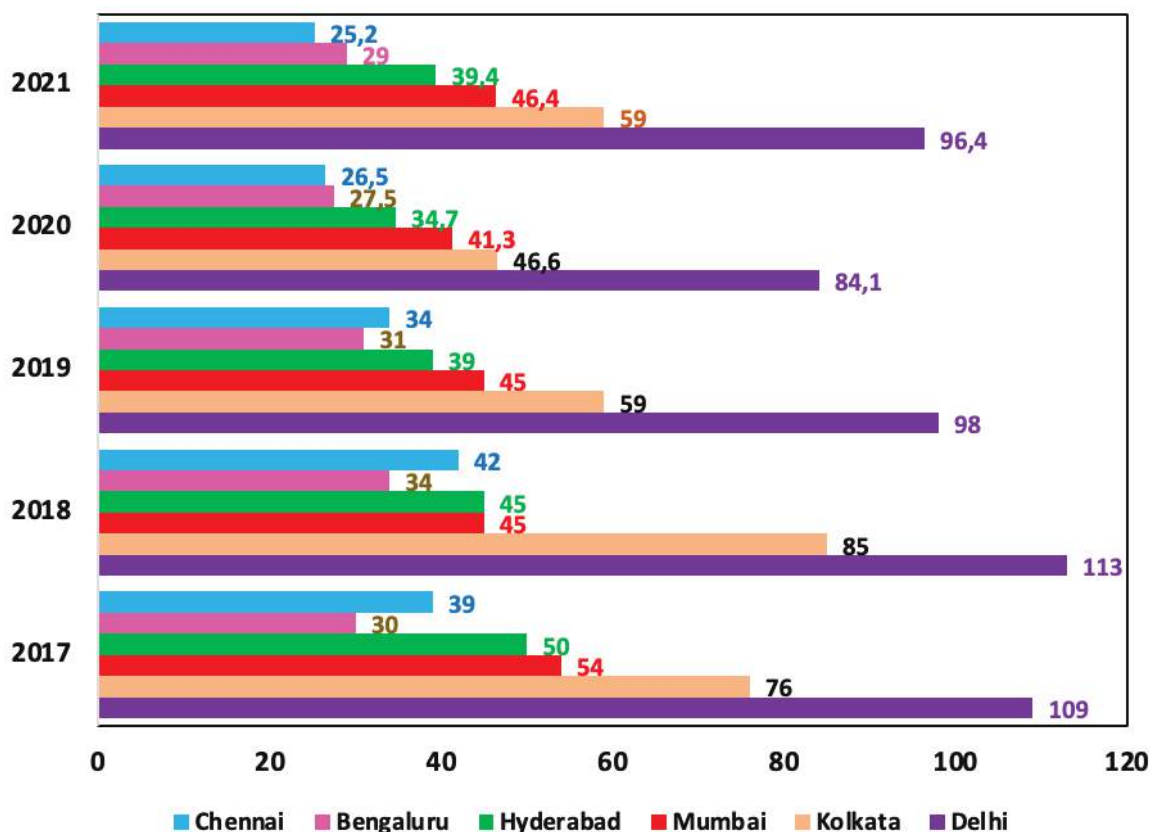


Figure 1. Annual average ($\mu\text{g}/\text{m}^3$) of PM 2.5 in Indian major cities (2017 – 2021).

the practical applications of internal combustion engines are growing gradually due to their high-power output and marginally low fuel consumption. In addition, they found almost all industrial applications, from small to large scales, in today's world.

The primary concern of these combustion engines is their toxic exhaust emissions[14]. In the present situation, the dangerous pollutants exerted by automobile engines, including carbon monoxide (CO), carbon dioxide (CO_2), nitrogen oxide (NO_x), particulate matter (PM), unburned hydrocarbons (UHC), and smoke, are considered as a severe threat to humankind. Different research groups have made various proposals to regulate this issue. Replacing internal combustion engines with electric motors and replacing existing fossil fuels with renewable and sustainable fuels are attractive proposals that are gaining wide attention. In addition, the entry of 17 sustainable development goals into the force has motivated the researchers to achieve *Affordable and Clean Energy* by 2030. Within this limited time frame, burning and using sustainable alternative fuels to replace fossil fuels are considered feasible solutions compared to replacing existing engines with other technology like electric motors. However, the fossil-derived petro-diesel fuels presently used in compression ignition (CI) engines are at

the exhaustion phase, which led to a focus on a sustainable energy source known as biodiesel.

EMERGENCE OF BIODIESELS

Biodiesels are widespread for their clean burning, high oxygen levels, and low carbon content[15] and are famous for producing less emissions than conventional diesel fuels [16]. Biodiesels are used in the CI engines as neat biodiesel or blended. They have been proven to be one of the emerging sources for replacing fossil fuels. Raman et al. [17] used rapeseed biodiesel to estimate the engine behavior of a conventional DI diesel engine. The investigation concluded that the B25 biodiesel blends enhanced engine performance, resulting in high BTE and reduced engine exhaust emissions. Similarly, Chozhavendhan et al. [18] discussed various parameters influencing biodiesel production, like lipid/fat content, catalyst, molar ratio, and purification process, to reduce the unit and operation cost to produce biodiesel. The study reported that biodiesel from lignocellulose, edible, non-edible oils, and micro and macroalgae emits less pollution than fossil fuels. Prasada Rao et al. [19] conducted an experimental investigation using palmyra biodiesel blends of POME10, POME20, POME30, and

POME40 to estimate the engine performance and emissions. The findings reported that the exhaust emissions for the blend POME20 are reduced for CO by 15.86 %, UHC by 18.5%, and smoke emissions by 14.28 % compared to diesel.

BIODIESEL GENERATIONS

Biodiesels are extensively categorized into first-generation (IG), second-generation (IIG), third-generation (IIIG), and fourth-generation (IVG). A detailed overview of the four-generation biofuels and their latest developments were highlighted [20], and various feedstocks, oil, and biodiesel yield percentages, and characterization of biodiesel fuel properties for the production of biodiesels were reviewed [21]. The first-generation biodiesels (IG) come under edible oils; these crops produce biodiesel and are used for food consumption. Gradually, the utilization of IG biodiesels has raised concerns about food vs. fuel. Hence, to overcome the drawbacks associated with biodiesel, IIG biodiesels exist; these are linked to the non-edible oil category, such as seed oil, waste vegetable oils, and animal fats [22]. Rezanian et al. [23] reviewed different non-edible sources (IIG) for biodiesel production using the transesterification method as these are economical, and there is no issue or debate on food vs. fuel, likewise in edible sources. However, various second-generation biodiesel feedstocks are tested on the CI engine to observe the performance characteristics and emission levels. Correspondingly, experimental studies are conducted by mixing additives [24–26] into the biodiesel blends.

Aruna Kumari et al. [27] tested the engine emissions using lemon peel biodiesel blends of 10%, 20 and 30% vol.% and trailed on the CI engine. The test observations revealed that CO, HC, and smoke emissions were recorded at 20% of lemon peel oil blend by 25%, 25.6%, and 15.44%, and an increase in NO_x was observed when compared with diesel. With this motivation, Enweremadu et al. [28] used canola and sunflower biodiesel feedstocks and simulated the engine behavior using C++ software on a Mercedes Benz DI engine. The output indicated that biodiesel possesses higher Specific fuel consumption (SFC), brake power (BP), and thermal efficiency than diesel. Further, high NO_x and low smoke emissions were recorded compared to diesel. Thus, several researchers extended their investigations to mitigate emissions by employing additives in IIG biodiesels [29–31].

Accordingly, Ahmad S et al. [10] mixed oxygenated additives, methanol or diethyl ether in Neem biodiesel, a second-generation biodiesel, due to the rise in NO_x emissions. The engine experimental test showed that BSFC was reduced by 10%, CO and HC were decreased by 25%, and NO_x emissions lowered when Neem biodiesel was tested with diethyl ether. In a similar form, the addition of heptane butanol and diethyl ether was mixed in *Jatropha* biodiesel, and the output results showed that there is an enhancement in BTE and an increase in BSFC by 5–20% and a reduction in engine emissions by 3–12%. Similarly, *Calophyllum inophyllum* biodiesel enhanced engine efficiency and lowered

emissions by adding oxygenated additives butanol, pentanol, decanol, and hexanol. Thus, overcoming a few shortcomings in biodiesels, such as higher density, kinematic viscosity, lower calorific value, and rise in NO_x, has instigated researchers to add oxygenated additives in different biodiesel generations.

Based on the investigations contributed by enormous numbers of researchers, it is understood that biodiesels can replace fossil fuels. Though IIG biodiesels such as waste cooking oil gained positive attention as this feedstock is obtained at low cost and readily available, the main set back observed over WCO biodiesel was the filtering and processing of WCO [32]. Thus, the hitches raised in IG and IIG biodiesels have directed the approach towards sustainable, clean, and green fuels, known as third-generation algae biodiesels (IIIG).

Algae are ubiquitous photosynthetic organisms on the earth, identified as the fastest-growing biomass. Algae are known for their abundant availability, require less land space, rapid growth, use of natural resources such as sunlight CO₂, and use as food supplements [33]. The processing of algae is a crucial agent in the preparation of algae oil for biodiesel [33]. In particular, algae cultivation is significant for growing and harvesting algae. However, a slight increase in economic viability is a setting concern during harvesting [34], and separating algae layer by layer [35] consumes more time in algae processing. Hence, algae are the best source of biodiesel, producing clean-burning renewable fuel. In recent years, several researchers [36–38] have turned their interest toward fuel production from one of the oldest living creatures on the earth, known as algae, serving as a biofuel source.

With this motivation, an experimental investigation was performed on one cylinder, 4-stroke, at a speed of 1500 rpm, for the algae biodiesel blends of 5%, 10%, 20%, and 30% [39] to estimate the engine combustion performance and emission characteristics. The tested results reported that at blend up to 5%, the BTE decreased, and the reduction in CO and UHC was observed up to 28%. Finally, a gradual increase in NO_x emission (up to 13%) was shown in the experimental tests. And a volume fraction of 20% exhibited better results compared to other blends. However, additive mixing in biodiesel plays a significant role in overcoming the rising NO_x emissions. Based on this, Sekharraj K et al. [40] conducted an experimental study on green microalgae biodiesel blends for 80–20 (80% diesel and 20% biodiesel) mixed with Bi₂O₃ nanoadditives to determine the performance and emission characteristics. The engine test results reported that using nano additives enhanced the engine's performance and reduced the engine emissions. The schematic representation of the biodiesel production based on different generations is represented in Figure 2 [22].

The fourth-generation biodiesel (IVG) deals with the genetic modification of microalgae, fungi, yeast, and cyanobacteria [1], [22]. The primary benefit of employing IVG biodiesel was its ability to grow microalgae and collect significant

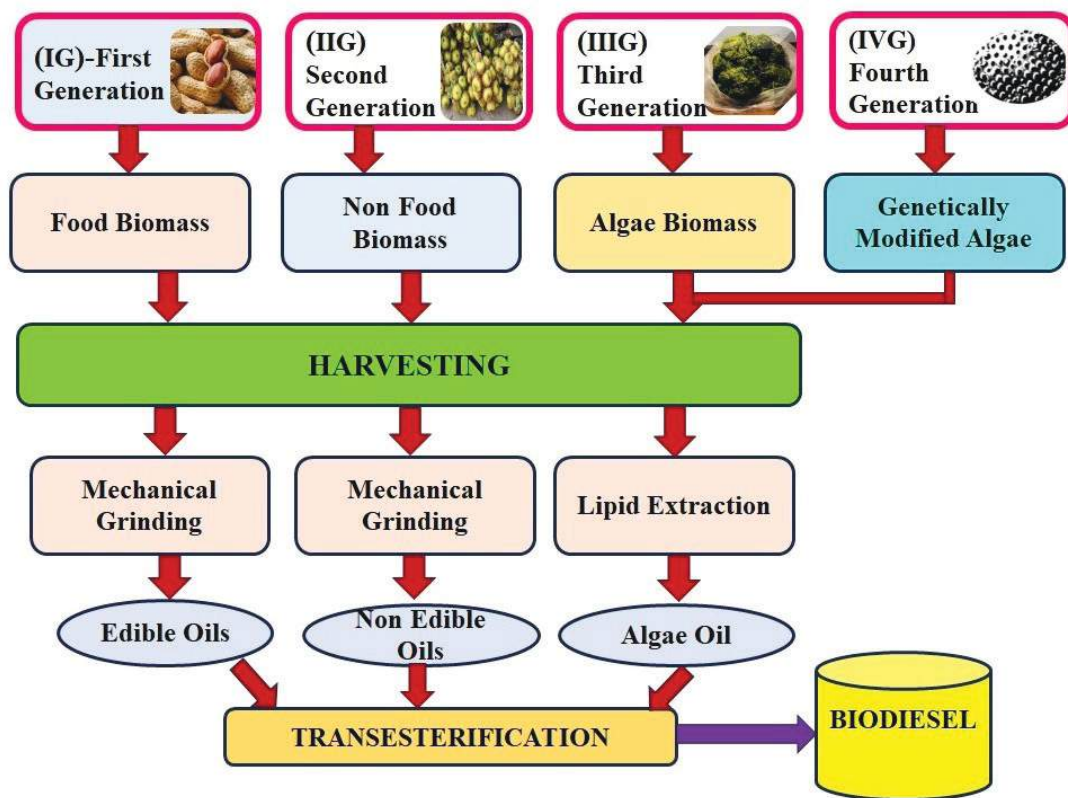


Figure 2. Different generations and their process of production of biodiesel.

amounts of CO₂ by boosting productivity and availability by incorporating modern technologies. However, this technology cannot be employed with all microalgae species due to a lack of gene and biological understanding [41]. Therefore, third-generation algae are recognized as the most significant potential to produce algae biodiesel and are considered one of the emerging sources of fuel to run and test on existing diesel to replace fossil fuels.

The observations from the biodiesel generations revealed that the IG edible oils created a debate on food vs. fuel, which hinders using fuel for transportation. Though IIG biodiesel (non-edible oils) play a vital role in improving efficiency and reducing emissions, the limiting factors, such as the availability of land to cultivate the non-edible oil crops and high fatty acid profiles, led to opt for IIIG algae biodiesel as one of the better options for replacing conventional fuels in CI engines since the land area required to cultivate the algae [36] is significantly less and reported high oil yields compared to the IG and IIG.

However, the rise in NO_x emissions in second-generation [27-28] and third-generation biodiesel [29] confirmed an increase in NO_x emissions. Thus, to mitigate emissions, several researchers extended their investigations by employing additives in IIG biodiesels [29-31] and IIIG biodiesels [41], resulting in improvements like complete combustion in ignition chambers and reduction in temperature,

leading to enhanced engine performance and lowers engine emissions.

Sustainable fuels like biodiesel are compact with existing engines, and their application doesn't require any major modifications in the existing engine designs, which made them attractive and popularly investigated by different research groups [42]. Therefore, in this review article, a clean burning and sustainable biodiesel from the 3rd generation oils is critically reviewed, highlighting its significance at different stages, such as its production process, significant properties, and applications in diesel engines in terms of performance and emissions. In addition, various additives that improve the overall properties with their addition to biodiesels are also reviewed. To further strengthen the readership of this review article, a bibliometric analysis [44] is carried out to understand the past and current research trends on this topic. For this purpose, the keywords "Algae Oil" AND "Production" are taken from the Scopus database, and their results as network graphs are highlighted in Figure 3, which clearly indicates that research on IIIG oils is emerging.

THIRD-GENERATION ALGAE OIL

Algae are living organisms primarily found in different breeds of aquatic environments and can reproduce faster than land-grown plants. These can survive in freshwater resources, ponds, rivers, lakes, rocks, marine, brackish,

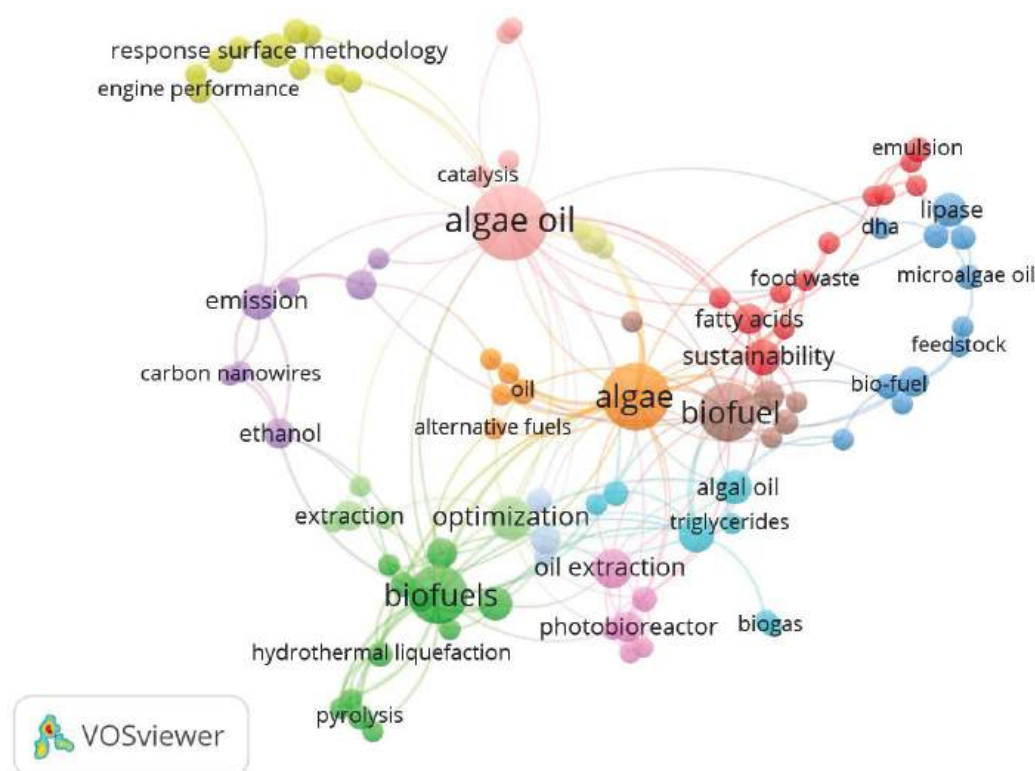


Figure 3. Bibliometric analysis for algae oil and its production.

municipal, industrial wastewaters, and moist soil surfaces. The significant features of algae are ease of cultivation, utilization of natural sunlight and CO₂, less land area for growth, more effective than terrestrial seed crops [26], higher growth rate, use as a food supplement, and mitigation of environmental pollution.

The invention of algae is not new in the 21st century; algae as biodiesel originated in the 19th century. These are classified as microalgae and macroalgae. The microalgae primarily consist of unicellular structures measured in micrometers. These usually grow in open ponds such as freshwater tanks, reservoirs, etc., and closed ponds (photobioreactors), whereas macroalgae are multicellular in structure, measured in inches, and grow large. These are found in marine seawater, wastewater, and ponds [36]. The main characteristics of microalgae growth depend on productivity, low production and maintenance cost, temperature control, pH, oxygen, and reliability. Among these, microalgae species have higher oil content and can extract oil in large amounts [44]. Thus, microalgae are considered the best form of green energy for supplying the world's demand for transportation fuels.

OVERVIEW OF ALGAE: TYPES OF ALGAE

Algae species are one of the oldest living organisms. Worldwide, over 60,000 microalgae species were spotted,

out of which 35000 species were recognized as the potential for the generation of biodiesel [44]. These are the most exciting future solutions for the energy crisis, especially that of transportation fuel.

Based on their growth and climatic conditions, microalgae species were grouped into Bacillariophyta (diatoms), Charophyta (Stoneworts), red (Rhodophyta), green (Chlorophyta), brown (Phaeophyta), blue-green (Cyanobacteria), red-green, golden-algae (Chrysophyta), phylankton, seaweeds and other algae strains [20], [26]. Among these [45], green microalgae (Chlorophyta) bagged with more essential benefits than other algae species since these are unicellular and aquatic. In India, green microalgae species grow abundantly, and the climate conditions are well-suited for growing green algae in large quantities. Thus, the most prolific strains are Chlorophyceae (green algae) [44], which can grow well in open ponds, accumulate lipids at low temperatures like 10°C, and have a better tolerance to CO₂. Some green-microalgae species, *Chlorella*, *Spirogyra*, and *Spirulina*, are cultivated and grown in an open atmosphere.

In particular algae species, the oil content ranges between 20 to 40% of its dry weight, whereas, in some specific algae strains, nearly 80 to 85% of the oil content was identified. Thus, if suitable species were identified, more than 85% of the oil could be extracted [38]. The oil yield and land area

requirement of various biodiesel feedstocks are presented in Table 2 and Figure 4.

Thus, it is observed from Table 2 that algae species are recorded with high oil yields. Figure 4 shows less land area required to cultivate the algae than second-generation biodiesel.

ALGAE PROCESSING METHODS

The selection of the site for algae cultivation is one of the basic requirements for algae biodiesel production. The four main processing methods of algae are cultivation, harvesting, oil extraction, and biodiesel preparation. Chowdhury et al. [46] demonstrated the importance of microalgae site selection for algae, reviewed different microalgae cultivation, harvesting, oil extraction, and biodiesel production methods, and reported that green algae possess a high growth rate. In an ideal environment, green algae can double their biomass in less than a day and possess massive lipid content (more than 50%), an excellent biodiesel production source.

ALGAE CULTIVATION

The essential factors required to absorb and convert natural sunlight into chemical energy are CO_2 and H_2O . The Algae can be cultured in two processes: natural cultivation and artificial cultivation process [26]. In the natural cultivation process, the algae are grown in open ponds,

Table 2. Oil yield produced by various biodiesel feedstocks

No.	Feedstock	Oil Yield (L/hect./year)
1	Camelina	915
2	Castor oil	1413
3	Corn	172
4	Cotton	327
5	Canola	1190
6	Coconut oil	2689
7	Jatropha oil	1892
8	Hemp oil	364
9	Mustard Seed	570
10	Maize	172
11	Palm oil	5950
12	Peanut oil	1057
13	Safflower	776
14	Sunflower	1090
15	Soybean oil	448
16	Microalgae biomass with 30 % oil (L/wt.)	58,700
17	Microalgae biomass with 70 % oil (L/wt.)	136,900
18	Algae Species	100,000

Source: [21], [45]

raceway ponds, and freshwater tanks, whereas in artificial cultivation, algae growth is initiated in closed reactors

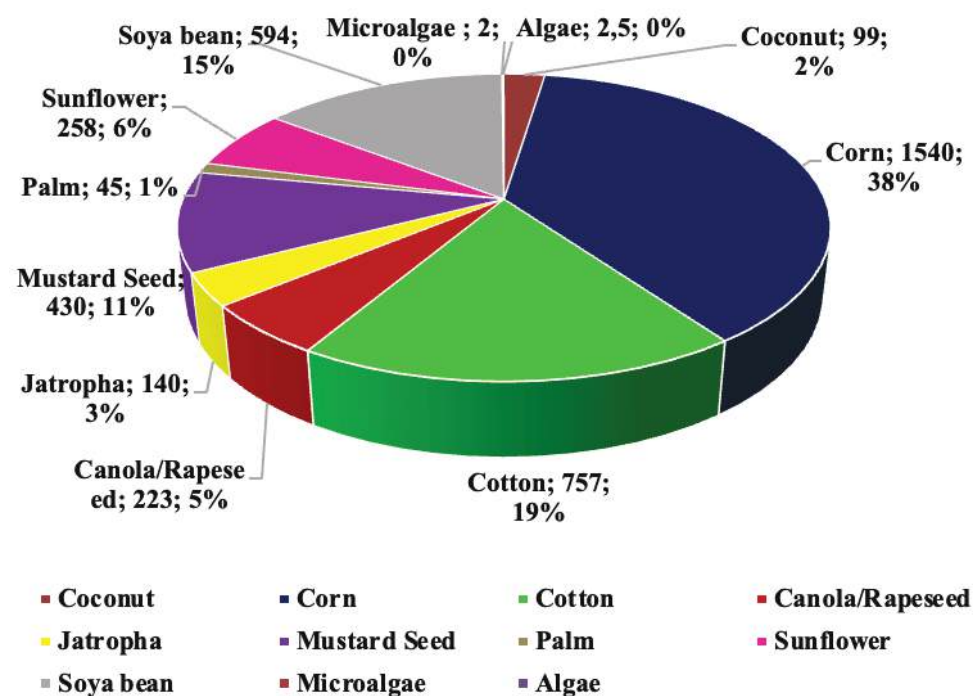


Figure 4. Land area required to produce oil (Mha).

(photobioreactors). Subsequently, the naturally-grown algae can be cultivated in freshwater tanks, directly collected from lakes, rivers, and ponds, and found as vats [45].

Kalyani et al.[35] cultivated the naturally grown green algae in the freshwater tank without chemical fertilizers; only sunlight, CO₂, and temperature are required to produce the algae. The basic parameters needed to maintain the algae growth and reproduction are temperature (20 °C to 30 °C), salinity, light intensity (33 μmol m⁻²s⁻² to 40 μmol m⁻²s⁻²), and pH value (5.5 to 7.5). Another researcher [47] collected the naturally grown green algae from open ponds and reservoirs without constructing a water tank for cultivation. Further, algae processing methods such as harvesting, oil extraction, and biodiesel production were processed using algae collected from open ponds.

HARVESTING

After algae growth, the wet algae biomass is harvested using various techniques [48], such as centrifugation, sedimentation, flocculation, and filtration. V. Ananthi et al.[48] studied the merits and demerits of several harvesting techniques and suggested that magnetic separation and chemical coprecipitation are the most commonly used methods. Thus, based on the above harvesting methods, the amount of processed dried algae biomass acquired can be further processed for algae oil extraction. J. Sen Tan et al.[49] provided information on microalgae culturing, harvesting, and extraction methods using advanced ionic liquids. Thus, using various algae harvesting methods, the algae is ready to extract the algae oil and determine the fuel properties.

OIL EXTRACTION PROCESS

The extraction process involves transferring the solid dried algae powder into a liquid phase. The oil extraction techniques were classified into chemical and mechanical methods [50]. Chemical methods include solvent extraction, Bligh and Dyer's, Ionic liquid extraction, Supercritical CO₂ (S-CO₂) techniques, and mechanical methods include oil expeller process, Osmotic shock, microwave-assisted and

ultrasonic assisted process[51],[52]. The oil yield is estimated using Equation 1.

$$\text{Oil Yield Efficiency (wt. \%)} = \frac{\text{Mass of the Algae Oil Extracted (grams)}}{\text{The total mass of the dried algae}} \times 100 \quad (1)$$

Studies from various methods have reported that supercritical CO₂, solvent extraction, and oil expeller techniques are viable methods to extract. Among these, the solvent extraction method is most efficient and recovers the oil up to 40 – 78%[45],[52],[53]. This method is economical, suitable for small scale, and has high extraction efficiency. However, to extract maximum oil, this method requires more time and a massive amount of algae powder [50].

Transesterification Process

Transesterification is the most widely adopted method to produce biodiesel at the micro and macro levels. The raw oil recovered from the crop seed and algae oil extraction process is converted into biodiesel using various preparation methods such as mixing raw oil, pyrolysis, dilution, microemulsions, and transesterification process[23],[51]. Among these methods, the transesterification method is the most appropriate method, as it reduces the high viscosity of the raw oil [53], and 98% of the methyl ester can be produced using this method [54].

The influence of different types of catalysts, such as homogeneous and heterogeneous catalysts used in the transesterification process, yielded high-purity biodiesel and glycerol, which depends on the long-chain fatty acids. Kalyani et al. [35] used homogeneous catalyst sodium hydroxide (NaOH) and methanol (CH₃OH) to initiate the transesterification reaction process for the algae biodiesel, and the saturated FFA was 73.95 wt.%. [55] used a waste chicken eggshell as a heterogeneous catalyst and obtained 1.9 wt.% FFA using the titration method. In a similar approach, Kalyani et al. [47] used chicken egg shell waste and methanol as a heterogeneous catalyst to prepare the algae biodiesel and obtained a saturated composition of FFA of 68.39 wt.% and an unsaturated composition of 29.3 wt.%.

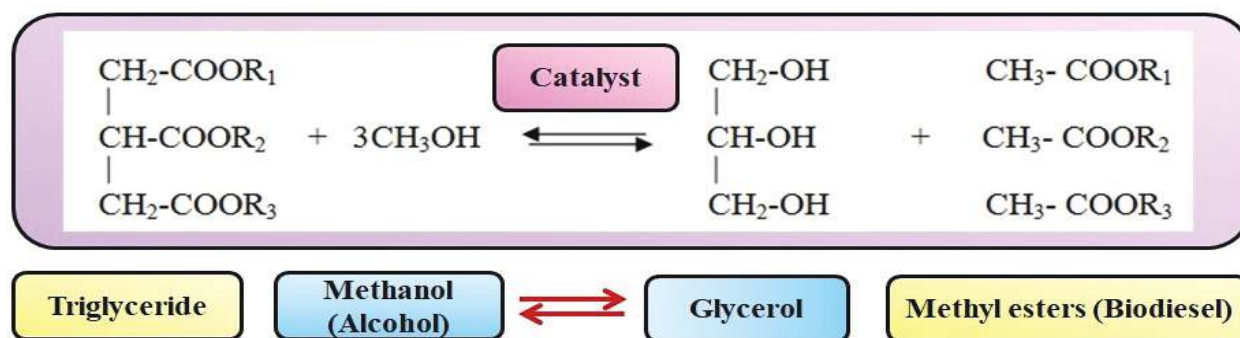


Figure 5. Transesterification reaction of triglycerides with alcohol.

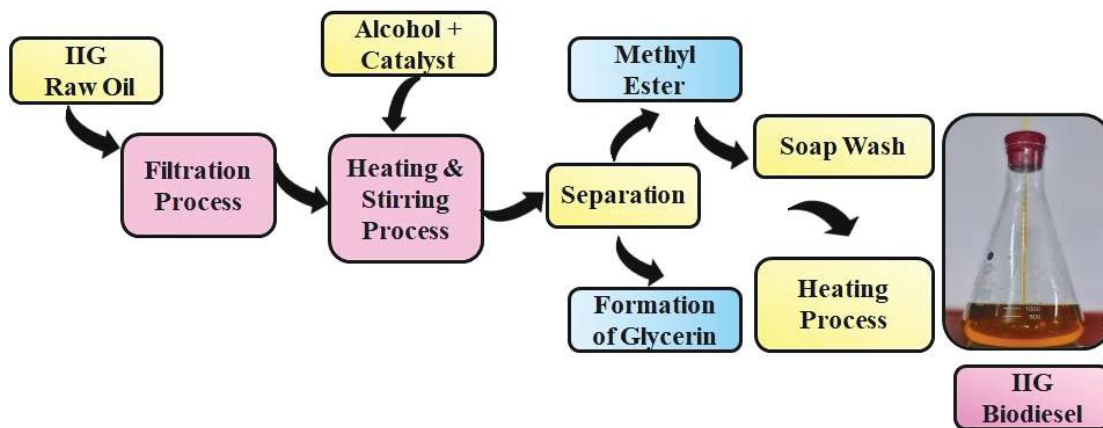


Figure 6. Preparation of second-generation biodiesel using transesterification process.

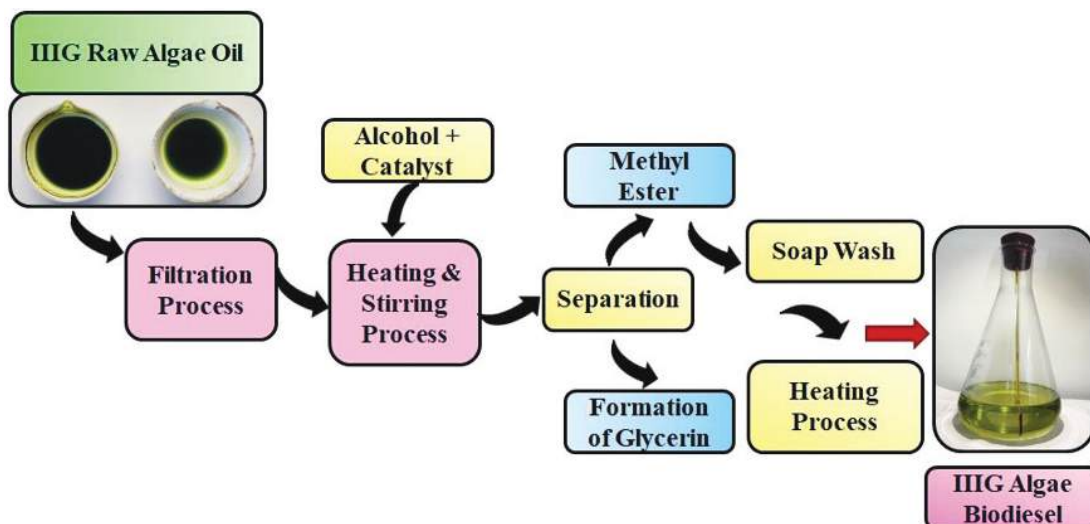


Figure 7. Preparation of third-generation biodiesel using the transesterification process.

Furthermore, If the FFA > 2.5 wt.%, the esterification process follows acidic treatment, i.e., a two-step transesterification process, and if the FFA < 2.5 wt.%, the base treatment is sufficient [42]. The influencing parameters that are used for acid and base treatments are molar ratio (MR), catalyst concentration (CC), reaction time (Time), and reaction temperature (Temp.) [41],[56]. Figure 5 shows the transesterification reaction of triglycerides with alcohol.

D.Singh et al. [41] have extensively assessed several biodiesel feedstocks, production methods, and biodiesel yield estimation. Therefore, a pictorial representation for preparing the biodiesel using the transesterification process for second and third-generation oil is shown in Figures 6 and 7 below [35],[47],[57].

The observations from the review suggested that third-generation (IIIG) algae biodiesel is better than second-generation (IIG) biodiesel because algae possess high

oil and energy content and emit less pollution. Thus, among all biodiesel production methods, transesterification is one of the most effective processes for turning raw oil into biodiesel. Further, the biodiesel was tested for its thermophysical properties and fatty acid compositions using the test apparatus. Therefore, the biodiesel is blended with diesel to meet future energy requirements and estimate the CI engine performance and emission parameters. The flow chart representing the algae processing methods and algae biodiesel as a fuel in diesel engines is depicted in Figure 8.

BIODIESEL BLENDING

Biodiesel of various feedstock obtained using a transesterification process was blended with diesel using a high-speed stirring device and whisked continuously for half an hour (30 mins.) from 10% to 90% with a 10% volumetric

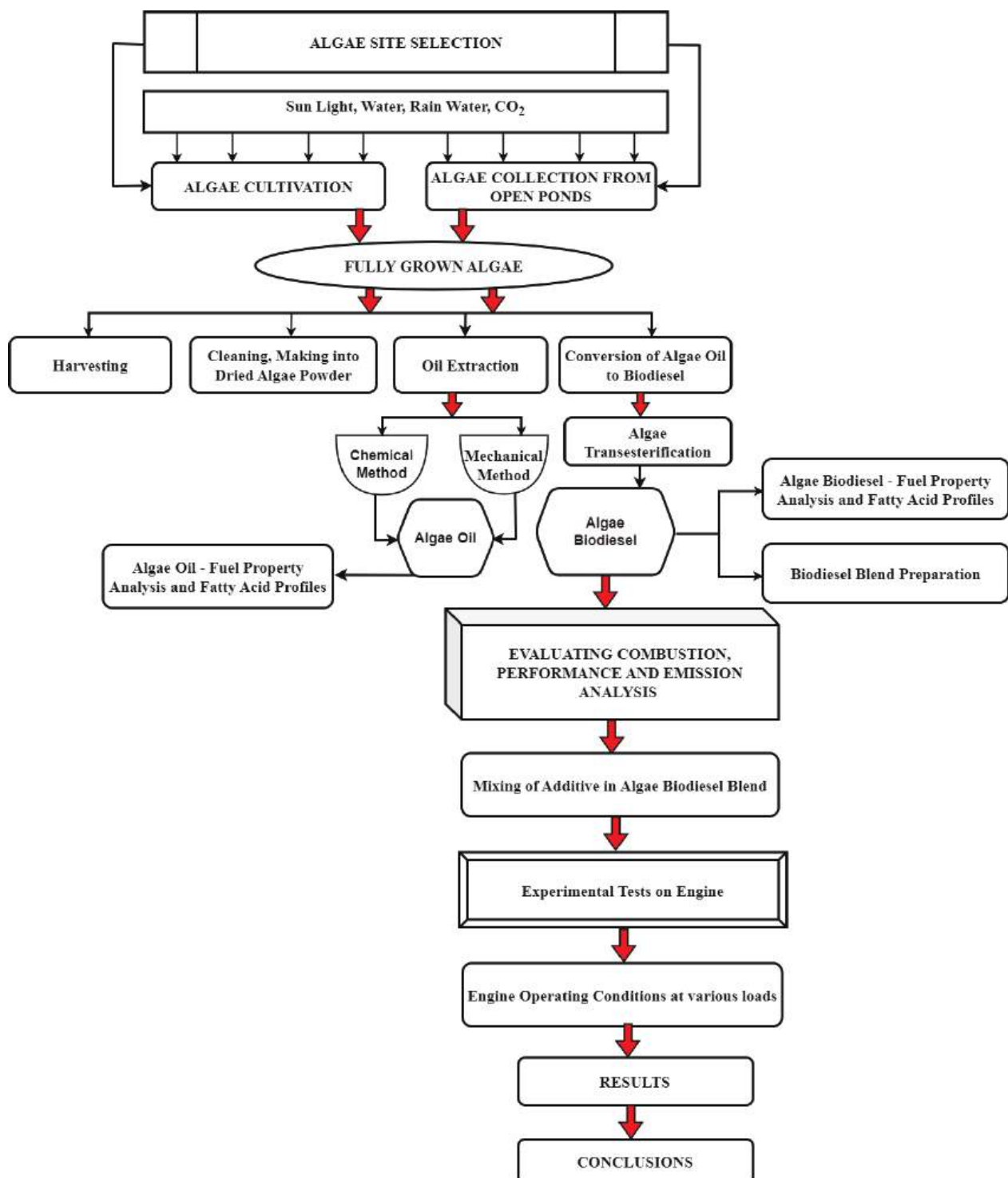


Figure 8. Flow chart representing the algae processing methods and engine experimentation.

increment of B10 to B100 [35]. The pictorial representation of the preparation of blends is shown in Figure 9. Biodiesels have significant benefits, including the technical viability of blending in any ratio and proportion with petroleum diesel fuels. Several investigators continued their research by testing both IIG and IIIG biodiesels for B10 and B20 [48], and

the test results were compared with the established international standards [58], [59], [60].

Several studies on the fuel refinement policies and experimental investigations [19], [61,62,63,64] were conducted on various biodiesel blends (i.e., B5 to B100) to determine the vital fuel properties and fatty acid compositions to

improve the engine performance characteristics. Hence, these draw a significant interest in mitigating the environmental problems to meet the future energy requirements in CI engines.

BIODIESEL FUEL PROPERTIES

One of the significant features of biodiesel is its ability to compete with diesel. These biodiesels are vital in storing, handling, transporting fuel, and commercialization. Studies revealed that fuel properties, such as flash and cloud points [65], are more excellent than diesel, representing a favorable sign to store the fuel. The literature suggests that engine performance depends upon the physicochemical fuel properties, and the vital fuel property values were compared with internationally recognized biodiesel fuel standards (ASTM). Several researchers [18], [65], [66] tested the biodiesel fuel properties of a few IIG and IIIG feedstocks and compared them with diesel are presented in Table 3.

Kinematic Viscosity (mm²/sec)

The kinematic viscosity (KV) of the working fluid plays a crucial part throughout the fuel injection process. Fuels with less viscosity were suggested to achieve improved fuel atomization during combustion and avoid blockage problems associated with fuel injectors. The kinematic viscosity was determined using a Redwood Viscometer according to ASTM D445 for the biodiesel blends. It is evident from Table 3 that the kinematic viscosity value of Spirogyra green algae biodiesel [36] is less than algae, canola, olive,

and cotton seed biodiesel. However, the KV values are higher than diesel KV values and fall within the range of ASTM D6751 standards. This change is due to the large size of triglyceride molecules in oils and the low volatility of biodiesels. Therefore, raw oils with high viscosity are not recommended to run on CI engines as they will cause incomplete combustion and could observe a high rise in engine exhaust emissions. In addition, some researchers tested using neat biodiesel on the CI engine running at a high compression ratio, and the results showed that the engine efficiency was enhanced [21]. Thus, increasing the diesel percentage can reduce the kinematic viscosity of the biodiesel.

Density (kg/m³)

The fuel density (D) is one of the key characteristics that mark its effects on the fuel performance, atomization quality, and combustion. The density was measured using a relative density meter, and ASTM D1298 standards were followed. Like kinematic viscosity, the density is higher for neat biodiesel fuels than diesel. This variation is due to the high molecular weight; however, the limit range is within the standard range. Similar to kinematic viscosity, the density of the neat biodiesel was reduced by adding a high percentage of diesel. The fuel's density influences combustion, atomization quality, and fuel performance, as the CI engines employ the fuel injection pump system to allow the fuel into the combustion chamber. The presence of FFA, H₂O content, and molar mass are the prime determinants of density in methyl or ethyl esters [47], [67].

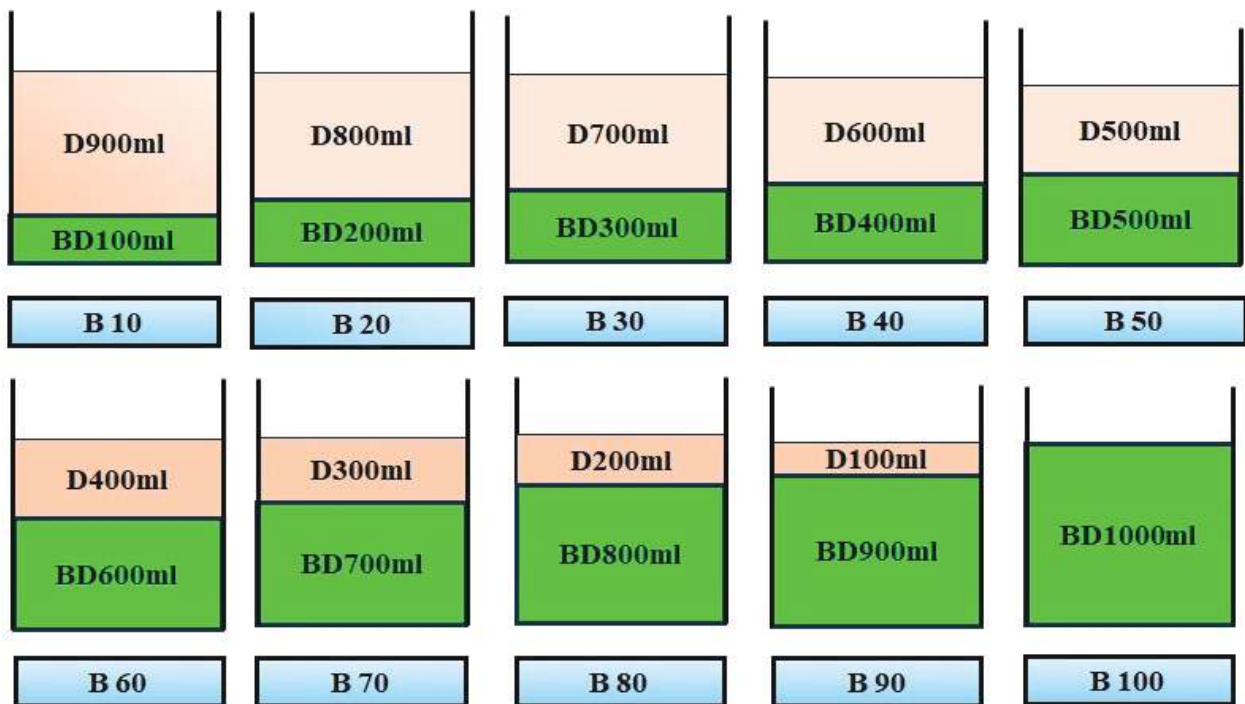


Figure 9. Biodiesel blending with diesel.

Cetane Number (CN)

Fuels with high cetane number (CN) are always preferred in diesel engines to improve combustion and engine performance. The fuel quality and combustion characteristics primarily depend on the cetane value as they can reduce the ignition delay period [66]. It is noted that the CN of biodiesel is higher than that of diesel because of the fatty acid composition of the fuel. An ignition quality tester was used to test the cetane number following ASTM D613 guidelines. The cetane number range for the biodiesel feedstocks was 48-67, depending on many significant factors, such as oil extraction and biodiesel preparation. However, the cetane number in biodiesels increases with the percentage of the blend mixed in diesel. Therefore, to run the engine efficiently and restrict the delay period during combustion, it is endorsed to maintain the cetane number in increasing order; however, in some algae biodiesel, the decrease in cetane number was observed because of the zero Sulphur and significantly less hydrocarbons [66].

Calorific Value (kJ/kg)

One of the most vital fuel characteristics is the calorific value (CV), which determines how much gasoline is used and how much power the engine produces. It provides data on the amount of heat released per fuel unit. Biodiesel from several terrestrial crops and animals has a lower heating value than diesel. Thus, biodiesels might vary due to the chemically attached oxygen (O_2) molecules. In addition, the CV of biodiesel increases when the amount of carbon molecules rises and lowers with the number of double-chain bonds [59]. The calorific value is determined using a Bomb Calorimeter of Make Widson following the ASTM D240 standard.

Flash Point ($^{\circ}C$)

Biodiesels have improved characteristics, such as less volatile and high flash point (FP) temperatures. These are well suited for safe storage and transportation, which are most important for fuel commercialization. Compared to diesel, the flash point of biodiesel is around 150% higher and guarantees a lower risk of a flammability hazard. The flash point for biodiesel was determined using a Pensky Martin closed-cup apparatus that adhered to ASTM D93 requirements [66]. Several aspects, such as the amount of carbon atoms, chemical composition, number of double bonds, and alcohol content, influence the flash point of biodiesel fuel. Most significantly, raising the level of fatty acid saturation in biodiesel fuels may raise their flash point; it is observed from Table 3. that the temperature of flash point for IIIG algae biodiesel and IIG terrestrial crop biodiesel was recorded more than diesel. This change was due to the high rise in carbon from fatty acids in a saturated state. However, this change benefits the storage and safe handling [47].

Pour Point and Cloud Point ($^{\circ}C$)

Particularly at lesser temperatures, the fuel's physico-chemical characteristics may change and can substantially

Table 3. Physicochemical fuel properties of some IIG and IIIG biodiesel feedstocks compared with diesel as per ASTM Standards

Fuel Properties	Test Apparatus [35]	Diesel [4], [68]	Spirogyra Algae Oil BD [35],[37],[63],[47]	Microalgae Oil BD [35],[63],[67]	Algae Oil BD [65]	Waste Cooking Oil (WCO) BD [2],[69]	Palm Oil (PO)BD [66][70][71]	Pongamia BD (PME) [4]	Mahua (MME) BD [4]	Jatropha (JME) BD [4]
Kinematic viscosity at 40 $^{\circ}C$ (mm^2/sec)	Redwood Viscometer	2.97 – 3.96	4.23	2 – 5.2	4.55	4.29	4.12 – 4.54	3.82	4.39	4.21
Density (kg/m^3)	Relative Density Meter	830 – 837	887	850 to 870	881	880	860 – 880	878	873	864
Cetane Number	Ignition Quality Tester	45-60	53.49	37 – 72	59	52	58-65	38559	38293	39455
Calorific Value (kJ/kg)	Bomb Calorimeter	44800 – 45700	41,243	37000 41000		38.08	38050 – 38540	55	52	58
Flash Point ($^{\circ}C$)	Pensky Marten Closed Cup	72-87	146	115	140	170	152	147	130	180
Fire Point ($^{\circ}C$)	Cleveland Apparatus	96	153			178	168			

impact the temperature change, which might be high or low. In biodiesels, the low temperature is observed because of the saturated and unsaturated fatty acid composition (FAC), which plays a major role in this transformation. Also, these biodiesels most likely react to atmospheric conditions. The pour point (PP) and cloud point (CP) are identified as low-temperature fuels and are measured using the PP and CP devices following ASTM D97 for the pour point and ASTM D2500 standards for the cloud point. The cloud point provides data near the lowest temperature at which cooling fuel helps the crystals of wax start to develop. Likewise, the fuel begins to take on gel forms for the pour point and loses its capacity to flow. Hence, the two fuel properties of the biodiesel blends are within the permissible bounds of international biodiesel fuel regulations. Thus, it is clear from various literature studies [36], [37, 47, 60, 63, 65, 67, 68] that the test outcomes of the significant fuel property values of all the diesel-biodiesel blends, mainly algae biodiesel compared to the fossil fuels falls within the acceptable ranges established standards (ASTM) and are suitable to operate in the existing CI engine.

Free Fatty Acids (FFA)

After obtaining the biodiesel, the fatty acid composition is tested using gas chromatography (GCMS) [44] to identify the saturated and unsaturated fatty acids in the biodiesel. The biodiesel fuel properties, such as density, kinematic viscosity, cetane number, and calorific value, significantly impact the FAC. Fevzi Yaşar [65] has comprehensively studied the vital fuel properties and fatty acid contents of 10 different IIG and IIIG biodiesel feedstocks. The test results suggested that cottonseed, olive, and algal

oil had the highest cetane numbers (58 for olive oil and 59 for algae biodiesel) and reported the highest saturated fatty acid composition concentration. The presence of rich oil in saturated fatty acids is more desirable because it will have better thermal efficiency and reduce NO_x emissions.

Saeed A. et al. [37] tested on green algae *S. Elongata*, which grows in freshwater. The experiments were conducted to extract algae oil and produce biodiesel, the catalyst KOH and zeolite. The highest fatty acids identified were oleic, lauric, myristic, and palmitic, and the results suggested that the freshwater green algae *S. Elongata* is a suitable algae feedstock to replace conventional fuels. Therefore, the comparison of the fatty acid composition tests of some second and third-generation biodiesel feedstocks gathered from literature studies is presented in Table 4 and Figure 10 (a – f).

The observations from Table 4 and Figure 10 (a – f), reveal that the unsaturated FAC in biodiesel fuels stimulates the increase in kinematic viscosity and lowers the calorific value. Simultaneously, FAC, with a highly saturated component, controls the increase in NO_x emissions during combustion [72] [73], proving that the third-generation algae biodiesel enhances engine performance and reduces emissions. With this motivation, a comprehensive review was conducted using several IIG and IIIG biodiesel fuels and tested on the CI engine to estimate the engine performance and emission characteristics. Further, studies related to various additives mixed in IIG and IIIG diesel – biodiesel blends were compared, and the findings were presented in Tables 5 & 6.

Table 4. Comparison of fatty acid composition for different second and third-generation biodiesels

Fatty Acid & Structure	Status	Spirogyra Algae Oil BD wt.% [35]	Algae Oil BD wt.% [66]	Palm Oil (PO) BD wt.% [66] [71]	Pongamia Oil (PME) BD wt.% [4]	Mahua Oil (MME) BD wt.% [4]	Jatropha Oil (JME) BD wt.% [4]
Behenic acid (C22:0)	Saturated	1.39	0.33				
Arachidic acid (C20:0)	Saturated	1.28	2.24	0.97	0.8	0.66	0.2
Stearic acid (C18:0)	Saturated	4.86	2.1	3.13	10.9	18.9	7.1
Palmitic acid (C16:0)	Saturated	28.63	15.64	37.48	14.1	21.53	15.2
Myristic acid (C14:0)	Saturated	15.29	-	1.97	0.1	0.08	0.1
Lauric acid (C12:0)	Saturated	21.9	-	0.79	-	-	-
Capric acid (C10:0)	Saturated	0.3	-	-	-	-	-
Caprylic acid (C8:0)	Saturated	0.3	-	0.39	-	-	-
Linoleic acid (C18:2)	Poly-unsaturated	2.5	19.56	9.17	17.3	19.55	32.7
Oleic acid (C18:1)	Mono-unsaturated	21.62	54.89	44.32	53.2	39.1	44.5
Palmioleic acid (C16:1)	Saturated	-	0.32	-	-	-	-
Linolenic acid (C18:3)	Mono-unsaturated	-	4.88	1.78	3.6	0.18	0.1
Others			0.04				0.1
Total Composition		100	100	100	100	100	100

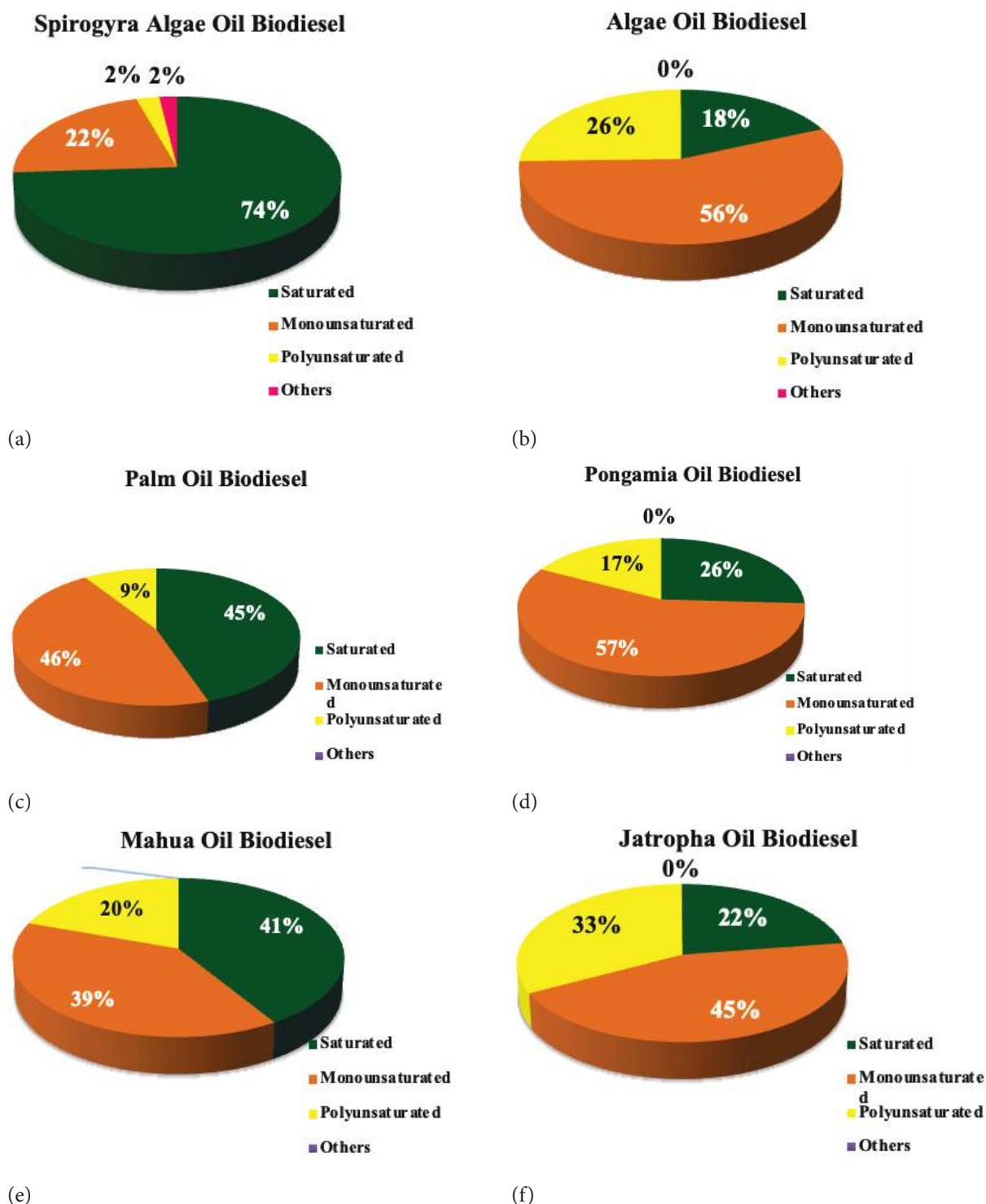


Figure 10 (a – f). Saturated and unsaturated fatty acid composition of different second and third-generation biodiesels.

EVOLUTION OF ALGAE TRANSPORTATION

Algae-derived biofuels can improve the world's transportation fuel and lower global GHG emissions. This renewable energy is expected to switch the world's energy supply by 2070 [74]. The control of emissions plays a pivotal role

when selecting the fuel/biodiesel. Due to increased global environmental pollution, stringent rules were imposed on the transportation industry, and several emission reduction measures such as exhaust gas recirculation [19], mixing of metallic, oxygenated additives [63], [75] mixing of nanoparticles [31], etc., were implemented.

Engine Emissions

The rising concerns about internal combustion engine exhaust emissions are due to their toxic nature. The main byproducts from the combustion of petro-diesel fuels are Unburnt hydrocarbons (UHC), carbon dioxide (CO₂), carbon monoxide (CO), sulfur oxide (SO_x), nitrogen oxide (NO_x), particulate matter (PM), soot and black smoke, which are dangerous to human health and also causes environment air pollution. Some of the human health issues like respiratory problems, cardiovascular diseases, cancer risks, neurological effects, irritation, and allergic reactions are witnessed in patients when they are exposed for a prolonged period to these exhaust gases [63].

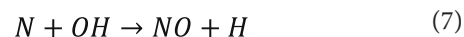
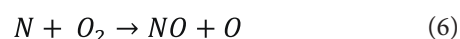
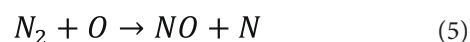
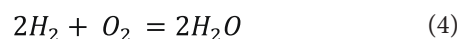
Different factors like fuel properties, air-fuel ratio, injection timing, engine speed, engine load, turbulence, and mixing play a significant role in the combustion of injected fuel and its byproducts of exhaust emissions. For instance, incomplete combustion of fuel will lead to the formation of carbon monoxide emissions, and this incomplete combustion is due to the unavailability of oxygen, poor mixing of air-fuel mixtures, and dissociation effects [76]. The chemical reaction, as shown in Equation 2, is a clear indication that a lower quantity of oxygen participation during the combustion reaction led to CO formation. It is also known that diesel engine combustion is heterogeneous combustion [77] and involves both liquid and gas phases. The hydrocarbon molecules react with available oxygen during the combustion process as a result of the formation of CO₂ and water (H₂O), which is also considered complete combustion, and the chemical reaction is shown in Equation 3 & 4.

The unburnt hydrocarbons are formed due to the incomplete oxidation process. Sometimes, the fuel does not receive enough oxygen to oxidize completely into CO₂ and H₂O during the combustion process; as a result, the unanticipated fuel remains unburnt hydrocarbons [78]. This phenomenon occurs particularly in fuel-rich areas where less air is available. Another significant contributing factor to harmful exhaust emissions is the combustion temperatures. High combustion temperatures result in high thermal NO_x formation, one of the most dangerous exhaust emissions.

The NO_x formation during the combustion process is governed by the *Zeldovich* mechanism [63], which describes the reaction of nitrogen with oxygen at elevated temperatures. At high temperatures (above 1200°C) the nitrogen molecule (N₂) in the air reacts with the available oxygen (O₂), and the chemical reactions are shown in Equations 5 to 7. Different studies [79] show that biodiesel combustion increases NO_x emissions, which may be true due to the inherent property of available oxygen in biodiesel being more than any other fuel, which helps increase combustion temperatures during its combustion. However, this case is applicable to neat biodiesel combustions.

To control the NO_x formation from biodiesel combustion, different techniques like low-temperature combustion, such as injecting the alcohols to maintain controlled

combustion temperatures and also with the implementation of different additives like oxygenated additives, nanoparticles regulate this NO_x formation to a greater extent [31],[64],[76],[81]. The Table 5 reveals the trends of exhaust emissions from biodiesels investigated by different research groups.



From Table 5, the above reference studies observed a positive incline toward the increase in engine performance and reduction in emissions. Interestingly, a drastic improvement in reducing emissions was observed using third-generation algae biodiesels (IIIG). However, an upsurge in NO_x emissions was reported as the signifying issue in most of the experimental tests [47,85,88,89]. Hence, to mitigate this raising concern, mixing additives in biodiesel feedstock is considered an alternate option to mitigate NO_x emissions.

INTRODUCTION TO ADDITIVES

Mixing additives in biodiesels is crucial in increasing the fuel's durability and decreasing the hazardous pollutants produced during fuel combustion. Also, the option to substantiate these issues was additive to avoid limitations of biofuel properties like auto-ignition, flash point, temperature, and cetane number. Additives primarily depend upon the selection, concentration ratio, fuel solubility, and physicochemical properties such as viscosity, density, toxicity, ignition quality, etc.

Various types of additives, such as metallic-based additives, oxygenated additives [25], metal-based additives, cold flow improver additives, cetane number improver additives, water, antioxidants, lubricity improvers, polymeric-based additives, and nanoparticles additives [24,26], are employed to blend with fuels used in diesel engines to achieve total fuel combustion, which further improves the performance of CI engines during combustion and reduce the engine emissions. Therefore, mixing additives in diesel - biodiesel blends result in improvements like complete combustion and temperature reduction, enhancing engine performance and reducing engine emissions [75], [89]. Many researchers have investigated various additives mixed in some IIG and IIIG biodiesel blends and tested them in

Table 5. Review on engine performance and emission characteristics of different second and third generation biodiesel feedstocks (continued)

Ref.	Biodiesel & BlendsEngine Test Condition	Performance Characteristics		Emission Characteristics				
		Performance and emission characteristics compared with diesel						
		BTE	BSFC	NOx	CO	CO2	UHC	Smoke, PM, PN
Kolakoti A.[73]	Palm oil biodiesel (POBD), Waste cooking biodiesel (WCBD), and Animal fat indirect injection biodiesel (AFBD) Diesel	↑ An increase in BTE was achieved for WCBD of (32.22%) followed by AFBF (31.99%) and POBD (30.93%) at 75% load.	↑ Increase in BSFC by 9.72% for POBD, 5.48% for AFBF & 4.90% for WCBD compared to diesel fuel.	↑ Increase in NOx by 2.52% for POBD, 17.16% for WCBD, and 8.09% for AFBF compared to diesel fuel at 100% load. And observed even at 75% and 50%	↑ Increase in CO ₂ by 11.23% for WCBD, 13.28% for AFBF, and decrease in CO ₂ for POBD	↓ Decrease in CO At 100% load 50% reduction is observed for POBD and 63.63% for WCBD and 58.18% for AFBF	↓ Decrease in UHC by 40.57% for POBD, 10.14% for WCBD and 37.68% for AFBF at 100% load compared with diesel fuel	↓ Decrease in smoke at 100% load by 38.18% for POBD, 63.63% for WCBD and 58.18% for AFBF
CA Rinaldini et al.,[81]	Lombardini, 4-cylinder, 4-stroke, naturally aspirated IDI engine, CR: 22.8:1 Bore: 75 mm Stroke: 77.6 mm	↓ Slight reduction (5-7%) of torque at high speed, full load	-----	↑ Increase up to 20% at high load, low speed (N<1500 rpm)	-----	↑ Increase up to 10% at full load, mid-low speed.	↓ Decrease in soot at high load (about 30%)	-----
Chand SM et al., [82]	Water-cooled 1-cylinder, 4-stroke CI engine; Loads: 0, 20, 40, 60, 80 and 100 N= 1500 rpm and CR: 17.5.	↓ Decrease in BTE at 100% load ↓ Decrease in BTE by 26.87% at B25 than B20.	↓ Decrease in BSFC by 0.29 kg/kw at B20 blend than all blends	↓ Decrease in NOx emission levels 1282 for B20, which is 5.78% higher than diesel. Compared to diesel, B20 emits 5.75% less rise.	↓ Decrease in CO for B20 Compared to diesel, B20 emits 8.33% less	↑ Increase in CO ₂ for B20. Compared to diesel, B20 emits 0.86% less	↑ Increase in UHC for B20 blend Compared to diesel, B20 emits 5.55% less.	-----
Rangasamy K et al., [83]	Stationary single-cylinder diesel engine. emersonni methyl ester (CEME) Blends: B10, B20, B30 B100 and D100	Compared to diesel, B30 exhibited better results than other blends.	Compared to diesel, B30 exhibited better results than other blends.	Compared to diesel, B30 exhibited better results than other blends.	Compared to diesel, B30 exhibited better results than other blends.	Compared to diesel, B30 exhibited better results than other blends.	Compared to diesel, B30 exhibited better results than other blends.	Compared to diesel, B30 exhibited better results than other blends.
		↑ Increase in BTE	↓ Decrease in BSFC	↑ Increase in NOx for B100, B30, B20	↓ Decrease in CO by 50%	-----	↓Decrease in UHC by 24%	↓ Decrease in smoke by 56%

Table 5. Review on engine performance and emission characteristics of different second and third generation biodiesel feedstocks (continued)

Ref.	Biodiesel & BlendsEngine Test Condition	Performance Characteristics		Emission Characteristics					
		Performance and emission characteristics compared with diesel							
		BTE	BSFC	NOx	CO	CO2	UHC	Smoke, PM, PN	
Bradley D Wahlen et al., [84]	Chaetoceros gracilis (microalgae strain), a yeast (Cryptococcus curvatus), and a bacteria (Rhodococcus opacus) Blends: B100	Kubota Z482-ES04 naturally aspirated, 2 cylinders, indirect injection Power: 7.9 kW	Compared with soybean oil						
		↓ Decrease in power output by about 93% for microalgae biodiesel, whereas 96% for soybean oil	↑ Increase in BSFC for microalgae	↓ Decrease in NOx	↓ Decrease in CO	↓ Decrease in CO ₂	↓ Decrease in HC	↓ Decrease in power output by about 93% for microalgae biodiesel, whereas 96% for soybean oil	
SS Satputaley et al., [85]	Microalgae Chlorella vulgaris Blends: Algae oil Algae Biodiesel (B 100) Diesel	Kirloskar TV1 model single cylinder, DI, water-cooled, CI engine Operated at 17.5 CR, 1500 fixed speed Power at 5.2 kW Variable load	↓ Decrease in BTE of 5.2% and 6.4% @ 5.15 kW brake power (BP) for Algae oil and Algae Biodiesel (B 100)	↑ Increase in BSFC for Algae oil and Algae Biodiesel (B 100) compared with diesel at all loading conditions.	↓ Decrease in NOx by 19 ppm at 5.15 kW BP for Algae oil.	↓ Decrease in both Algae oil and Algae Biodiesel (B 100)	↓ Decrease in both Algae oil and Algae Biodiesel (B 100)	↓ Decrease in both Algae oil and Algae Biodiesel (B 100)	
Upendra Rajak et al., [86]	Spirulina microalgae biodiesel Blends: AB100	DI, CI engine Full load	↓ Decrease in BTE by 2.73 %	↑ Increase in SFC by 6.4%	↓ Decrease in NOx by 0.5%	↓ Decrease in CO ₂ by 6.1%	-----	↓ Decrease in PM by 60%	
G Ospina et al., [87]	Jojoba, Algae, and chocolate waste methyl esters	VCR, IDI engine.	-----	-----	↓ A decrease in pollution levels was identified using Algae methyl ester. &				
								A drastic ↑ increase in emissions was recorded in jojoba methyl ester. (Tested in two phases: First Set : N, CR, IT, P = constant Second Set varied IT (20° to 45°) bTDC; keeping N, CR, Loads = constant).	

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Second Set varied IT (20° to 45°) bTDC, keeping N, CR, Loads = constant).

Table 6. Review on engine performance and emission characteristics of mixing additives in different second and third generation biodiesel feedstocks

Ref.	Biodiesel + Additive & Mixing of Blends	Engine Test Conditions	Performance Characteristics				Emission Characteristics		
			Additive mixed in biodiesel and compared with diesel						
			BTE	BSFC	NOx	CO	CO ₂	UHC	Smoke, PM, PN
N Yilmaz and A Atmanli [29]	Waste oil biodiesel+ 1-pentanol	Subaru RGD 3300H, one cylinder, air cooled engine, at 0,1.5,3 kW loads, with speed 2000 rpm	↓ A decrease in BTE by 5.33% was observed for D70B20Pen10 compared with diesel	↑ An increase in BSFC by 20.85% was observed at 1.5 kW for D70B20Pen10 compared with diesel	↑ Increase by 30.22%, 36.87% and 29.13% for D75B20Pen5, D75B20Pen10 and D75B20Pen20 as compared to D80B20	↑ Increase by 6.44%, 18.93% and 29.34%, for D75B20Pen5, D75B20Pen10 and D75B20Pen20 compared to D80B20	-----	↑ Increase by 49.06%, 22.59% and 116.01% for D75B20Pen5, D75B20Pen10 and D75B20Pen20 compared to D80B20	-----
	D80B20								
	D75B20Pen5, D70B20Pen10								
	D60B20Pen20								
Yesilyurt MK et al., [30]	Sunflower biodiesel + Pentanol	Operated on four loads at 3000 rpm fixed speed.	↓ Decrease in BTE by 22.75%, 21.82%, and 20.96% for diesel fuel, B20, and B20P5 for all blends at 750W	↑ Increase in BSFC up to 13.90% compared to diesel	↓ Decrease in NOx by 3.16%, 11.85%, 21.58%, and 31.44%, for B20P5, B20P10, B20P15, and B20P20 compared to B20 fuel blend	↓ Decrease in CO by 0.19%, 5.66%, 11.57%, 26.81%, and 31.61%, for B20, B20P5, B20P10, B20P15, and B20P20 compared to diesel	↑ Increase in CO ₂ by 13.15 vol% for B20P20 fuel was 38.72% higher than diesel fuel and 21.87% higher than B20 fuel blend	↓ Decrease in HC, between 1.51% and 4.48% for B20 compared to diesel	↓ Decrease in smoke between 1.20% and 3.40% for B20P5, between 3.19% and 6.11% for B20P10, between 4.92% 7.75% for B20P15, and between 6.25% and 8.15% for B20P20 compared to diesel
Sekharraj K et al., [40]	Green Microalgae Biodiesel + Bi ₂ O ₃ Nanoparticles	Kirloskar TV – I, Power: 5.2 kW, Speed 1500 rpm, CR 17.5:1	↑ Increase in BTE for B20 with Bi2O3	↓ Decrease in BSFC for B20 + Bi ₂ O ₃	↑ Increase in NOx for B20 + Bi ₂ O ₃	↓ Decrease in CO for B20 + Bi ₂ O ₃	-----	↓ Decrease in UHC for B20 + Bi ₂ O ₃	↓ Decrease in smoke for B20 + Bi ₂ O ₃
	Blends:	Mechanical injection system	B20 with Bi2O3	↑ Increase in BSFC for diesel		↑ Increase in CO for Diesel		↑ Increase in UHC for Diesel	
	B20								
	B20 + Bi ₂ O ₃ 25ppm,								
	B20+ Bi ₂ O ₃ 50ppm,								
	B20+ Bi ₂ O ₃ 75ppm, and B20+ Bi ₂ O ₃ 100ppm								
	Diesel								

Table 6. Review on engine performance and emission characteristics of mixing additives in different second and third generation biodiesel feedstocks (continued)

Ref.	Biodiesel + Additive & Mixing of Blends	Engine Test Conditions	Performance Characteristics				Emission Characteristics			
			Additive mixed in biodiesel and compared with diesel							
			BTE	BSFC	NOx	CO	CO ₂	UHC	Smoke, PM, PN	
B Aisosa Oni et al.,[90]	Algae Botryococcus Braunii + Acetylene Induced at a mass flow rate of 100, 150, 200, and 300 g/hr.	4-stroke (Kirkoskar AV1), dual fuel CI engine with enhanced IT	↑ Increase in BTE by 3, 4, 3.6 and 5%	-----	↑ Increase in NOx	↓ Decrease in CO	↓ Decrease in CO ₂	↓ Decrease in UHC	-----	
Gökhan Tüccar,et al.,[91]	Microalgae + Butanol Blends: Diesel D80B20 D70B20But10 D60B20But20	Mitsubishi Canter 4D34-2A, direct injection, 4stroke, 4-cylinder diesel engine. Power 89 kW 1200 - 2800 rpm at full-load	With the addition of butanol for all the blends							
			↓ Decrease in Power output with the addition of butanol	↑ Increase in BSFC about 10.9%, 17.1% and 29.7% for D80B20 D70B20But10 and D60B20But20 compared with diesel fuel	↓ Decrease in NOx with the addition of butanol for all the blends	↓ Decrease in CO with the addition of butanol for all the blends	-----	All test fuels are less than diesel fuel D80B20 reduced by 3.1% D70B20But10 by 27.3% and D60B20But20 by 49.5%	↓ Decrease in smoke with the addition of butanol for all the blends	
Ali Zare et al.,[92]	Waste cooking biodiesel + Triacetin Blends: D100, B100, T100, T4B96, T8B92 T10B90 D60B35T5	6-cylinder turbocharged diesel engine Power 162 kW Speed 2000 rpm CR17.3:1 Friction Power	↑ Increase in BTE for BD100 compared to diesel.	↑ Increase in BSFC, BTE	↑ Increase in NOx by 31%.	↑ Increase in CO up to 82% at higher loads	↓ Decrease in CO ₂ up to 2.5 %	↓ Decrease in HC up to 64%	↓ Decrease in PM and PN both up to 90%	
John Kennedy Mwangi et al., [93]	Chlorella vulgaris and Chlorella Sorokiniana + Butanol & Water Addition Blends: B2, B2But20 and B2But20W0.5 (water addition)	Mitsubishi 4M40-2AT1, 4-cylinder, direct injection CR 20.1:1 Power 109 kWh at 3700 rpm	↑ Increase in BTE for B2- But20-W0.5 compared to diesel.	↑ Increase in BSFC for B2- But20-W0.5 compared to diesel	↓ Decrease in NOx by 25.0% and 28.2% for B2But20 and B2But20W0.5	↑ Increase in CO for B2- But20-W0.5 compared to diesel.	↓ Decrease in CO ₂	↑ Increase in HC by 18.9% and 69.8% with water addition and ↓ Decrease by 50.2% for B2 blend compared to diesel	↓ Decrease in PM by 22.0%, 57.2%, and 59.5% - for B2, B2-But20 and B2-But20-W0.5 compared to diesel	

Table 6. Review on engine performance and emission characteristics of mixing additives in different second and third generation biodiesel feedstocks (continued)

Ref.	Biodiesel + Additive & Mixing of Blends	Engine Test Conditions	Performance Characteristics						Emission Characteristics			
			Additive mixed in biodiesel and compared with diesel						Additive mixed in biodiesel and compared with diesel			
			BTE	BSFC	NOx	CO	CO ₂	UHC	Smoke, PM, PN			
Talamala V et al., [94]	Rice bran methyl ester (RBME) + Isopropanol as an additive. Blends: 2/98%, 3/97%, 4/96% and 5/95%, by volume	IDI engine test rig, Bajaj RE100 diesel engine with an engine displacement of 447.3 cc operated at varying loads	↑ Increase in BTE by 4.3% at 2% additive at maximum engine-operated load	↓ Decrease in BSFC by 2.63% at 2% additive for 2.7kW load	↓ Decrease in NOx by 36.5% reductions were observed at 2% additive at maximum engine-operated load	↓ Decrease in CO by 14% reductions were observed at 2% additive at maximum engine-operated load	↑ Increase in CO ₂ with the increase in additive in percentage.	↓ Decrease in HC for 2% additive at maximum engine-operated load	↓ Decrease in smoke by 27.5% reductions were observed at 2% additive at maximum engine-operated load			
Ilipilla M et al., [95]	TiO ₂ nano particles + dispersant (QPAN80) Blends: BD20+ TiO ₂ at 50, 75, 100 mg/L+ dispersant (QPAN80) 1:1 ratio.	Diesel engine Injection pressures of 200, 225, and 250 bar	↑ Increase in BTE	↑ Increase in BSFC	↓ Decrease in NOx by 846 ppm	↓ Decrease in CO by 0.02%	-----	↓ Decrease in HC by 33 ppm	↓ Decrease in PM by 37.58%	The experimental reports revealed that BD20+ TiO ₂ at 75 mg/L+ dispersant (QPAN80) @ 250 bar pressure exhibited superior stability compared to the other combinations.		

diesel engines to analyze the engine performance and emission characteristics.

In this approach, [28] tested the engine using waste oil biodiesel mixed with various volume proportions of 1-pentanol additive. The ternary blends prepared for testing were DB, D75B20Pen5, D70B20Pen10, and D60B20Pen20, and conducted experiments at 0.1, 0.5, 3 kW load conditions running at 2000 rpm. Therefore, the results suggested that the increase in additive ratio resulted in the rise of brake-specific fuel consumption (BSFC), exhaust gas temperature (EGT), carbon monoxide (CO), unburned hydrocarbons (UHC), and nitrogen oxides (NOx) emissions, where a slight reduction in brake thermal efficiency (BTE) and combustion efficiency was observed. Thus, a minimum quantity of additive mixing in biodiesel is preferable.

In a similar case [30], the sunflower biodiesel was mixed with pentanol to estimate the engine characteristics. The blends prepared for testing are B20, B20P, B20P10, B20P15, and B20P20, and they are operated on four loads at a 3000-rpm fixed speed. The results showed a decrease in BTE and an increase in BSFC of 13.90% compared to diesel. Also, the increase in pentanol addition observed a reduction in NOx and an increase in CO₂ emissions, and the heat release rate and ignition delay were lowered by 15%. Therefore, the study summarized that if the alcohol percentage (pentanol) increases, there can be an improvement in engine behavior.

Another researcher [80] mixed Al₂O₃ nano additive 50ppm and 100ppm in waste cooking oil and neem biodiesel with diesel and tested for the blends B10 and B20. The experimental investigation was carried out on a single-cylinder, four-stroke diesel engine at a fixed speed of 1500rpm. The observed results confirmed that 50ppm in the B10 blend improves the combustion with high exergetic efficiency and low exergy destruction. Also, at 100% load, the blend B20 with 50 ppm of useful work increases by 33.1% more than other blends. Also, the study quoted that at 100% load, the B10+ 50 ppm Al₂O₃ nano additive-observed high exergetic efficiency of 38.69%, enhanced engine combustion performance, and engine emissions such as UHC, CO, and NOx reduced after the addition of nano additives. In spite of all the changes with nano additives, a similar approach was observed with the oxygenated additive. Some researchers investigated oxygenated additives in IIG and IIIG biodiesel, which is continued in the discussion below.

An experimental investigation was conducted using the oxygenated additive Triacetin (T) mixed in coconut oil methyl ester (COME) [75] for the determination of the engine characteristics. Diesel, COME, and COME mixed with Triacetin in the vol.% of 5 %, 10%, 15 %, 20%, and 25% were tested on the DI engine without any change. The output results reported that exhaust emissions lowered positively, near the blend of 10% triacetin mixed with 90% COME. Correspondingly, better engine performance results of BSFC and BTE were achieved with the oxygenated additive mixed in the biodiesel blend. Therefore, the

study concluded that adding oxygenated additives to biodiesel resulted in a favorable agreement to replace with conventional fuels.

In a similar approach, Kalyani et al. [63] mixed oxygenated additive triacetin in an algae biodiesel blend (ABD40) to estimate the engine combustion, performance, and emission characteristics. The oxygenated additive Triacetin was mixed in ABD40 at various percentages of 2%, 4%, 6%, and 8% by volume and conducted experimental tests on a cylinder, water-cooled, four-stroke diesel engine operated at constant speed and varying loads. The observations confirmed that at full operating load (100%), it was revealed that at 4% Triacetin in ABD40 achieved high combustion pressure compared to diesel and other blends. and a high BTE of 30.39%. Also, at 100% load, lower engine exhaust emissions of CO (25.51 %), NO_x (12.81 %), UHC (70.53 %), and smoke (14.85 %) were recorded for the 4 % triacetin in ABD40, recorded with low exhaust emissions compared to diesel. Thus, the investigation proved that the addition of oxygenated additive Triacetin in algae biodiesel enhanced the engine performance and emissions. A detailed literature review conducted by enormous researchers on the IIG and IIIG biodiesel mixed with additives is presented in Table 6 below.

CONCLUSION

The studies contributed by the enormous number of researchers on various second and third-generation biodiesel feedstocks, fuel properties, fatty acid composition, and mixing of additives in diesel-biodiesel of various blends suggested that third-generation green microalgae biodiesel is one of the best sources of green fuel to enhance engine performance and mitigate the emissions in compression ignition engine.

The findings revealed that:

- Third-generation green microalgae biodiesel is an emerging renewable energy source, as these species can be cultivated in open and closed ponds.
- Besides natural and artificial cultivation, the green algae can be collected from rivers, lakes, and reservoirs as these species double every 3-4 hours.
- The microalgae species have higher oil content and can extract algae in large quantities.
- The solvent oil extraction process is the best-suited method to extract algae oil compared to other extraction techniques. Though the extraction process of the algae cellular components consumes more time, it can recover 40 – 50% of the algae oil.
- The cultivation of algae requires less land space compared to second-generation biodiesel. Though second-generation biodiesels play a significant role in improving efficiency and reducing emissions, the prevailing limiting factors, such as the availability of land to cultivate the non-edible oil crops and high fatty acid profiles, led to opt for third-generation algae biodiesel

as one of the better options for replacing conventional fuels in compression ignition engines.

- Fatty acid composition in green algae biodiesel was reported with high saturated fatty acid composition percentage of 74%wt.% followed by second-generation palm oil biodiesel with 46%wt.%. Thus, this observation indicated that third-generation biodiesel plays a significant role in controlling the generation of NO_x emissions during combustion.
- The engine experimentations using second- and third-generation biodiesel stated that the third-generation algae biodiesel operated in diesel engines and observed enhancement in engine combustion, performance, and efficiency. These improvements reduced engine emissions such as carbon monoxide, carbon dioxide, unburnt hydrocarbons, smoke, and particulate matter, and a slight increase in NO_x emissions was observed.
- Mixing of additives in second and third-generation biodiesel improved the vital fuel properties, regulated the combustion issues formed inside the engine cylinder, and controlled the escalation in nitrogen oxide emissions.
- Reduction in maintenance cost is the foremost important aspect of microalgae biodiesel, making them commercially viable.
- Finally, the review studies elucidated that third-generation algae biodiesel is one of the emerging sources of fuel to run and test on existing diesel engines. Thus, algae biodiesel is considered one of the best alternatives to replace fossil fuels, and it holds great potential as a feedstock for future economically sustainable production.

NOMENCLATURE

IG	First-Generation Biodiesel
IIG	Second-Generation Biodiesel
IIIG	Third-Generation Biodiesel
IVG	Fourth-Generation Biodiesel
BD	Biodiesel
BSFC	Brake Specific Fuel Consumption
BMEP	Brake Mean Effective Pressure
CI	Compression Ignition
CR	Compression ratio
DI	Direct Injection
FAC	Fatty Acid Composition
GHG	Green House Gas
IDI	Indirect Direct Injection
IT	Injection Timing
IMEP	Indicated Mean Effective Pressure
VCR	Variable 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.

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