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Research Article

Optimization of exergetic performance and payback period of organic rankine cycle integrated vapor compression refrigeration system based on exergy, economic, and environmental criteria

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ABSTRACT

In this paper, a multi-objective optimization study is carried out to determine the optimal exergetic performance (η_{ex}) and the payback period (PB) of the organic Rankine cycle (ORC) integrated vapor compression refrigeration (VCR) system based on exergy, economic, and environmental criteria using non-dominated sort genetic algorithm-II (NSGA-II). Moreover, a sensitivity analysis is carried out to improve the exergetic performance of the system. The working fluid, the pinch-point temperature difference of the ORC evaporator (ΔT^{p}_{oe}), condenser (ΔT^{p}_{cond}), and the VCR evaporator (ΔT^{p}_{ve}), cooling water inlet temperature (T_{cw1}), chilled fluid temperature (T_{cf2}), and heat-source inlet temperature (T_{h1}) are taken as the decision variables. The results of this study indicate that butane is the most suitable working fluid for optimal exergetic efficiency, η_{ex} (33.7%) and payback period, PB (4.9 years) of the system. The optimal values of other decision variables such as T_{h1}, T_{cw1}, T_{cf2}, ΔT^{p}_{oe}, ΔT^{p}_{ve} , and ΔT^{p}_{cond} are 378 K, 313 K, 276 K, 10 K, 5 K, and 5 K respectively.

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INTRODUCTION

Electricity will be vital for the fastest growing economy like India in the fulfillment of its development goals in the 21st century. A significant fraction of the total electrical energy is consumed by the refrigeration sector. As per a report, India's demand for cooling energy will double by 2027 [1]. The vapor compression refrigeration (VCR) system is extensively used in refrigerators, chillers, air conditioners, etc. due to its higher energy efficiency rating and its adaptability to a wider range of refrigerants [2,3]. However, it requires mechanical power to run its compressor that is provided by the rotating shaft of the electric motor. The electric motor operates on the consumption of electricity. Therefore, there is a need to design and manufacture other devices that can provide the required mechanical energy without consuming electrical energy. Recently, the organic Rankine cycle (ORC) has gained attention among the scientific community to produce mechanical power from

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low-grade heat sources such as solar thermal energy, industrial waste heat, internal-combustion engine exhaust heat, geothermal heat, etc. [4,5]. Researchers have integrated the VCR system with ORC as an ORC-VCR system to have a cogeneration system that can fulfill cooling energy demand without the consumption of fossils. Numerous researches have been carried out on the ORC-VCR system to select the most suitable working fluid for the higher thermal performance of the system and to identify the operating parameters that affect the performance of the system significantly. Li et al. [6] recommended hydrocarbons such as butane, propane, etc. as excellent fluids for the system based on higher COP. Bu et al. [7] proposed the ORC-VCR system as an ice-making unit using waste heat of the fish boats and recommended isobutane as the suitable working fluid for the system. Saleh [8] recommended the use of hydrocarbons with higher critical temperatures as working fluids for higher COP of the system. In the next study, Saleh [8] analyzed the operating parameters such as working fluid, temperatures of condenser, evaporators, etc. to have an impact on the coefficient of performance (COP) of the ORC-VCR system. Zheng et al. [9] analyzed the system in the working range from 268 K to 353 K and achieved the exergetic efficiency of 22.12% using isobutane as the working fluid. Ghorbani, Yari, and Mohammadkhani [10] analyzed the ORC-VCR system driven by the waste exhaust heat of the internal combustion engine. They proposed R245fa and butane as suitable working fluids for the higher thermal performance of the system. They further determined the maximum refrigeration capacity of the system as 20 kW and 130 kW when using waste exhaust heat of petrol and diesel engines respectively. Ashwni et al. [11] analyzed the influence of operating parameters on the system performance based on exergy, economic and environmental criteria. The inlet temperatures of cooling water and chilled fluid outlet temperature were found to be the most influential parameter followed by the heat source inlet temperature. They used the Technique for Order of Preference by Similarity to Ideal Solution (TOPSIS) algorithm to optimize the overall performance of the system. The optimal exergetic efficiency, total cost, and the environmental cost of the system were determined to be 28.4%, $69949, and $3491 respectively. In another study, Ashwni et al. [12] modified the system by adding a flash tank in the VCR sub-cycle to reduce the irreversibility in the expansion process and compared the thermodynamic performance of the simple system with that of the modified system. They found an improvement of 41% in the COP of the modified system over the simple system. Ashwni and Sherwani [13] analyzed the ORC-VCR system driven by a hybrid heat source using different hydrocarbons such as hexane, heptane, octane, etc. as the working fluids. Heptane was found to be the most suitable fluid among the other working fluids for the higher thermal performance of the system. Zhou et al. [14] proposed the system for the cryogenic air separation units and recommended the use of hydrocarbons as the working fluid for the system. Further, they analyzed the impact of operating parameters such as evaporation temperature, condenser temperature, and temperature of compressed air on the economic and environmental performance of the system. They recommended the ORC-VCR system as a promising technology with a great decarbonization potential. Sanaye and Khaakpay [15] utilized the exhaust heat of the gas engine to power the ORC-VCR unit and determined the overall exergetic efficiency of the system at 63%. Mahmoudan et al. [16] analyzed the ORC-VCR system using emergy based method and found that increase in the expander inlet temperature and the isentropic efficiency of the expander had greater improvement in the emergy based performance of the system. However, the emergy-based system performance degraded with the increase in the condenser and VCR evaporator temperature.

Based on the above-mentioned literature review, the following points can be concluded as follows:

- ORC-VCR system is a promising technology that can be driven by low-grade heat sources for refrigeration applications.
- Hydrocarbons are promising fluids for their higher thermodynamic performance in the ORC-VCR system.
- Although the operating temperatures of the condenser and the evaporators have been found to have a significant impact on the ORC-VCR system performance yet no research has been carried out to analyze the impact of their pinch point temperature differences, ΔT^p on the exergy, economic and environmental performance of the system. ΔT^p , which is the minimum temperature difference between two fluids in the heat exchanger, is directly related to the size and eventually to the cost of the heat exchanger. A lower value of ΔT^p is desired for higher exergetic performance of the system but it will also increase the investment cost of the system. On the contrary, a higher value of ΔT^p will result in lower investment cost but it will increase the operating cost of the system due to lower exergetic performance of the system. Therefore, it is imperative to determine the values of ΔT^p of the condenser and the evaporators for the optimal exergetic and economic performance of the system.
- There is a need to determine the payback period of the system not only based on economic criteria but also based on its environmental criteria too.

In this study, exergetic efficiency and the payback period of the system have been considered as the performance characteristics of the system based on exergy, economic and environmental criteria. Heat-source inlet temperature, cooling water inlet temperature, and the chilled fluid outlet temperature directly affect the operating temperature of the ORC evaporator, the condenser, and the VCR evaporator and thus, have been taken as decision variables in addition to the pinch-point temperature difference of the condenser, ORC evaporator, and the VCR evaporator.

Therefore, this study analyzes the impact of the above-mentioned decision variables on the exergetic efficiency and the payback period of the system to determine their operating values using non-dominated sort genetic algorithm-II (NSGA-II) for optimal performance of the system. Moreover, sensitivity analysis of the system is carried out to identify the areas where efforts can be put to improve the exergetic efficiency of the system.

SYSTEM DESCRIPTION

Figures 1 and 2 show the schematic and T-s diagram of the ORC-VCR system. The thermodynamic processes executed by the system are explained below:

Thermodynamic Modeling

The system is assumed to work under steady and adiabatic conditions and pressure losses inside the heat exchangers and the interconnected pipes are taken negligible. The state of the fluid at the exit of evaporators and

Process	Explanation
1 → 2	Cooling effect by isobaric evaporation in the VCR evaporator
2 → 3	Irreversible adiabatic compression in the compressor
3,8 → 4	Isobaric mixing in the mixer
4 → 5	Heat rejection to the cooling water by isobaric condensation
5 → 1	Isenthalpic expansion in the expansion valve
5 → 6	Liquid pumping to the ORC evaporator pressure
6 → 7	Heat gain by isobaric evaporation in the ORC evaporator
7 → 8	Irreversible adiabatic expansion process in the expander

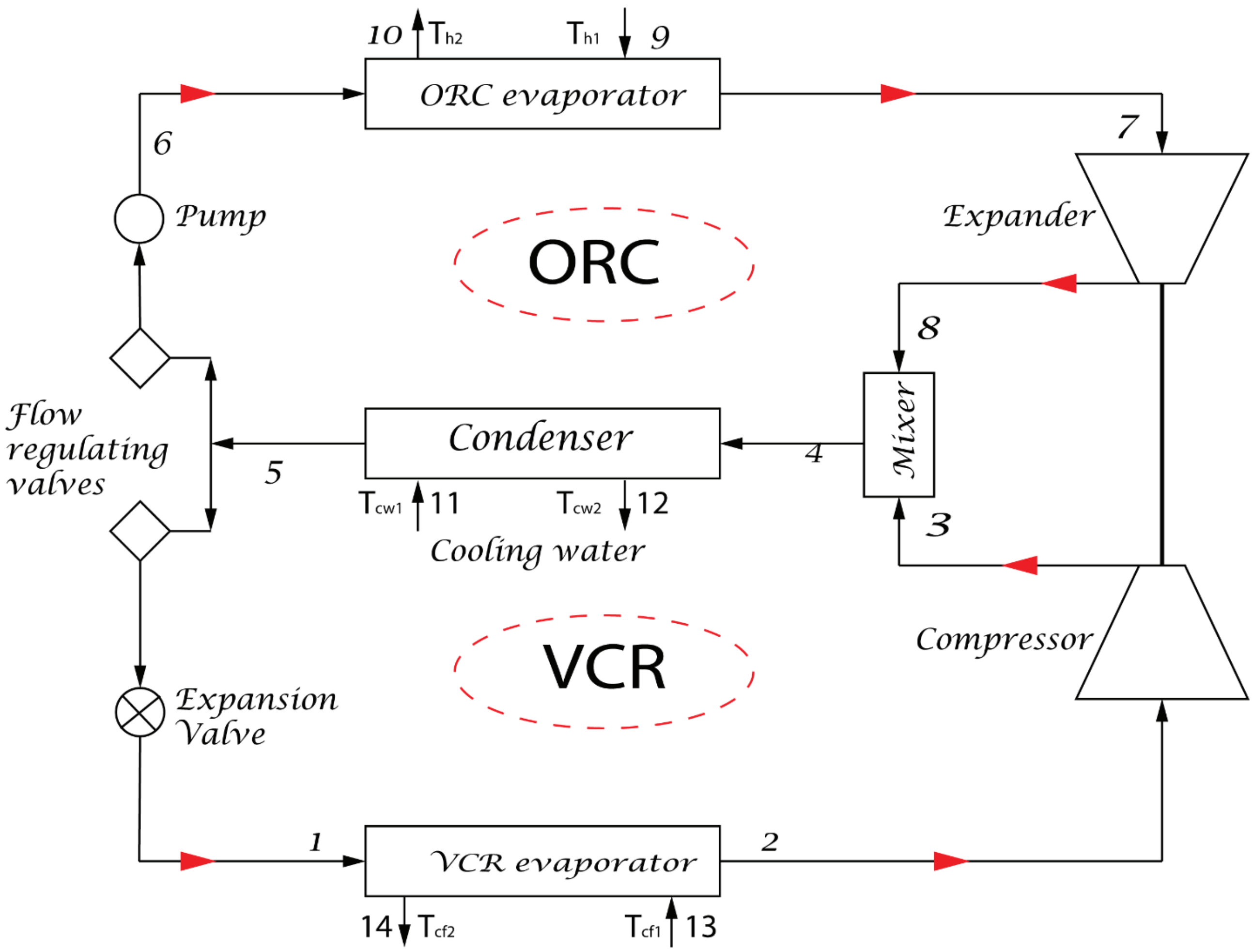

Figure 1. Schematic diagram of the system.

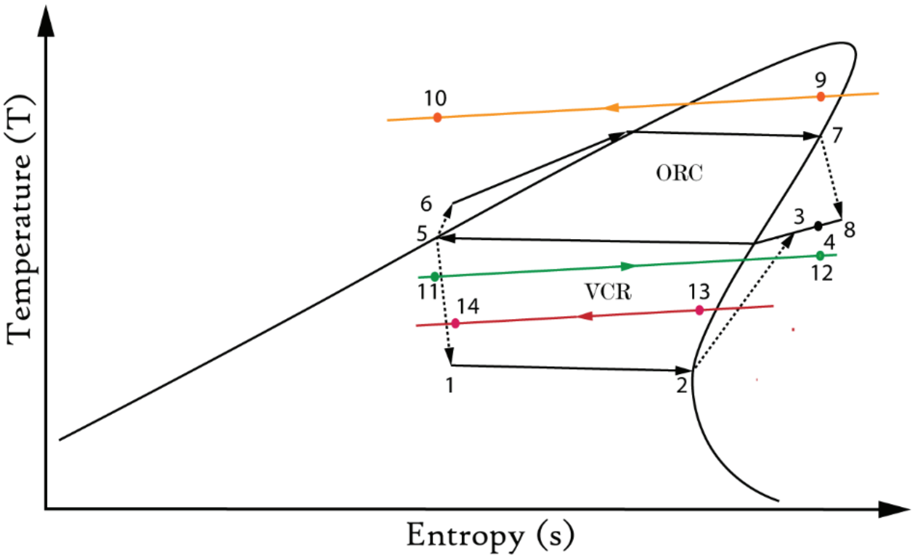

Figure 2. Temperature-Entropy diagram.

the condenser is assumed saturated state [17]. Based on these assumptions, the thermodynamic modeling of the ORC-VCR system is done using basic governing equations such as mass and energy conservation equations, exergy balance relation as given in reference [11]. In this paper, exergetic efficiency (Equation 1) is a measure of the degree of approximation of the actual performance of the system to its best possible performance. Hence, it is used to analyze the second law performance of the system.

$$\eta_{ex} = 1 - \frac{E\dot{X}D_{tot}}{\dot{Q}_{oe}\left\{1 - \frac{T_o}{\frac{T_6 - T_7}{ln\left(\frac{T_6}{T_7}\right)}}\right\}} \quad (1)$$

Where $\dot{Q}_{oe}$ is the heat extracted by fluid in the ORC and VCR evaporators, given by Equation 2.

$$\dot{Q}_{oe} = \dot{m}_{orc}(h_7 - h_6) \quad (2)$$

$E\dot{X}D_{tot} = \sum E\dot{X}D_k \ \forall \ k \in$ mixer, expander, expansion valve, pump, condenser, VCR evaporator, and ORC evaporator.

The exergy destruction rates for individual components can be determined using the Guoy-Stodola equation given below:

$$E\dot{X}D_i = T_o\dot{S}_{gen} \quad (3)$$

Economic Analysis

The economic evaluation of the system (Equation 4) is done by determining its total cost, C_{tot} (Equation 4). The total cost of the system is made up of the two cost components called investment cost, C_{inv} (Equation 5), and operating and maintenance cost, C_{opm}.

$$C_{tot} = C_{inv} + C_{opm} \quad (4)$$

$$C_{inv} = C_p + C_t + C_{ve} + C_{oe} + C_c + C_{cond} \quad (5)$$

The investment cost of the system is the sum of the investment cost of individual system components (refer to Equation 6). The cost equation has been referred to from the research paper [18].

Pump	$C_p = a_3 W_p^{0.71}$ and $\dot{W}_p = \dot{m}_{orc}(h_6 - h_5)$
Expander	$\log_{10} C_t = 2.6259 + 1.4398 \log_{10} \dot{W}_t - 0.1776 \log_{10} \dot{W}_t^2$ $\dot{W}_t = \dot{m}_{orc}(h_7 - h_8)$
VCR evaporator	$C_{ve} = 516.621 + 268.45A_{ve}$
ORC evaporator	$C_{oe} = 516.621 + 268.45A_{oe}$
Compressor	$C_c = \frac{573\ \dot{m}_{vcr}}{0.8996 - \eta_c}\left\{\frac{Pcond}{Poe}\right\} log \frac{Pcond}{Poe}$
Condenser	$C_{cond} = 516.621 + 268.45A_{cond}$ (6)

A_{cond}, A_{oe}, and A_{ve} are the surface areas of the condenser, ORC evaporator, and the VCR evaporator respectively. The detailed procedure to determine the above-mentioned areas is taken from subsection 2.4 of the reference [11]. The operating and maintenance cost of the system, C_{opm} is taken 2% of the investment cost per annum as taken in reference [19].

Environmental Analysis

The impact of the system on its environmental sustainability can be analyzed by calculating its environmental cost using Equation 7 [11].

$$E_{Co_2} = 14.5\, m_{Co_2} \tag{7}$$

m_{CO_2}: Mass of CO_2 (in ton) avoided being emitted in the atmosphere annually because the present system does not depend on fossils for its operation (Equation 8).

$$m_{CO_2} = 2 \times \dot{W}_t \times (\text{no. of operating hours in 1 year}) \tag{8}$$

Payback Period

The payback period of the system is also calculated using Equation 9. Equation 9 is a modified form of that given in reference [20] because it also incorporates the environmental cost in the determination of the payback period.

$$PB = \frac{C_{inv}}{p_e(W_t * t_{op} + E_{Co_2} - C_{opm})} \tag{9}$$

p_e is the unit electricity cost and it varies from 0.077 \$ kWh^{-1} for households to 0.116 \$ kWh^{-1} for industries [20], [21].

t_{op} is the number of operating hours for the system for a year. The literature survey shows that most refrigeration systems are designed for 1000 hours to 7000 hours on annual basis [21]. In this study, the number of operating hours for the system for a year is taken as 6000 hours and the unit electricity cost is taken 0.1 \$ kWh^{-1} at the design condition.

Selection of Working Fluid and the Operating Parameters

In this study, hydrocarbon fluids such as butane, hexane, and heptane are taken as working fluid based on the literature review (refer to introduction). The thermophysical properties such as critical temperature and pressure, environmental properties such as global warming potential and ozone depletion potential, and safety properties of these fluids are given in Table 1.

The present system is designed as a water chiller with a fixed cooling capacity of 66.67 kW. The isentropic efficiencies of the compressor, expander and pump are 80%, 85%, and 90%, respectively based on the recommendation of the reference [12]. The ambient temperature and pressure are taken 298 K and 101.325 kPa respectively. The heat transfer fluid that transfers the heat to the working fluid in the ORC evaporator is pressurized water [22]. The cooling in the condenser is done by cooling water due to its excellent specific heat. The shell and tube-type heat exchangers have been considered in this study [11]. The specification of the heat exchangers is given in Table 2.

CSB Technique

The coefficient of structural bond (CSB) technique is used to identify the components where efforts can be concentrated to improve their design to improve the overall performance of the system significantly. CSB value for each component can be determined using Equation 8.

$$\mathrm{CSB_i} = \left(\frac{\mathrm{dE\dot{X}D_{tot}}}{\mathrm{dE\dot{X}D}_i}\right)_{\mathrm{P_i=constant}} \tag{10}$$

P_i is the operating parameter that affects the performance of the ith component significantly. The components, which have a significant impact on the exergetic efficiency of the system, have CSB values greater than one.

Table 1. Thermophysical, environmental, and safety properties of butane, hexane, and heptane

Fluid	**Butane**	**Hexane**	**Heptane**
T_{cr} (K)	425	507	540
P_{cr} (kPa)	3796	3034	2736
GWP	Low	Low	low
ODP	0	0	0
Safety group	A3	A3	A3

Table 2. Specification of the heat exchangers [23,24]

Length of the tubes	**Tube Material**	**Tube thickness**
6.096 m	Copper	1 mm
Thermal conductivity of copper	**Velocity of tube side fluid**	**Outside diameter**
401 W m^{-2} K^{-1}	2 m s^{-1}	15.87 mm
Number of tube passes	**Condenser and VCR evaporator**	**ORC evaporator**
	4	2

Optimization Technique

In this paper, there are two objectives; one is exergetic efficiency and the other is the payback period of the system. The design efforts are always directed to increase exergetic efficiency and to have a minimum payback period to make the system commercially competitive. There is a need for an optimization algorithm for such types of problems having conflicting objectives. NSGA-II is a multi-objective algorithm that can be used to optimize both the exergetic efficiency and the payback period of the system simultaneously. It involves fewer mathematical complexities compared to another multi-objective algorithm like TOPSIS [11]. The detailed procedure to apply the NSGA-II algorithm can be tracked down in the reference [18]. The mathematical formulation of the optimization problem consists of the following steps:

Step 1: Definition of the objective functions.

$$f_1 = PB \text{ and } f_2 = -\eta_{ex} \tag{11}$$

The objective function that is to be minimized is taken positive. The objective that is be maximized is multiplied by -1 to make it a minimization objective.

Step 2: Selection of decision variables.

T_{h1}, T_{cw1}, T_{cf2}, ΔT^p_{cond}, ΔT^p_{ve}, ΔT^p_{oe} are taken as the decision variables as they have a strong impact on the performance of the system. Moreover, the range of the input values of the operating parameters is selected based on the exhaustive literature available on the ORC-VCR system (refer to the introduction).

Step 3: Specification of the range of the decision variables (Table 3).

Step 4: Specification of the input parameters for optimization

The input parameters for carrying out optimization analysis are tabulated in Table 4.

RESULTS AND DISCUSSIONS

The proposed ORC-VCR system is designed to serve as a water chiller. The cooling capacity of the system is fixed at 66.67 kW. The mathematical code of the thermodynamic model of the system is made in the MATLAB 2017b. The properties of the working fluid at the different state points are calculated with the help of REFPROP 9.0. The work of the paper [25] has been taken as a reference to validate the results of the present study (refer to Table 5). The results of this study are in confirmation of that of the paper [25].

CSB Analysis

The higher the CSB value (CSB>1) of a system component, the greater is the impact of the improvement in the exergetic performance of the component on the overall exergetic performance of the system. Therefore, designers can focus their efforts on those components whose CSB values are higher because a small improvement in the

Table 3. Range of the operating parameters

Operating parameters	Range	Operating parameters	Range
T_{h1} (K)	373-413	**ΔT^p_{oe} (K)**	5-15
T_{cw1} (K)	303-313	**ΔT^p_{cond} (K)**	5-15
T_{cf2} (K)	274-278	**ΔT^p_{ve} (K)**	5-15

Table 4. Input parameters for optimization [18]

Input parameter	Value	Input parameter	Value
Population size	200	**Selection method**	Tournament
Tournament size	2	**Crossover probability**	0.8
Mutation Probability	0.1	**Maximum number of generations**	100* number of variables

Table 5. Validation results

	Working Fluid	η_{ex} (%)	Working Fluid	η_{ex} (%)
Present study	Butane	30.7	Hexane	31.3
Reference [25]	Butane	30.6	Hexane	31.2

exergetic performance of those components will result in a major improvement of the overall exergetic performance of the system. In this study, six operating parameters such as T_{h1}, T_{cw1}, T_{cf2}, ΔT^p_{cond}, ΔT^p_{ve}, ΔT^p_{oe} are taken as the decision variables as they have a strong impact on the performance of the system. These operating parameters are relevant to the condenser and the ORC and VCR evaporators. Therefore, a CSB analysis of only these three components is carried out in this study.

CSB Analysis of ORC Evaporator

Figures 3 and 4 show the variation of the CSB values of the ORC evaporator, CSB_{oe} with heat source inlet temperature, T_{h1} and pinch point temperature in ORC evaporator, ΔT^p_{oe} respectively.

CSB_{oe} values are below 1 for ORC evaporator as T_{h1} varies from 373 K to 413 K. It indicates that any effort to reduce its exergy destruction rate, $E\dot{X}D_{oe}$ will not reduce the total exergy rate, $E\dot{X}D_{tot}$ w.r.t. T_{h1}. However, CSB_{oe} values are slightly above 1 for ΔT^p_{oe}, which indicates that the pinch point temperature of the ORC evaporator is crucial not only for reducing $E\dot{X}D_{oe}$ but also $E\dot{X}D_{tot}$.

CSB Analysis of VCR Evaporator

As the chilled fluid outlet temperature, T_{cf2} varies from 274 K to 278 K, CSB values of the VCR evaporator, CSB_{ve} are well above 35 for all three fluids (refer to Figure 5). However, CSB_{ve} are above 2.90 when ΔT^p_{ve} is varied (refer to Figure 6). It indicates that although the chilled fluid temperature is the most significant parameter yet the importance of ΔT^p_{ve} cannot be ignored in designing the VCR evaporator to improve the exergetic efficiency of the system.

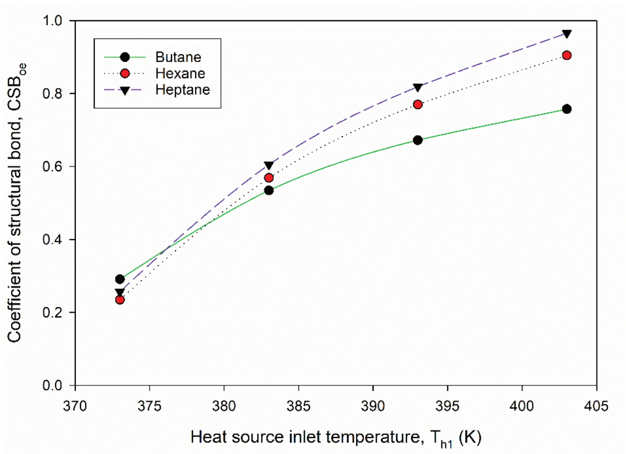

Figure 3. Variation of CSB_{oe} with heat source inlet temperature, T_{h1}

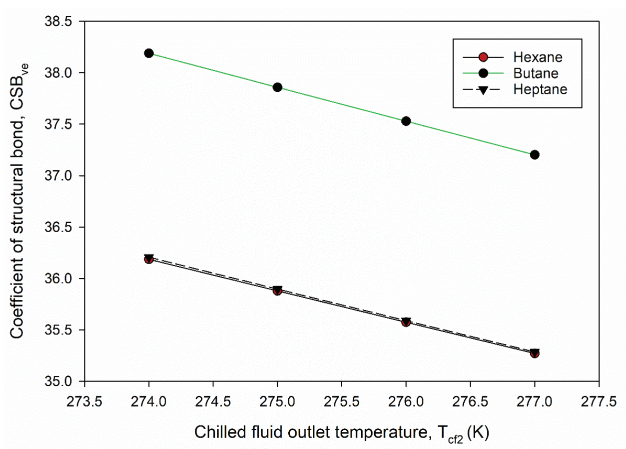

Figure 5. Variation of CSB_{ve} with cold fluid outlet temperature, T_{cf2}

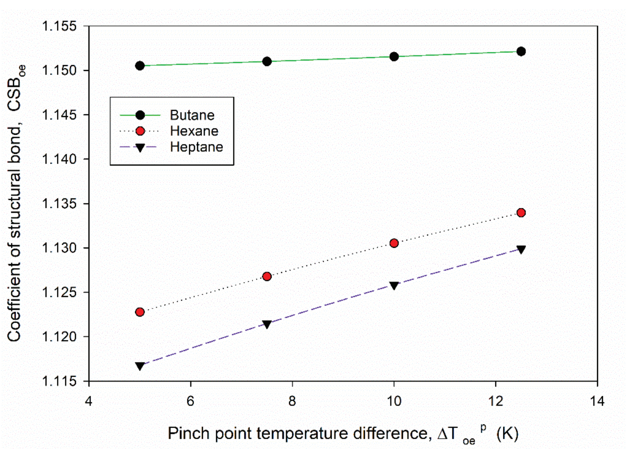

Figure 4. Variation of CSB_{oe} with ORC evaporator's pinch point temperature, ΔT^p_{oe}

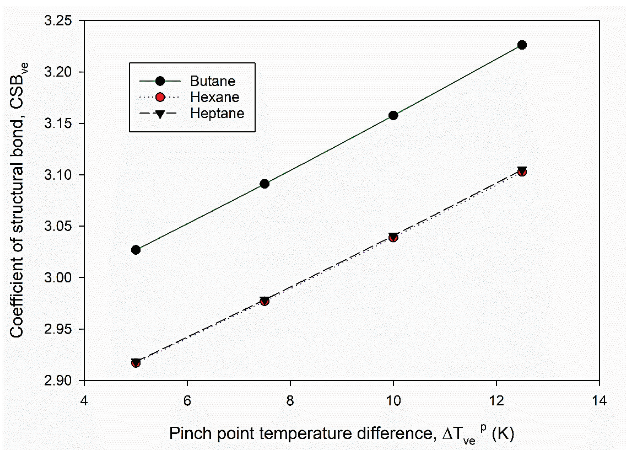

Figure 6. Variation of CSB_{ve} with VCR evaporator's pinch point temperature, ΔT^p_{ve}

CSB Analysis of Condenser

CSB values of the condenser, CSB_{cond} are above 3 as cooling water inlet temperature, T_{cw1} is varied from 303 K to 313 K for all fluids (refer to Figure 7). However, it is slightly above 1 in terms of design parameter ΔT^{p}_{cond}. Therefore, the impact of T_{cw1} is greater than ΔT^{p}_{cond} in designing the condenser to reduce the total exergy destruction rate, $E\dot{X}D_{tot}$ of the system (refer to Figure 8).

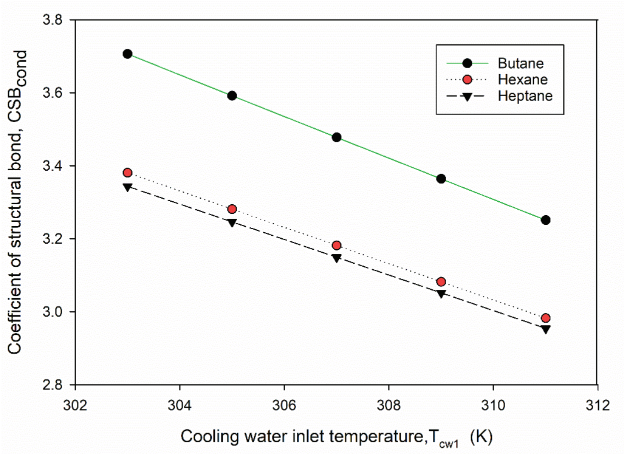

Figure 7. Variation of CSB_{cond} with cooling water inlet temperature, T_{cw1}.

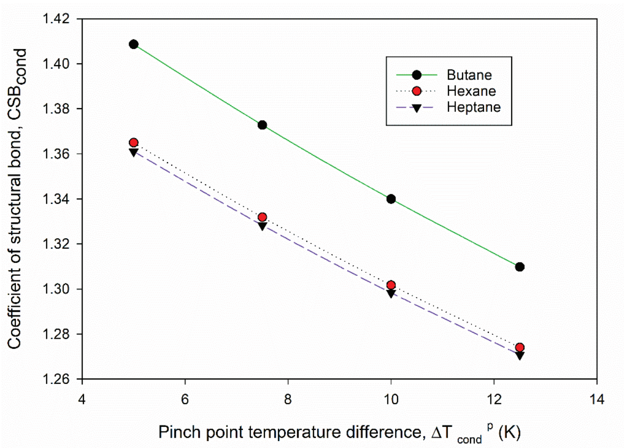

Figure 8. Variation of CSB_{cond} with condenser's pinch point temperature, ΔT^{p}_{cond}.

The CSB analysis of the condenser, VCR evaporator, and ORC evaporator gives us the following insights:

- CSB values are greater than one for all above-mentioned components i.e. condenser, VCR evaporator, and ORC evaporator when the design parameter is pinch point temperature difference. It indicates the importance of the pinch-point temperature difference besides the operating temperatures in designing these components to improve the exergetic efficiency of the system. Any effort in reducing the exergy destruction rates of these components w.r.t. pinch point temperature difference will also reduce the total exergy destruction rate of the system and increase the exergetic efficiency of the system.
- However, it is the VCR evaporator whose CSB values are higher compared to the ORC evaporator and the condenser. The CSB values of the VCR evaporator are above 35 and 2.9 for the design parameters such as chilled fluid outlet temperature, T_{cf2}, and pinch point temperature difference, ΔT^{p}_{ve} respectively. Therefore, a small improvement in the exergetic performance of the VCR evaporator will result in a significant improvement in the overall exergetic performance of the system compared to other system components.

OPTIMIZATION RESULTS

ORC-VCR system with high exergetic efficiency may have higher investment costs and eventually a longer payback period. On the contrary, the ORC-VCR system with a shorter payback period may not have higher exergetic efficiency. Therefore, an optimal solution is desired in which neither the payback period is longer nor exergetic efficiency is smaller. An optimal solution can be obtained by generating a Pareto front. Figures 9 to 11 represent the Pareto fronts for butane, hexane, and heptane respectively. These are normalized Pareto fronts as recommended by [18]. The normalization of the payback period and the exergetic efficiency is done by using Equations 12 and 13.

$$PB^{*} = \frac{PB_{i}-PB_{min}}{PB_{max}-PB_{min}} \tag{12}$$

$$\eta^{*}_{ex} = \frac{\eta^{i}_{ex}-\eta^{min}_{ex}}{\eta^{max}_{ex}-\eta^{min}_{ex}} \tag{13}$$

The optimal solution is the one that is closest to the ideal solution, I [18]. Therefore, it the point O which is closest to point I and thus represents the optimal solution. The x and y coordinate values of point O for butane are 0.136 and 0.192 respectively. The corresponding values of exergetic efficiency and the payback period are 33.7% and 4.9 years respectively. For hexane, the x and y coordinate values of the optimal solution are 0.089 and 0.113 respectively. The corresponding values of payback period and exergetic efficiencies are 38.8% and 14 years respectively.

Similarly, 0.078 and 0.113 are the x and y coordinate values of the optimal solution and 39.4% and 32 years are the values of exergetic efficiency and payback period for the heptane respectively. it can be concluded from this analysis that although the exergetic efficiency of the system using

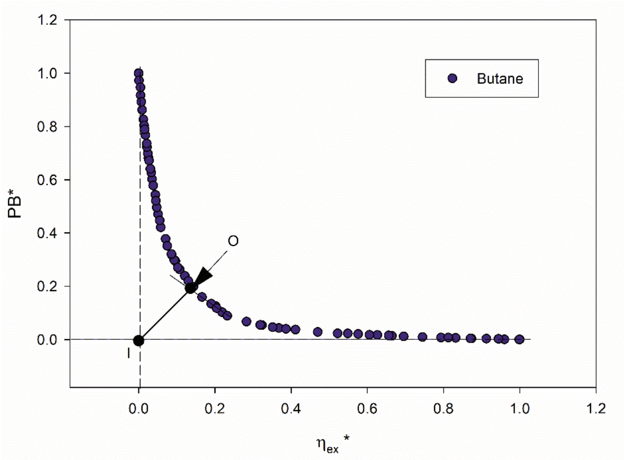

Figure 9. Normalized Pareto front for butane.

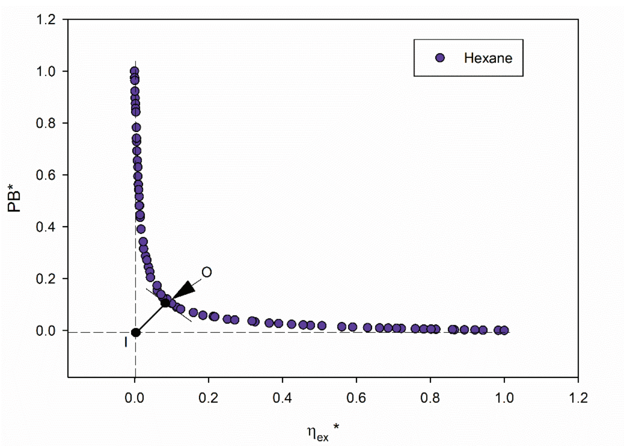

Figure 10. Normalized Pareto front for hexane.

butane is 15.2% and 17.1% smaller than that of hexane and heptane respectively yet its payback period is 187.5% and 556.1% shorter than that of hexane and heptane respectively. Therefore, butane is the most suitable fluid for the proposed ORC-VCR system for higher exergetic efficiency (33.7%) and a shorter payback period (4.9 years). The corresponding values of the operating parameters for the optimal performance of the system using butane are given in Table 6.

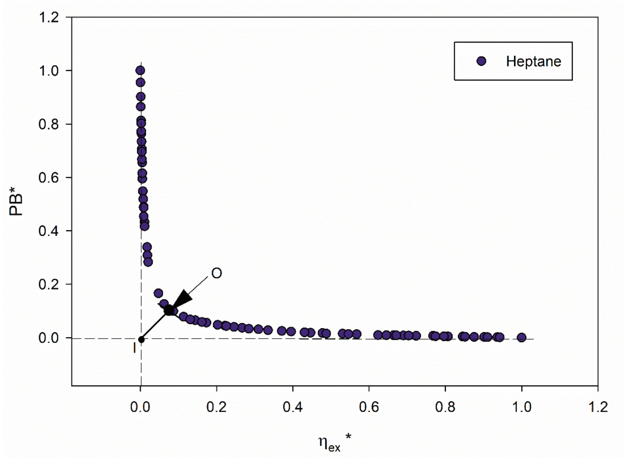

Figure 11. Normalized Pareto front for heptane.

Further, the effect of the operating hours and the unit cost of electricity on the system economics and payback period reveals that as the operating hours are increased from 6000 hours to 7000 hours and the unit cost of electricity is decreased by 0.1 $ kWh^{-1}to 0.077 $ kWh^{-1}; the payback period of the system reduces from 4.9 years to 4.4 years and the total cost of the system is reduced from $62351 to $60936. Therefore, if economic incentives are provided in terms of discounted unit electricity cost for sustainable development and the system is operated for increased operational hours then the cost of the system can be decreased further to make it commercially competitive compared to its commercial counterparts.

CONCLUSIONS

In this study, a low-grade heat source-driven ORC-VCR system is proposed as a water chiller of a fixed capacity of 66.67kW. The impact of six key decision variables on the exergetic performance and payback period of the system is analyzed and the overall system performance is optimized using the NSGA-II algorithm. Moreover, sensitivity analysis is performed to further improve the exergetic performance of the system. The following conclusion can be drawn from this study.

- Sensitivity analysis indicates that pinch point temperature of ORC evaporator, ΔT^{p}_{ORC} is crucial not

Table 6. Optimal values of operating parameters

Operating parameters	Optimal value	Operating parameters	Optimal value
T_{h1} (K)	378	ΔT^{p}_{oe} (K)	10
T_{cw1} (K)	313	ΔT^{p}_{cond} (K)	5
T_{cf2} (K)	276	ΔT^{p}_{ve} (K)	5

only for reducing its exergy destruction but also total exergy destruction of the system, $E\dot{X}D_{tot}$.

- Moreover, chilled fluid temperature, T_{cf2} is the most significant parameter followed by pinch point temperature of VCR evaporator, ΔT_{ve}^{p} in designing the VCR evaporator to improve the exergetic efficiency of the system.
- The impact of cooling water inlet temperature, T_{cw1} is greater than that pinch point temperature of the condenser, ΔT_{cond}^{p} on the exergetic performance of the condenser.
- A small improvement in the exergetic performance of the VCR evaporator will result in a significant improvement in the overall exergetic performance of the system compared to other system components.
- Butane is the most suitable fluid for optimal exergetic efficiency and payback period of the system. The corresponding values of the exergetic efficiency and payback period are 33.7% and 4.9 years respectively.
- The optimal values of the pinch point temperatures for the condenser, ORC evaporator, and the VCR evaporator are 5 K, 10 K, and 5 K respectively.

NOMENCLATURE

A	Heat exchanger area (m^2)
$E\dot{X}D$	Exergy destruction rate (kW)
$\dot{m}$	Mass flow rate (kg s^{-2})
$\dot{Q}$	Heat rate (kW)
T	Temperature (K)
C	Cost ($)
h	Specific enthalpy (kJ kg^{-1})
P	Pressure (kPa)
s	Specific entropy (kJ kg^{-1} K^{-1})
$\dot{W}$	Power (kW)

Greek Symbols

η	Efficiency

Superscripts

p	Pinch point

Subscripts

inv	Investment
oe	ORC evaporator
c	Compressor
gen	Generation
p	Pump
ve	VCR evaporator
ex	Exergetic
opm	Operating and maintenance
cond	Condenser
tot	Total
o	Ambient
t	Expander
cr	Critical

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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