THERMODYNAMIC ANALYSIS OF COMBINED ORC-VCR SYSTEM WITH RECUPERATOR AND REHEATER

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Article history: Received 31 March 2022, Received in revised form 3 April 2022, Accepted 4 May 2022, Available online 4 May 2022.

Highlight
The study evaluates the performance of modified ORC-VCR system for various refrigerants.

Abstract
The trend of utilization of low-grade thermal energy gain huge attention due to increase in energy demand and depletion of conventional resources of energy. Low grade energy can be used in ORC-VCR cycle for refrigeration purpose. In the present work, to improve the performance a modified ORC-VCR cycle, recuperator and reheater are integrated in the cycle. The thermodynamic analysis of the modified system has been conducted with R600a, R600, R290 and R1270 as working fluids under various operating conditions viz. evaporator temperature, condenser temperature, boiler exit temperature. Different parameters evaluated to assess the performance are overall COP, mass flow rate per kW cooling capacity, expansion ratio and compression ratio. From the analysis, butane is found as a best choice for the modified ORC–VCR cycle. It was found that for the modified ORC-VCR cycle at boiler exit temperature of 90°C and condenser temperature 40°C has system COP of 0.5542 with butane, which is 7.1% and 18% higher than that of ORC-VCR cycle with recuperator and simple ORC-VCR cycle, respectively.

Keywords
low grade energy; organic rankine cycle; refrigeration; hydrocarbons; recuperator; reheater.

Introduction
The trend of utilization of renewable energy gain huge attention due to increase in energy demand and depletion of conventional resources of energy. The countries should focusses on sustainable development and to asses it, Khanova et al. [1] proposes a multidimensional Sustainable Development Index. Various renewable resources of energy such as, wind, geothermal, solar and biomass can be used for electricity generation [2–4]. Heat exchangers is one of the main component used in various renewable and non-renewable device and its sustainability was assessed by Kumar et al. [5] for different types of nanoparticles using intuitionistic fuzzy combative distance-based assessment ($IFCODAS$) method and observed that Carbon based nano particles has more reliable and sustainable thermal systems. The waste heat and low-grade thermal energy can also be considered as renewable energy source and utilized in the electricity generation with the help of Organic Ranking Cycle (ORC) [6,7]. Organic Rankine cycle (ORC) is most suitable to recover low-grade thermal energy and can be applied with solar thermal devices, power plant where large amount of waste heat generated. In present context, ORC cycle has market share

of 48% in biomass, 31% in geothermal, 20% in waste heat recovery and 1% solar application [8]. Due to lack of awareness, ORC market share has not much augmented in solar appliances but has a huge potential.

Researchers always try to improve the performance of ORC using either different working fluid or different coupled system. Xu et al [9] presented a novel review on utilization of zeotropic fluids in ORC. They theoretically analyse the performance of different zeotropic fluid and recommended that zeotropic fluid have vast application in ORC. Andreasen et al [10] considered propane, i-butane, i-pentane, R1234yf and their blends as working fluid in ORC and observed that R1234yf is more economical than other working fluids. ORC cycle was coupled with liquid flooded expansion system using R1233zd(E) as working fluid by Li et al [11] in their experimental work. They pointed out that the liquid flooded expansion system improves the isothermal efficiency of expander, resulting in better performance of ORC cycle. An another modification was proposed by Chen et al [12], where they couple ORC with absorption heat pump and reported that coefficient of performance of absorption heat pump (COP) can be obtained from 1.38 to 2.37 depending upon the operating parameters. Recently, Kumar et al [13] utilizes solar flat plate collector for harnessing solar thermal energy as heat source to ORC and studies energetic end exergetic performance of ORC using MWCNT/R141b nano refrigerant. They found enhancement of 8.4% and 6.2% in energetic and exergetic efficiency, respectively.

In the present scenario, massive portion of the generated electricity is consumed by the refrigeration system [14]. Refrigeration effect can also be directly produced by waste thermal energy with the help of vapour absorption and ejector refrigeration system. However, such systems are only suitable when waste heat available in range of 100 to 200^{o}C and can produce refrigeration effect at higher temperature of 5^{o}C or more [15]. ORC-vapour compression refrigeration system (VCR) cycle is another way to produce refrigeration effect from low grade heat in range of 90 to 100^{o}C. In ORC-VCR cycle, waste heat is supplied in ORC to produce power and VCR cycle run by that power [16].

The ORC-VCR cycle consist working fluid with low NBP which is shared by both the cycles. In this system, compressor and expander are directly attached through shaft. The power generated by expander is utilized to run boiler feed pump of ORC and compressor of VCR. So far, several studied have been done on ORC-VCR cycle which reveals the overall performance of cycle is significantly affected by the working fluid. Li et al. [17] investigated ORC-VCR cycle with four hydrocarbons viz. R600, R290, R600a and R1270. Yue et al. [18] investigated ORC-VCR cycle with R134a, n- propane, cyclopentane, and R245fa and found R134a as best candidate. Rawat et al. [19] presented energy analysis of ORC-VCR cycle with hydrocarbons. Saleh [16] has performed the parametric study of ORC-VCR system and concluded R602 is the most suitable among fourteen fluids viz. R602, R601a, R601, R600, R600a, C5F12, RC318, R236fa, R152a, R236ea, R245fa, R245ca, R1234ze(E), RE245cb2. Pektezel et al. [20] proposed ORC-VCR cycle with single and dual evaporator and found that performance of the system decreased with extra evaporator.

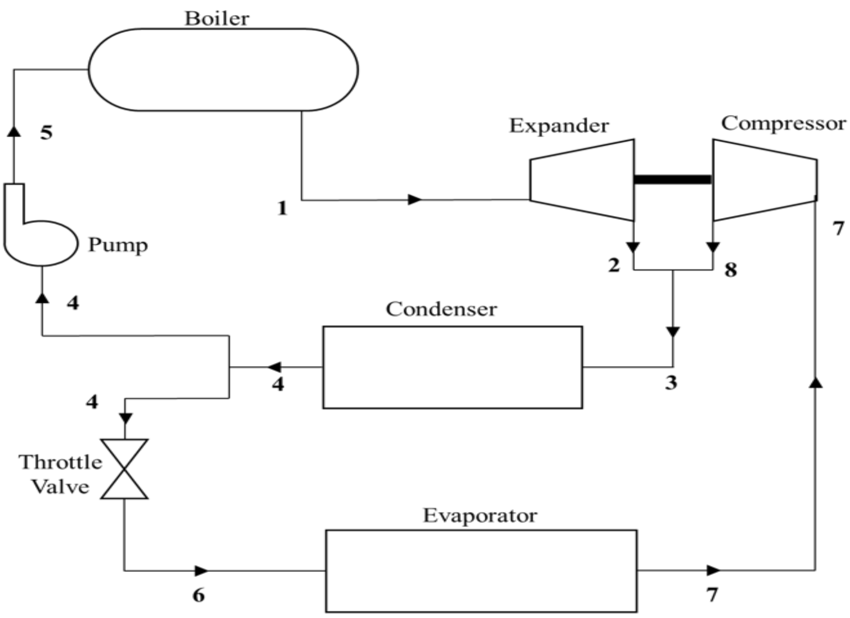

Figure 1. Schematic diagram of ORC-VCR system.

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Nabati and Saadat-Targhi [21] proposed ORC-VCR cycle to produced cold water for hospital using solar energy. They concluded that R22 is more suitable working fluids among R134a, R143a, R11, R22, R500, and R600. More studies had been conducted on ORC-VCR cycle with CFCs, HCFCs, HFCs, HFOs and HCs as working fluids. However, CFC, HCFCs, HFCs have adverse impact on environment. On the other hand, natural refrigerants have excellent thermo-physical properties and environment friendly nature (zero ODP, low GWP) [22,23]. Several researchers also employed various optimization technique to improve the performance of ORC-VCR cycle [24–28]. Present study deals with modified ORC-VCR cycle using low-grade thermal energy. In the modified ORC-VCR cycle, recuperator and reheater are integrated to improve the performance. The thermodynamic analysis of the modified system has been conducted with R600a, R600, R290 and R1270 as working fluids under various operating conditions viz. evaporator temperature, condenser temperature, boiler exit temperature to determine their effect on the performance parameters viz overall COP, mass flow rate per kW cooling capacity, expansion ratio and compression ratio.

System Overview

Schematic illustration of the modified ORC–VCR system with recuperator and reheater is shown in the Figure 2. In system we have two cycles, first is power cycle (1-2-3-4-5-6-7-8-9-1) and the other is refrigeration cycle (7-10-11-12-7) both cycles share same refrigerant as a working fluid. This modified system has a recuperator and two expanders which are operating at same shaft speed and compressor is also directly coupled to this shaft. At outlet of the boiler, working fluid is in vapour phase which enters in first expender and pressure decrease up to intermediate pressure then again it is send to the boiler for reheating. After reheating, working fluid expands into second expander up to condenser pressure. Now working fluid enters in the recuperator which is a counter flow energy recovery heat exchanger. Recuperator is the placed between the second expander exit and the condenser inlet or between the pump exit and boiler inlet for heat recovery. In the recuperator liquid is preheated before entering the boiler by absorbing heat from the working fluid coming out from the second expander.

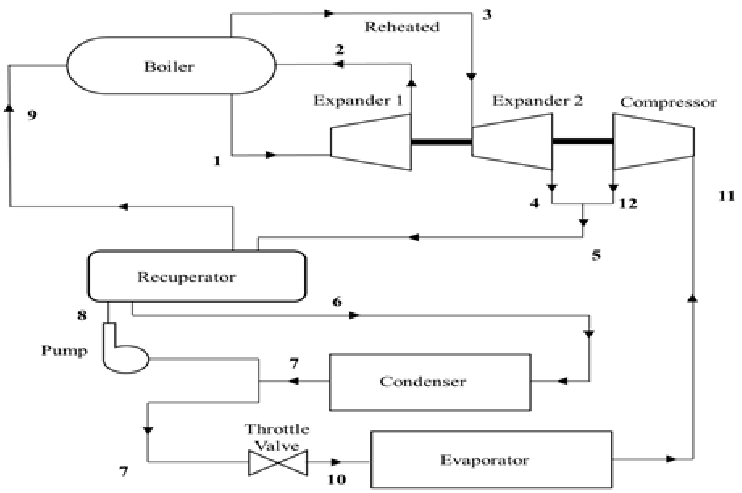

Figure 2. Schematic illustration of modified ORC-VCR system.

The T–S diagram of the modified ORC–VCR system is depicted in Figure 3. The different processes which describe the system are 1→2 actual expansion through the high pressure expander; 2→3 reheating at constant pressure; 3→4: actual expansion through the low pressure expander; 4→5: constant pressure cooling of low pressure expender vapour by mixing with compressor discharge vapor; 5→6: The expander exhaust is used to preheat the working fluid exiting the pump; 6→7: Isobaric heat rejection (condensation); 7→8: actual pumping work; 8→9. The working fluid is preheated by the expander exhaust; 9→1: Isobaric heat addition in the boiler; 7→10: isenthalpic expansion through the throttle valve in the VCR cycle; 10→11: Isobaric heat absorption (evaporation) in the VCR cycle; 11→12: actual compression through the compressor.

https://doi.org/10.32933/ActaInnovations.44.3 • ISSN 2300-5599 •

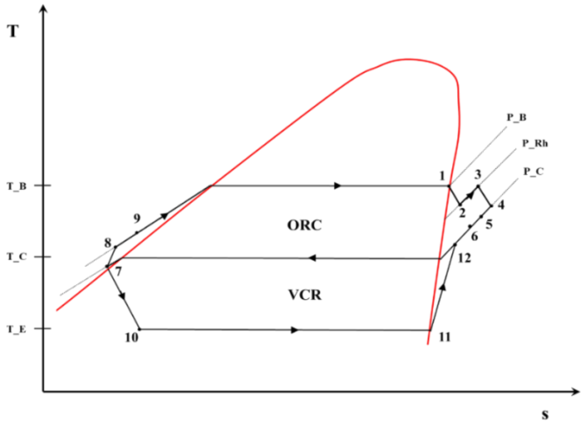

Figure 3. T-S diagram of the modified ORC-VCR System.

Methods

In the present work, thermodynamic analysis of modified ORC-VCR system has been carried by considering all components are in steady-state conditions and there are no heat & friction losses. The thermo-physical properties of the working fluids for different states have been calculated using a software package Energy equation solver (EES). Software EES is a vital tool for thermodynamic calculations of refrigeration and cryogenic systems [18,19].

Work obtained by the first expender

(1) $$\dot{w}_{E1} = \dot{w}_{1-2} = \dot{m}_{ORC}(h_1 - h_{2s})\eta_{exp1}$$

(2) $$\dot{w}_{E1} = \dot{m}_{ORC}(h_1 - h_2)$$

Heat supplied in the reheater

(3) $$\dot{Q}_{Re} = \dot{m}_{ORC}(h_3 - h_2)$$

Work obtained by the second expender

(4) $$\dot{w}_{E2} = \dot{w}_{3-4} = \dot{m}_{ORC}(h_3 - h_{4s})\eta_{exp2}$$

(5) $$\dot{w}_{E2} = \dot{m}_{ORC}(h_3 - h_4)$$

Recuperator

(6) $$(\dot{m}_{ORC} + \dot{m}_{VCR})(h_5 - h_6) = \dot{m}_{ORC}(h_9 - h_8)$$

The effectiveness (ε) is the ratio of the actual to maximum possible heat transfer rates and is expressed as

(7) $$\varepsilon = \frac{T_5 - T_6}{T_5 - T_8}$$

Heat rejected in condenser

(8) $$\dot{Q}_C = (\dot{m}_{ORC} + \dot{m}_{VCR})(h_6 - h_7)$$

Work supplied in Pump

(9) $$\dot{w}_P = \dot{w}_{7-8} = \dot{m}_{ORC}(h_{8s} - h_7)/\eta_{pump}$$
$$\dot{w}_P = \dot{m}_{ORC}(h_8 - h_7)$$

Heat supplied in boiler

(10) $$\dot{Q}_{Boiler} = \dot{m}_{ORC}(h_1 - h_9)$$

Network obtained from the ORC

(11) $$\dot{w}_{net} = \dot{w}_{E1} + \dot{w}_{E2} - \dot{w}_P$$

Total heat supplied to ORC

(12) $$\dot{Q}_{Supply} = \dot{Q}_{Boiler} + \dot{Q}_{Re}$$

Efficiency of ORC

(13) $$\eta_{ORC} = \frac{\dot{w}_{net}}{\dot{Q}_{Supply}}$$

Heat absorbed in the evaporator

(14) $$\dot{Q}_E = \dot{m}_{VCR}(h_{11} - h_{10})$$

Work supplied to the compressor

(15) $$\dot{w}_C = \dot{w}_{11-12} = \dot{m}_{VCR}(h_{12s} - h_{11})/\eta_{comp}$$
$$\dot{w}_C = \dot{m}_{VCR}(h_{12} - h_{11})$$

(16) $$COP_{VCR} = \frac{\dot{Q}_E}{\dot{w}_{net}}$$

Work supplied to the compressor of VCR = Net work obtained from ORC

(17) $$\dot{w}_C = \dot{w}_{net}$$

(18) $$COP_{system} = COP_{VCR} \times \eta_{ORC}$$

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The expansion ratio (EPR) across the expander is proportional to the expander size and has been evaluated by

$$\mathrm{EPR} = \mathrm{V_1}/\mathrm{V_4} \tag{19}$$

While the compression ratio across the compressor is calculated by

$$\mathrm{CMR} = \mathrm{P_{12}}/\mathrm{P_{11}} \tag{20}$$

The input parameters and boundary conditions are tabulated in Table 1. The maximum boiler exit temperature is set as 90°C, which corresponds to the heat source temperature of about 100°C. This temperature level can be easily achieved by flat-plate solar collectors or water-dominated geothermal energy.

Table 1. Input parameters and boundary conditions.

Parameters	**Boundary**	**Range**
Boiler exit temperature (T_b)	80°C	60 - 90°C
Condenser temperature (T_c)	40°C	30 - 55°C
Constant subcooling in condenser	3°C	
Evaporator temperature (T_e)	5°C	-15 - 15°C
High pressure expander isentropic efficiency (η_{exp1})	0.80	-
Low pressure expander isentropic efficiency (η_{exp2})	0.80	-
Compressor isentropic efficiency (η_{comp})	0.80	-
Boiler feed pump isentropic efficiency (η_{pump})	0.75	-
Effectiveness of Recuperator (ε)	0.90	-
Working fluid mass flow rate in ORC ($\dot{m}$)	1kg/s	-
Reheat exit temperature (T_b)	-	60 - 90°C
Reheat pressure	$(P_e * P_c)^{1/2}$	-

Results and Discussion

Figure 4(a) Illustrates the effects of the boiler exit (BE) temperature on the COP_{system}. It can be observed that, COP_{system} of the modified ORC–VCR system increase with BE temperature. The increase in boiler temperature leads to rise in turbine inlet enthalpy. Moreover, to satisfy the constant turbine and compressor power, $\dot{m}_{ORC}$ value decreases. This results in lesser heat input to boiler and increased efficiency of ORC cycle. The increase in value of η_{ORC} enhances the COP_{system}. As the BE temperature increase from 60 to 90°C, the COP_{system} increases by 90.92%, 103.01%, 123.04% and 125.8% for R1270, R290, R600a and R600, respectively. The maximum COP_{system} is reported for R600, owing to the higher critical temperature of this fluid. With the BE temperature at 90°C, the COP_{system} for the R600 case is 0.5542 which is greater than that of the R600a, R290 and R1270 cases by about 6.6%, 26.3% and 37.89, respectively.

Figure 4(b) shows the effect of BE temperature on the $\dot{m}/Q_0$ (flow rate mass per kW refrigeration capacity). $\dot{m}/Q_0$ decreases with the increase in BE temperature for all working fluids as work generated by the expanders increase. The lowermost flow rate is found for R600 and followed by R600a, R290 and R1270. $\dot{m}/Q_0$ is highest for R1270 at 90°C reaching to a value of 0.01869 kg/kW-s. This is due to reduction in $\dot{m}_{ORC}$ as stated in explanation of Figure 4(a).

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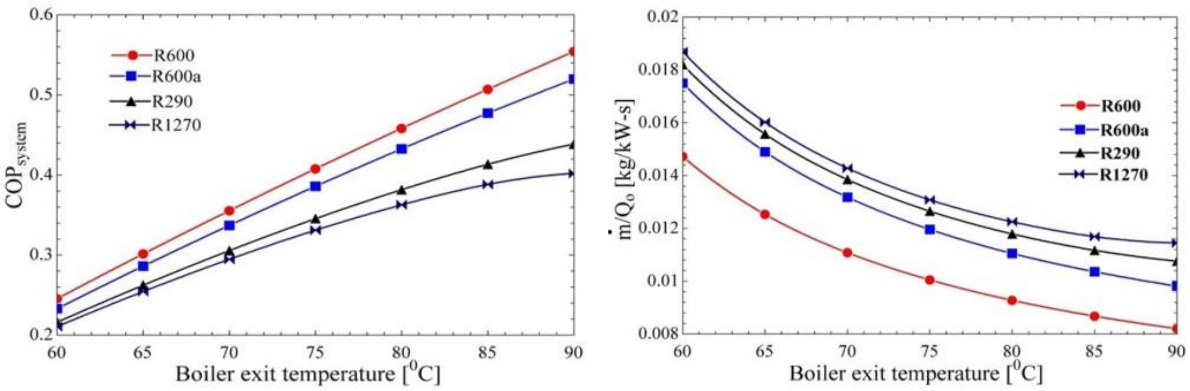


(a) (b)

Figure 4. Variation of (a) $\mathrm{COP_{system}}$, (b) $\dot{m}/_{Q_0}$ with BE temperature.

The impact of BE temperature on the EPR is depicted in Figure 5(a). For increase in BE temperature with fixed condenser temperature, EPR in ORC increases, because with the increase in boiler temperature, saturation pressure also increases. EPR at the BE temperature at 90°C approximately three times that at 60°C for the wet refrigerants viz. R290 and R1270 and two times for dry refrigerants viz. R600 and R600a. The largest EPR is found in the case of R1270, followed by R290, R600, and R600a. However, it should be mentioned that the differences between EPR for these hydrocarbons are initially small, and the maximum appears at about 37.86% between the cycle of R600 and R1270 at boiler exit temperature of 90°C.

Condenser temperature has a major impact on ORC–VCR system performance, as illustrated in the Figure 5(b). In ORC-VCR system, condenser is shared by both cycles, and it also depends on the surrounding temperature. For fixed boiler and evaporator temperature, too high value of condenser temperature is undesirable. Also, it can be deduced from the plot that with increase in condensation temperature, the $\mathrm{COP_{system}}$ decreases. This is attributed to increase in exit enthalpy of turbine. Furthermore, to guarantee the constant turbine power, $\dot{m}_{ORC}$ must increase and η_{ORC} must decrease. On comprising different working fluid, R600 has shown better $\mathrm{COP_{system}}$ than R290, R600a and R1270. The $\mathrm{COP_{system}}$ of the ORC–VCR systems using R1270, R290, R600a and R600 as working fluids gives 0.1367, 0.1446, 0.1675 and 0.1821, respectively, at condensation temperature of 55°C.

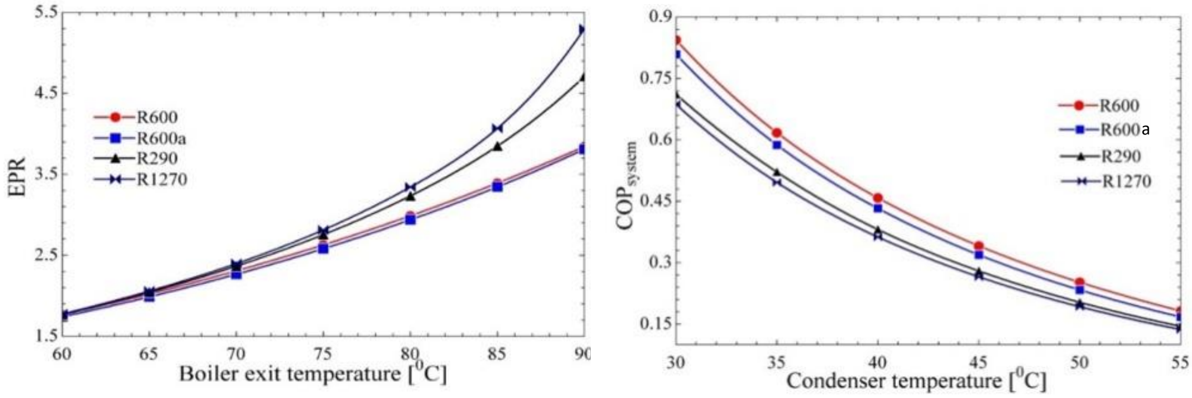
Figure 6(a) shows the variation of mass flow rate per kW cooling capacity with the condenser temperature. Increase in Condenser temperature causes an increase in the total mass flow rate per kW cooling capacity due to increase in $\dot{m}_{ORC}$. Under the same operating conditions, the lowest mass flow rate per kW cooling capacity is found in case of R600 with the value of 0.02113 kg/kW-s, whereas the highest occurs for R1270 reaching to 0.03076 kg/kW-s.

It is deduced from Figure 6(b) that, the EPR in the ORC decreases with increase in the condenser temperature. This is obvious taking when the thermo physical properties of these working fluids into consideration. Th differences among the expander ratio values for the four hydrocarbons are similar for 30 to 55°C condenser temperature. Comparing with the other fluids the decrement in the EPR in the cases of R1270 and R290a are more prominent. It is interesting to report that with the condenser temperature higher than 50°C, the differences among the EPR for dry hydrocarbons viz. R600 and R600a are quite small.

From Figure 7(a), it is observed that CMR value increase with condenser temperature as pressure ratio in the VCR cycle increases. However, the value of CMR significantly increase with condenser temperature for R600 and R600a. This is obvious due to thermo physical properties of the working fluids.

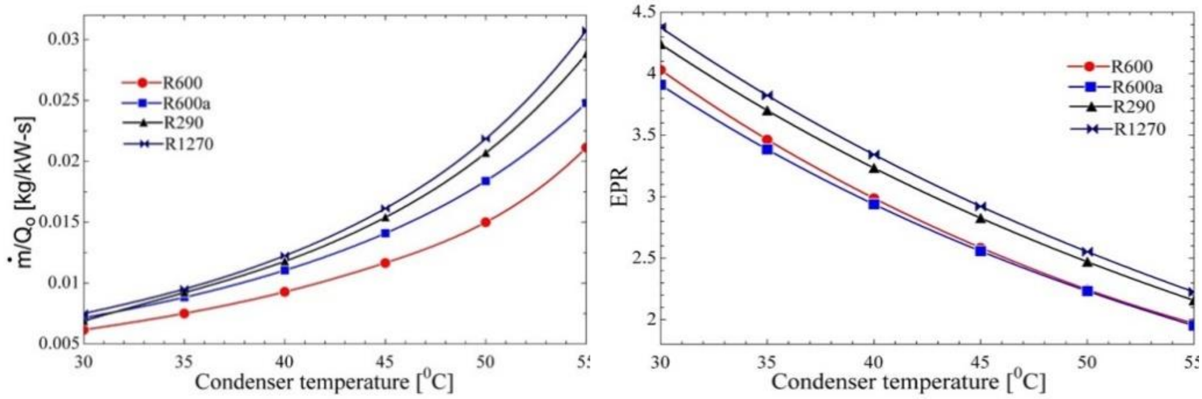
Figure 7(b) shows the effect of the evaporation temperature on the $\mathrm{COP_{system}}$. It can be observed that, as evaporation temperature increases, the $\mathrm{COP_{system}}$ of the ORC–VCR system also increases. The pressure ratio decreases therefore the compressor work decreases when evaporator temperature increases. This increases the COP of VCR system due to which the overall system performance enhances. As the evaporation temperature varies from -15 to 15°C, the $\mathrm{COP_{system}}$ increases by about 179.7% for the four cases. Moreover, under similar operating conditions, R600 case exhibits better performance. For example, with the evaporation temperature at

−15°C, the $\mathrm{COP_{system}}$ of the ORC–VCR system using R600 is approximately 5.1%, 19.76% and 26.36% higher than the cases of R600a, R290 and R1270, respectively.



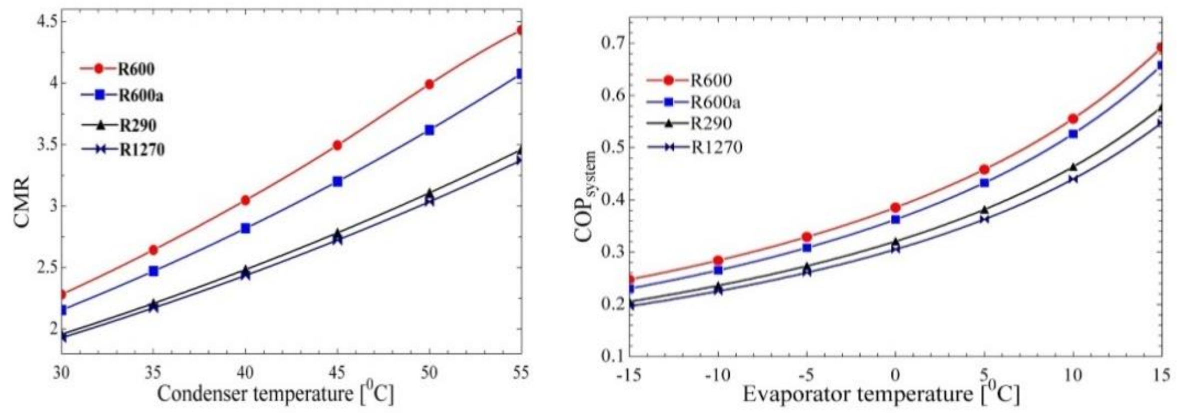
(a) (b)

Figure 5(a) Variation of EPR with BE temperature, (b) Variation of $\mathrm{COP_{system}}$ with condenser temperature.



(a) (b)

Figure 6(a) Variation of condenser temperature with $\dot{m}/_{Q_0}$, (b) Variation of condenser temperature with EPR.

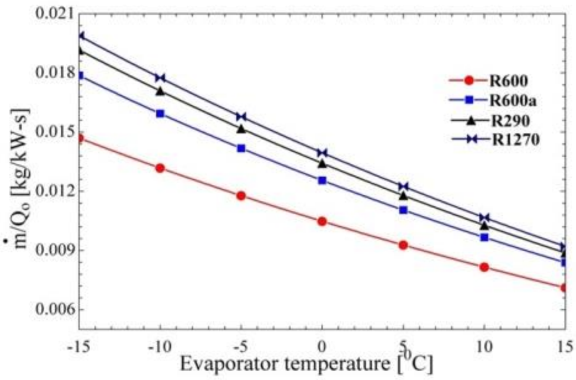


(a) (b)

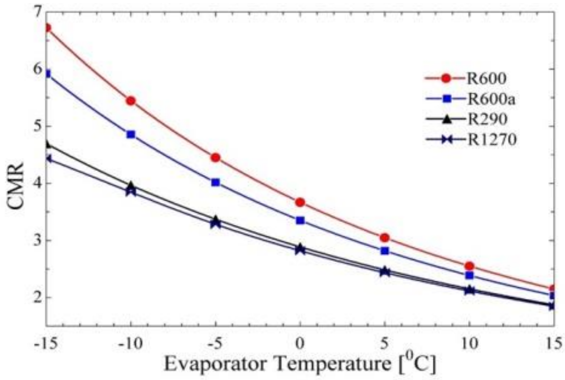
Figure 7(a) Variation of CMR with condenser temperature, (b) Variation of $\mathrm{COP_{system}}$ with evaporator temperature.

Figure 8(a) shows the variation of $\dot{m}/Q_0$ with the evaporator temperature. The value of $\dot{m}/Q_0$ decreases with the increase in evaporator temperature. This is because of decrease in compressor power and turbine power with increase in evaporator temperature. The decrease in turbine power results in decreases in $\dot{m}_{ORC}$. Therefore, $\dot{m}/Q_0$ for combined system also decreases. As the evaporator temperature varies from -15 to 15°C, the $\dot{m}/Q_0$ decreases and it is lowest for butane. Moreover, under similar operating conditions, R1270 a gives highest mass flow rate per kW cooling capacity. $\dot{m}/Q_0$ is about 17.8%, 24.79% and 29.43% lower than that of R600a R290 and R1270, respectively.

Figure 8(b) depicts the variation of evaporator temperature with CMR. As the evaporation temperature varies from -15 to 15°C, CMR decreases, and it is lowest for the propylene. Propane and propylene both are wet fluid refrigerants and their CMR are almost similar. On increasing evaporator temperature at fixed condenser temperature, the pressure ratio decreases, hence the size of compressor reduces.



(a)



(b)

Figure 8(a)Variation of evaporator temperature with $\dot{m}/Q_0$, (b) Variation of CMR with evaporator temperature.

The performance of the modified ORC-VCR system with recuperator and reheater is also compared with simple ORC-VCR system with no additional component and ORC-VCR system with recuperator only in Figure 9. For the modified ORC-VCR cycle, at boiler exit temperature of 90°C and condenser temperature 40°C the system COP with butane is 0.5542 which is 7.1% and 18% higher than that of ORC-VCR cycle with recuperator only and simple ORC-VCR cycle, respectively.

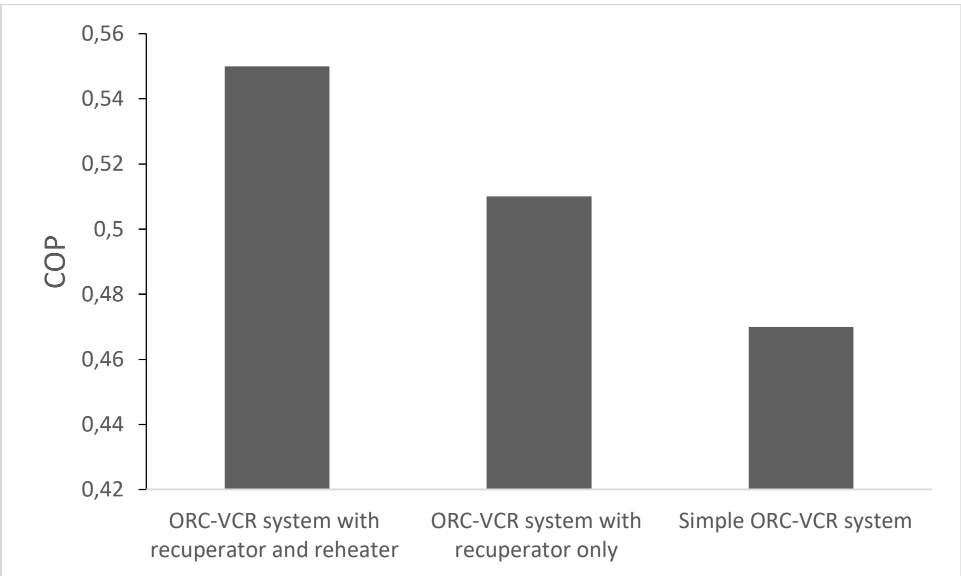


Figure 9 Comparison of COP for different configuration of ORC-VCR System.

https://doi.org/10.32933/ActaInnovations.44.3 • ISSN 2300-5599 •

Conclusions
In the present work, modified ORC-VCR cycle integrated with recuperator and reheater has been investigated using R600a, R600, R290 and R1270 as working fluids. From the analysis, effect of various operating conditions viz. evaporator temperature; condenser temperature; BE temperature on overall COP, mass flow rate per kW cooling capacity, expansion ratio and compression ratio are concluded here. The performance of modified ORC-VCR cycle with recuperator and reheater has shown favourable characteristics for R600 and R600a in comparison to R290 and R1270. Moreover, for same conditions, the performance of the modified ORC-VCR cycle is higher in comparison to simple ORC-VCR system and ORC-VCR cycle with recuperator.

Impact
The increase in demand for energy and depletion of conventional energy resources has led researcher to find either non-conventional energy resources or efficient conventional systems. The non-conventional energy resources have limited potential and still in developing phase. However, conventional system performance can be significantly improved by optimizing the working parameters and reducing the waste energy. The thermodynamic Organic Rankine cycle (ORC) has ability to utilize low grade thermal energy from either renewable sources or waste heat from power plant. ORC has been coupled with different systems like liquid flooded expansion system and absorption heat pump, vapour compression refrigeration system (VCR), and others to augments the performance. The present work focuses on improving the performance of ORC-VCR cycle by firstly integrating the recuperator and an intermediate reheater, and then operating parameters like refrigerant type, boiler exit temperature, and others are also optimized. Working fluids plays a crucial role in the ORC system's efficient functioning particularly in case of waste heat having low-grade temperature. From present study, the Butane as a working fluid in ORC-VCR system has yielded best performance among considered working fluid, so it is recommended for application in ORC-VCR cycles.

Conflict of interest
There are no conflicts to declare.

Funding information
There is no fund involved in this work.

Acknowledgments
This research has not been supported by any external funding.

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