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Research Article

Thermodynamic Efficiency Analysis of a Combined Power and Cooling (ORC-VCRC) System Using Cyclopentane (C_5H_{10}) as a Substitute for Conventional Hydrocarbons

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Abstract

The present study investigation aims to contribute to the field of energy engineering by exploring the performances of cyclopentane gas as promising working fluid in combined power and cooling (ORC–VCRC) system. The present research emphasizes the comparative computation of various thermodynamic performance characteristics of (ORC–VCRC) system activated by low temperature heat sources using cyclopentane gas as a substitute to the conventional hydrocarbons (butane, isobutene, propane and propylene) widely used in (ORC–VCRC) system. A computer code was developed using MATLAB software for the numerical simulation. The performance characteristics computed are the performance indicators (overall coefficient of performance (COP_{oval}) and working fluid mass flow rate of per kW cooling capacity (MkW), expansion ratio in expander (EPR) and compression ratio in compressor (CMR). Furthermore, the effects of different operating parameters (e.g., boiler, condenser, and evaporator temperatures, isentropic efficiency of expander (η_{exp}), and isentropic efficiency of compressor (η_{comp})) on performance indicators are also examined for each working fluid. Results showed that under the same operating parameters, the use of cyclopentane gas as a working fluid in (ORC–VCRC) system exhibited a higher COP_{oval} and lower MkW compared with conventional hydrocarbons. When boiler temperature reaches 90 °C, the COP_{oval} of cyclopentane increase by 14 %, 19.8 %, 43.8 % and 59 % compared to those of butane, isobutene, propane and propylene, respectively. However, the MkW of cyclopentane reduced by 19.1 %, 29.2 %, 44.3 % and 53.7 % compared to same fluids, respectively. On another hand, the study revealed that the COP_{oval} rises as the temperature of the boiler, evaporator, η_{exp} and η_{comp} rises. Conversely, when the condenser temperature rises, the COP_{oval} value falls for all fluids. Overall, the study confirms that cyclopentane gas could be a promising working fluid in terms of performance indicators for (ORC–VCRC) system.

Keywords: *Hybrid (ORC–VCRC) system; cyclopentane gas; thermodynamic analysis; performance indicators.*

1. Introduction

Due to increasing energy and environmental problems, the development of new energy technologies and the exploration of alternative working fluids with environmentally friendly properties have become the most researched areas in energy engineering, particularly in the areas of power and cooling applications.

Today's, hybrid thermodynamic cycles, which combined the energy systems, such as power systems, refrigeration systems, heat pumps and air-conditioning systems are regarded as an efficient way to utilize the medium and low temperature heat sources of the renewable energies (solar heat, biogas, biomass,…, etc) [1-2] for its various advantages (high energy conversion efficiency, low cost, easy maintenance, environmentally friendly, etc.) [3], where the use of this new technologies is growing significantly in the field of building energy.

One of this hybrid thermodynamic cycles that has been gaining attention is the hybrid ORC–VCR system, which combined the organic Rankine cycle (ORC) with the vapor compression refrigeration (VCR) to convert the thermal energy input obtained from renewable energies sources at low and medium temperature into beneficial cooling or electrical power [4-5].

The working fluid selection has a considerable influence on the performance of the hybrid ORC–VCR system. At present, the choice of new working fluid is a challenge for the hybrid power-cooling system, where the good working fluid should be safe and environmentally friendly, should have high overall performances and adapt to the temperature of the available heat source [6-9]. In this context, many studies have been published in recent years using different working fluids on this type of hybrid system to improve the performances of the energy conversion at low temperatures by several researchers.

Saleh [10] proposed hydrofluoroolefins and common hydrofluorocarbons as working fluids for an ORC–VCR system that is powered by low-grade thermal energy. The

ORC–VCR system combines vapor compression refrigeration and the organic Rankine cycle. With a maximum overall performance of 0.718 at a condenser temperature of 30 °C and basic values for the remaining parameters, the results showed that working fluid R600 is the best candidate compared to the other substances suggested for the hybrid (ORC–VCR) system. Still, its flammability ought to draw sufficient notice.

Aphornratana and Sriveerakul [11] assessed the two working fluids, R22 and R134a, to determine which was best for the heat-powered refrigeration cycle and a combined Rankine–vapor–compression refrigeration cycle. The system can be powered by low-grade thermal energy as low as 60 °C and produce cooling temperatures as low as −10 °C. The results showed that R134a achieves the best system performance.

The performance and working fluid selection for a (VCR–ORC) system which recover the waste heat rejected by the condenser of air-conditioning system were examined by Asim et al. [12] Based on thermodynamics (energy and exergy) and thermo-economic analysis, R600a-R123 was chosen as the fluid pair for the integrated system, where the authors concluded that the combined coefficient of performance (COP) of the system could be improved from 3.10 to 3.54.

Mole´s et al. [13] investigated a combined (ORC–VCR) system under various operating conditions that was activated by low temperature heat sources, using low GWP fluids with R134a for the power and refrigeration cycles. They concluded that R1336mzz(Z) and R1234ze(E), respectively, are the best candidates for the power and refrigeration cycles.

The performance of an ORC–VCR system was assessed by Li et al. [14] using the four hydrocarbons propylene, butane, propane, and isobutane. The system's optimal fluid, according to the results, is butane, which has an overall system coefficient of performance of 0.470.

Bu et al. [15] looked into six working fluids: R134a, R123, R245fa, R290, R600a, and R600. Their goal was to find the best working fluids for an ORC–VCR system that was activated by geothermal energy. They concluded that R600a is the best option. Nevertheless, enough attention should be paid to R600a's flammability.

Wang et al. [16] studied an (ORC–VCR) system using two different working fluids for the organic Rankine cycle and conventional vapor compression cycle, namely R245fa and R134a, respectively. The overall system coefficient of performance reached nearly 0.50.

Based on thermodynamics (energy and exergy), Nazer and Zubair [17] and Egrican and Karakas [18] examined an ORC–VCR system using the refrigerants R114 for the Rankine cycle and R22 for the vapor compression cycle. They concluded that it is critical to have the least amount of irreversibility possible in the system to complete the task more cheaply and with a more economical use of natural resources.

Kim and Perez-Blanco [19] examined an (ORC–VCR) system using eight working fluids (R143a, R22, R134a, R152a, propane, ammonia, isobutane, and butane), arranged according to their critical temperatures. The system was activated by low-grade sensible energy. The findings showed that because of its relatively high efficiencies, isobutane provides a sensitivity analysis in a few unique situations.

Three different refrigerants were evaluated by Jeong and Kang [20] to determine which was the best fit for the ORC–VCR system: R123, R134a, and R245ca. The R123 case is found to provide the highest thermal efficiency.

The combined (ORC–VCR) thermodynamic model was developed by Bing et al. [21] for ship air conditioning to transfer the heat from flue gas waste and effectively use cooling water. Using five widely used working fluids: R22, R141b, R236ea, R218 and R601.The system performance was examined. It was determined through calculations that R601 was the best working fluid.

The thermal performance analysis of an ORC–VCR system with a common shaft was the focus of the study of Khatoon et al. [22]. Two refrigerants, R245fa and Propane, were selected for the organic Rankine cycle and three, R245fa, R123, and R134a, were chosen for the vapor compression cycle. When R123 was used as the working fluid in the vapor compression cycle and propane was used in the organic Rankine cycle, the results showed that the former had the highest efficiency (16.48%) and the latter had the highest coefficient of performance (2.85) at 40°C.

The energy and exergy analysis of a combined refrigeration and waste heat driven organic Rankine cycle system was assessed by Cihan and Kavasogullari [23] using five different organic fluids, which are R123, R600, R245fa, R141b, and R600a. They concluded from result of energy and exergy analysis that R141b is the most appropriate organic working fluid.

Kavasogullari et al. [24] studied the performance indices such as the cooling coefficient of performance (COP), exergy destruction and exergy efficiency of dual-ejector refrigeration system (DER) which constructed by adding a second ejector and a refrigeration pump to the classical single-ejector refrigeration system (SER). In their analysis, two different refrigerants (R134a and R600) are employed and compared at the same conditions. According to their results, with the DER system, the maximum cooling COP and energy efficiency are attained by 7.52 and 38.8%, respectively. R134a has a minimum exergy destruction of 9.3 kJ/kg in the DER system at evaporation temperatures of 10 °C and condensing temperatures of 40 °C. Furthermore, at 55 °C for the condenser and 5 °C for the evaporator, R600 produces 5.3% increases in the cooling COP and exergy efficiency. The outcomes also demonstrated that at high condensing and low evaporation temperatures, the DER system produces greater gains in cooling COP and energy efficiency.

Küçük and Kılıç [25] examined a hybrid (ORC–VCR) system operating under diverse conditions for the purpose of producing power and cooling. In the analysis, authors are using the working fluids (R114, R123, R600, R600a, and R245fa) in the ORC system and (R141b, R600a, R290, R134a, R123, R245fa and R143a) in the subsystem VCRC. They concluded that the highest energy utilization factor, exergy efficiency, the system coefficient of performance, and net power is obtained for the R123-R141b fluid pair.

Al-Sayyab et al. [26] looked at the working fluid selection and performance of modified compound organic Rankine-vapor compression cycle using ultra-low global warming potential working fluids (R1234ze(E), R1243zf, and R1234yf). The system can be adapted to three operating modes, depending on the ground source temperature, ranging from 55 to 90 °C: power-cooling, power-heat pump heating, and power-ground source heating. The results indicate that this system notably increases the overall performance of all investigated refrigerants.

Compared to conventional organic Rankine and vapor compression cycles (ORC and VCC), the R1234ze(E) power-cooling mode shows the highest coefficient of performance (COP) increase, 18 %. Besides, including a recapture heat exchanger for condenser waste heat recovery can increase power generation by 58 %. At ground source temperatures up to 65 ℃, power generation and thermal efficiency increased in the power-heating mode due to the absence of the compressor power consumption.

Zhar et al. [27] investigated the performance of combining power and refrigeration system, which consists of an Organic Rankine Cycle driven Vapor Compression Cycle from waste heat source using R123, R11, and R113 as working fluids. The numerical model of the system is developed under Engineering Equation Solver software. Their findings indicate that the R123 is the best working fluid where the highest energy and exergy efficiencies using R123 as working fluid are over 1.02 and 0.53 respectively.

In the study of Wang et al. [28], authors experimented the performance of the (ORC–VCC) system with a zeotropic mixture of R245fa/R134a (0.9/0.1) at various evaporation temperatures and cooling conditions. In addition, the coupling effect of cooling water temperature and flow rate on the performance of the ORC-VCC system are investigated. They concluded that the cooling water temperature has a greater impact on system operational characteristics than cooling water flow rate. As the cooling water temperature, decreases and its flow rate increases, the cooling capacity of the system increases, while the coefficient of performance changes little.

From the abovementioned review about the previous researches, it is clear that there is still a need to search for suitable alternative working fluids for the (ORC–VCR) system that can meet the requirements of cycle performance, environmental performance, thermo-physical properties and safety at the same time, where it was noted that most of the investigations have been focusing on hybrid ORC–VCR system performance evaluation operating with pure working fluids (CFC, HCFC, and HFC) have strong climate impacts, which has several drawbacks, such as environmental performance.

On other hand, the use of conventional hydrocarbons (HCs), such as (butane (R600), isobutene (R600a), propane (R290), etc.) as suitable working fluids in ORC–VCR system is good choice, because these hydrocarbons are rather cheap, plentiful and environmentally benign chemicals (zero ODP and near zero GWP), good thermodynamic performances, thermo-physical properties, non-toxic and have many outstanding properties. However, there are very limited researches, which focused on thermodynamic efficiency analysis of combined power and cooling (ORC–VCRC) system by using this type of working fluids, where we can find research gaps in the cycle performances of the hydrocarbons in the published literature.

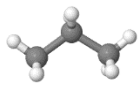
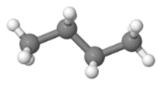
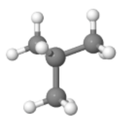
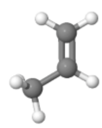
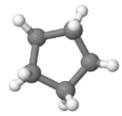
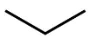
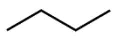
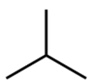
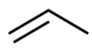
There is recently a renewed interest in the use of cyclopentane gas (C_5H_{10}) as promising working fluid in the thermodynamic systems [29]. This study is conducted for this purpose.

This paper aims to contribute to the field of energy engineering by exploring the performances of cyclopentane gas as a substitute to the conventional hydrocarbons (butane, isobutene, propane and propylene) widely used in (ORC–VCRC) system activated by low-temperature renewable energies having a temperature around 100 °C. Therefore, a comparative computation of various thermodynamic performance characteristics of cyclopentane, butane, isobutene, propane and propylene was carried out under the same operating conditions.

The investigated performance characteristics are (performance indicators (overall coefficient of performance (COP_{oval}) and working fluid mass flow rate of per kW cooling capacity (MkW), expansion ratio in expander (EPR) and compression ratio in compressor (CMR)). Furthermore, the effects of different operating parameters (e.g., boiler, condenser, and evaporator temperatures, isentropic efficiency of expander (η_{exp}), and isentropic efficiency of compressor (η_{comp})) on performance indicators are also examined for each working fluid.

The basic physical and environmental properties of the investigated working fluids are shown in Table 1 [30-36].

Table 1. Basic physical and environmental properties of investigated hydrocarbons [30-36].

Specifications	Unit	Propane	Butane	Isobutene	Propylene	Cyclopentane
Component properties						
Chemical formula	-	C_3H_8	C_4H_{10}	Iso-C_4H_{10}	$CH_3–CH=CH_2$	C_5H_{10}
Type of working fluid	-	HC	HC	HC	HC	HC
Molecular structure	-					
Chemical structure	-					
CAS Registry Number	-	74-98-6	106-97-8	75-28-5	115-07-1	287-92-3
Basic physical properties						
Molar mass	(kg/kmol)	44.096	58.122	58.122	42.08	70.133
Critical temperature	(K)	369.89	425.13	407.85	365.57	511.72
Critical pressure	(MPa)	4.2512	3.796	3.6400	4.665	4.5712
Normal boiling point	(K)	231.04	272.66	261.4	225.46	322.41
Basic environment properties						
ODP (R11 = 1)	-	0	0	0	0	0
GWP (CO2 = 1, 100 yrs)	-	3	20	3	4	<25

2. Description of Hybrid (ORC–VCRC) System

Figure 1 (a) illustrates the schematic diagram of the studied hybrid (ORC–VCRC) system, which consists of two subsystems: the organic Rankine cycle (ORC) identified as (1-2-3-4-1) and the vapor compression refrigeration cycle (VCRC) identified as (5-6-3-7-5). The studied hybrid (ORC–VCRC) system comprises essentially seven components, which are: a feed pump, a boiler, an expander, a condenser, a compressor, a throttle valve and an evaporator.

Figure 1 (b) depicts the corresponding temperature–entropy (T-s) diagram of the state points and various processes of the studied hybrid (ORC–VCRC) system. The various processes of the sub-systems (ORC and VCRC) of the studied hybrid (ORC–VCRC) system can be described as follows:

In the subsystem ORC: process (1→2s) is an isentropic expansion process across the expander, process (1→2) is the condenser, process (3→4s) is an isentropic pumping process, process (3→4) is the actual pumping process and (4→1) is a heat addition process in the boiler.

In the subsystem VCRC: process (5→6s) is an isentropic compression across the compressor, process (5→6) is the actual compression process, process (6→3) is a heat rejection (condensation) process across the condenser, process (3→7) is an isenthalpic expansion across the throttle valve and process (7→5) is a heat absorption (evaporation) in the evaporator.

The two cycles work respectively as follows:

- In the subsystem ORC, the condensed working fluid is pressurized by the feed pump and enters the boiler where it is heated by the low temperature heat source. Then the vapor of working fluid produced in the boiler at high-pressure flows into the expander, which produces mechanical work to drive the compressor in the VCRC. Subsequently, the working fluid returns to the condenser and the ORC is completed.
- In the subsystem VCRC, the liquid working fluid out of the condenser goes through the throttle valve and enters, the evaporator where the low pressure and low temperature working fluid vaporizes and cools the conditioned space where it produces cold. In the sequence, the vapor of working fluid is sucked into the compressor where it is pressurized and then discharged into the condenser to complete the VCRC.

3. Methodology and Mathematical Formulation

3.1 Thermodynamic Assumptions

To develop the thermodynamic models of the studied hybrid (ORC–VCRC) system, the following thermodynamic assumptions are made:

- Steady-state flow in each component is considered;
- The pressure and heat losses in the hybrid (ORC–VCRC) system are negligible;
- The potential energy and kinetic energy are not considered in the hybrid (ORC–VCRC) system;
- The transformations in all heat exchangers are isobaric process;
- The working fluids at the exit of the boiler and evaporator are assumed to be saturated;
- The condenser has a given subcooling of 3 °C to prevent boiler feed pump cavitation;
- The flow through in the throttle valve is isenthalpic process.

According to the given thermodynamic assumptions, the mathematical models used to determine the thermodynamic performance characteristics of the studied hybrid (ORC–VCRC) system could be established.

3.2 Modelling of Components and Energy Analysis

After applying the first law of thermodynamics to various components of a combined (ORC–VCRC) system, the following mathematical models were constructed:

Organic Rankine Cycle (ORC):

Expander:

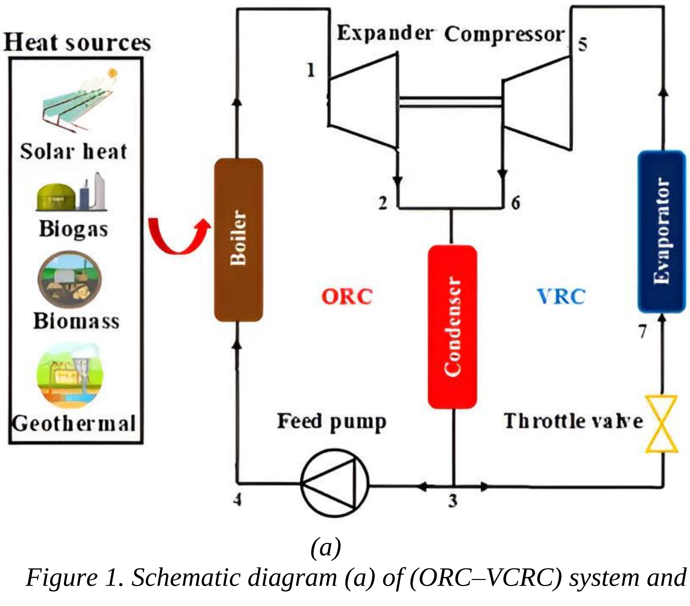
$$W_{\exp} = m_{ORC}\left(h_1 - h_{2s}\right)\eta_{\exp} \tag{1}$$

Pump:

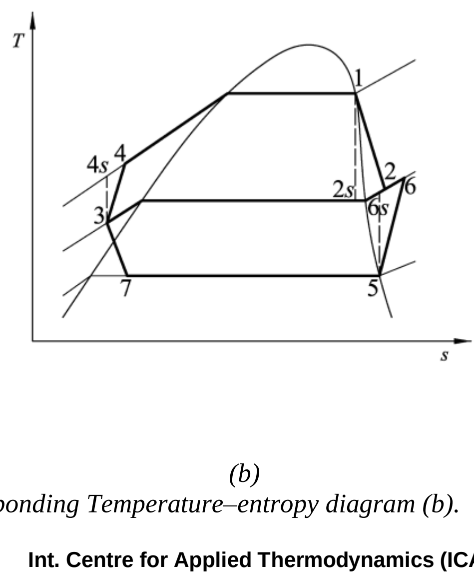
$$W_{pump} = \frac{m_{ORC}\left(h_{4s} - h_3\right)}{\eta_{pump}} \tag{2}$$

Boiler:

$$Q_{boil} = m_{ORC}\left(h_1 - h_4\right) \tag{3}$$



(a)



(b)

Figure 1. Schematic diagram (a) of (ORC–VCRC) system and its corresponding Temperature–entropy diagram (b).

The thermal efficiency of the organic Rankine cycle (ORC) is defined as:

$$\eta_{ORC} = \frac{W_{net}}{Q_{boil}} \tag{4}$$

Vapor Compression Refrigeration Cycle (VCRC):
Evaporator:

$$Q_{ev} = m_{VCRC}(h_5 - h_7) \tag{5}$$

Compressor:

$$W_{comp} = \frac{m_{VCRC}(h_5 - h_{6s})}{\eta_{comp}} \tag{6}$$

The COP of vapor compression refrigeration cycle (VCRC) is given by:

$$COP_{VCRC} = \frac{Q_{ev}}{W_{comp}} \tag{7}$$

With [14]:

$$W_{net} = W_{\exp} - W_{pump} \tag{8}$$

$$W_{comp} = W_{net} \tag{9}$$

The overall coefficient of performance (COP_{oval}) of the hybrid (ORC–VCRC) system can be calculated as follows:

$$COP_{oval} = \eta_{ORC} COP_{VCRC} \tag{10}$$

The working fluid mass rate of per kW cooling capacity in the ORC–VCRC system is expressed as:

$$MkW = \frac{m_{ORC} + m_{VCR}}{Q_{ev}} \tag{11}$$

The expansion ratio in expander (EPR) and compression ratio in compressor (CMR) are calculated for the required compressor and expander sizes, respectively, and described as follows:

$$EPR = \frac{v_2}{v_1} \tag{12}$$

$$CMR = \frac{p_6}{p_5} \tag{13}$$

3.3 Operating Conditions for the Combined Power and Cooling (ORC–VCRC) Calculation

The operating conditions for the system calculation were:

- The temperature of heat source was 100 °C;
- The working fluid mass flow rate in ORC is taken to be 1.0 kg/s;
- The boiler temperature ranged from 60 to 90 °C;
- The condensation temperature ranged from 30 to 55 °C;
- The evaporation temperature ranged from -15 to 15 °C;
- The typical values of boiler temperature, condensation temperature and evaporation temperature are set as 80, 40 and 5 °C, respectively;
- The isentropic efficiency of the compressor, feed pump and expander are taken to be 75, 75 and 80 %, respectively [14].

Based on the assumptions and mathematical models above, a computer code was written and developed in MATLAB software and integrated with REFPROP Version 9.0. With the given input parameters, the code calculates all thermodynamic properties of each point of the (ORC–VCRC) system and simulate the thermodynamic performance characteristics of the hydrocarbons (propane, butane, isobutene, propylene, and cyclopentane) in (ORC–VCRC) system in a wide range of working conditions.

4. Results and Discussion

4.1 Model Validation

To validate the present model, the performance indicators (overall coefficient of performance (COP_{oval}) and working fluid mass flow rate of per kW cooling capacity (MkW)) have been compared with the available data in the literature using the isobutene as working fluid in hybrid (ORC–VCRC) system.

The results of this study were compared with the simulation data published by Li et al. [14] under the same operating conditions (boiler temperatures vary from (60 to 90 °C), constant condensation temperature of 40 °C, constant evaporation temperature of 5 °C).The simulation results are illustrated in Figure 2.

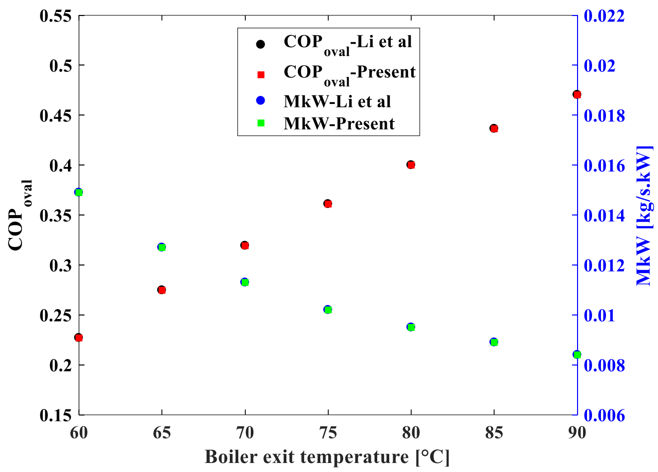


Figure 2. Validation of the present work.

Figure 2 shows an explicit agreement between the performance indicators results obtained in this study and the data published in the preceding literature.

The comparison results indicate that the deviation between the calculated performance indicators (COP_{oval} and MkW) and the reference values are small, with a mean deviation being 0.1 % and 0.2 % for COP_{oval} and MkW, respectively. These deviations are very acceptable and indicate that the developed model can calculate the thermodynamic performances of combined power and cooling (ORC–VCRC) system, which confirms the validity of our simulation model.

The following section presents the simulation results of the comparative evaluation of thermodynamic performance characteristics of the investigated hydrocarbons in (ORC–VCRC) system.

4.2 Performances Analysis

4.2.1 Effect of Boiler Exit Temperature

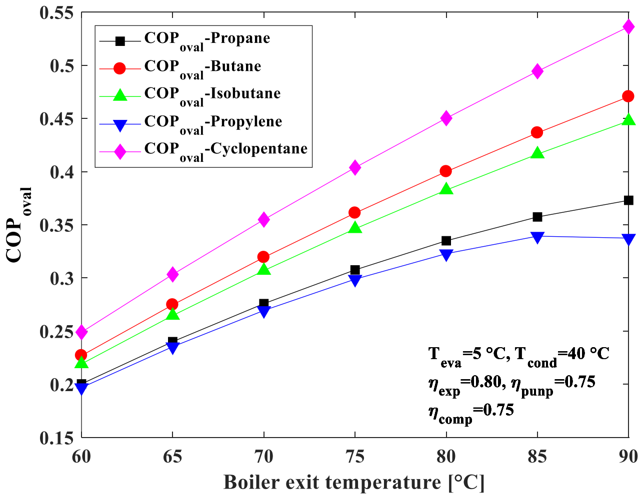


Figure 3. Effect of variation of boiler exit temperature on COP_{oval}.

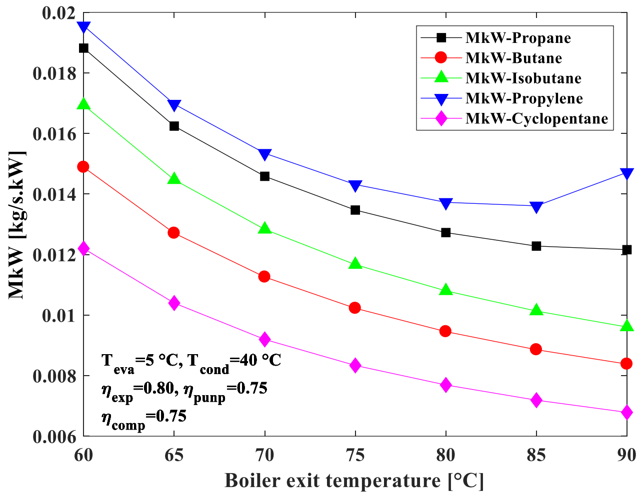


Figure 4. Effect of variation of boiler exit temperature on MkW.

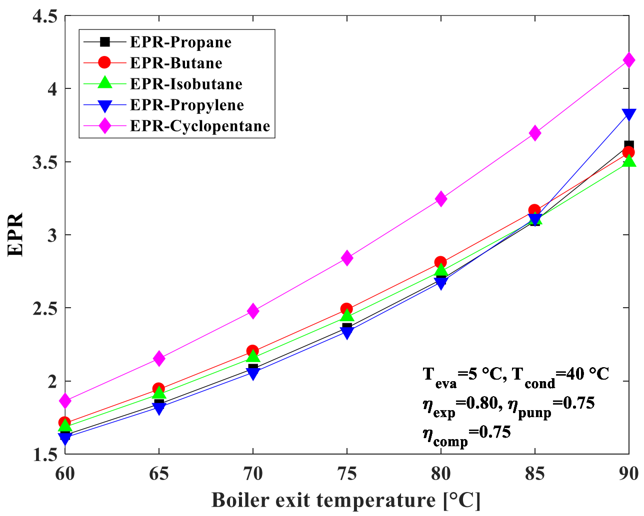


Figure 5. Effect of variation of boiler exit temperature on EPR.

Figure 3 illustrates the simulation results of the effect of variation of boiler exit temperature (T_{boil}=60 to 90 °C) on COP_{oval} of combined power and cooling (ORC–VCRC) system, while maintaining the other variables unchanged, using the hydrocarbons: propane, butane, isobutene, propylene, and cyclopentane as working fluids.

From the curves of the variation of the COP_{oval}, it was noticed that the COP_{oval} keeps increasing as T_{boil} increases for each working fluid, where it can be observed that the lower value of boiler exit temperature (T_{boil}=60 °C) led to a lower value of COP_{oval}, however, as the T_{boil} increased, the COP_{oval} also increased. The following provides an explanation of the situation's cause. Both the compressor's cooling capability and power consumption are constant when the condenser and evaporator temperatures remain constant. Therefore, the COP_{VCRC} of VCRC system does not change. On the other hand, η_{ORC} of the ORC system will increase when boiler temperature increases and condenser temperature is constant. Since the input power to the compressor must be equal to the net output power of the expander, the ORC with a high boiler temperature produces the same power with a lower mass flow rate. This raises the ORC's thermal efficiency by reducing the amount of heat added to the boiler. The (ORC–VCRC) system becomes more efficient as the boiler temperature rises because of these two factors. On the other hand, the results showed that the maximum COP_{oval} of combined power and cooling (ORC–VCRC) system working with the propylene is lower than that obtained with the other studied working fluids, while for the cyclopentane, the COP_{oval} is higher than the obtained with butane, isobutene, propane and propylene for all the boiler exit temperature range studied. The results indicate that when boiler temperature reaches 90 °C, the COP_{oval} of cyclopentane increase by 14 %, 19.8 %, 43.8 %, and 59 % compared to those of butane, isobutene, propane and propylene, respectively.

When the boiler exit temperature increases from 60 to 90 °C, the COP_{oval} of the five working fluids (cyclopentane, butane, isobutene, propane, and propylene) increases from (0.2490 to 0.5362), (0.2270 to 0.4704), (0.2192 to 0.4476), (0.2004 to 0.3730) and (0.1969 to 0.3374), respectively.

Figure 4 exhibits the simulation results of the effect of variation of boiler exit temperature (T_{boil}=60 to 90 °C) on MkW of combined power and cooling (ORC–VCRC) system using the hydrocarbons: propane, butane, isobutene, propylene, and cyclopentane as working fluids.

From the curves of the variation of the MkW, it was noticed that the MkW keeps decreasing as boiler exit temperature increases for each working fluids, where it can be observed that the lower value of boiler exit temperature (60 °C) led to a higher value of MkW, however, as the boiler exit temperature increased, the MkW decreased.

By comparing the simulation results obtained, the results showed that the MkW of combined power and cooling (ORC–VCRC) system working with propylene is higher than that obtained with the other studied working fluids, while for cyclopentane, the MkW is lower than the obtained with propane, butane, isobutene and propylene for all the temperature range studied, which confirms that it could be a good working fluid for the (ORC–VCRC) system. The results indicate that at (T_{boil}=90 °C), the MkW of cyclopentane reduced by 19.1 %, 29.2 %, 44.3 %, and 53.7 % compared to those of butane, isobutene, propane, and propylene, respectively.

When the boiler exit temperature increases from 60 to 90 °C, the MkW of the five working fluids (cyclopentane, butane, isobutene, propane, and propylene) decreases from (0.0122 to 0.0068), (0.0149 to 0.0084), (0.0169 to 0.0096), (0.0188 to 0.0122) and (0.0196 to 0.0147), respectively.

The effect of the boiler exit temperature (T_{boil}=60 to 90 °C) on the expansion ratio in expander (EPR), while maintaining the other variables unchanged, using the hydrocarbons: propane, butane, isobutene, propylene and cyclopentane as working fluids is displayed in Figure 5.

As shown in the figure, like the COP_{oval}, the expansion ratio in expander (EPR) increases as the boiler exit temperature increases from 60 to 90 °C for all investigating working fluids. The following provides an explanation of the situation's cause. The boiler temperature (saturation pressure) increases while the temperature of the condenser (saturation pressure) is constant. As a result, the specific volume drops and the pressure at the expander inlet rises. The expansion ratio in expander (EPR) rises because of this.

From the curves of the variation of the expansion ratio in expander (EPR), it was noticed that the expansion ratio in expander (EPR) of combined power and cooling (ORC–VCRC) system, which working with the cyclopentane is higher than the obtained with the other working fluids.

When the boiler exit temperature increases from 60 to 90 °C, the EPR of the five working fluids (cyclopentane, butane, isobutene, propane, and propylene) increases from (1.8636 to 4.1951), (1.7112 to 3.5607), (1.6850 to 3.4951), (1.6302 to 3.6109) and (1.6126 to 3.8321), respectively.

4.2.2 Effect of Condensation Temperature

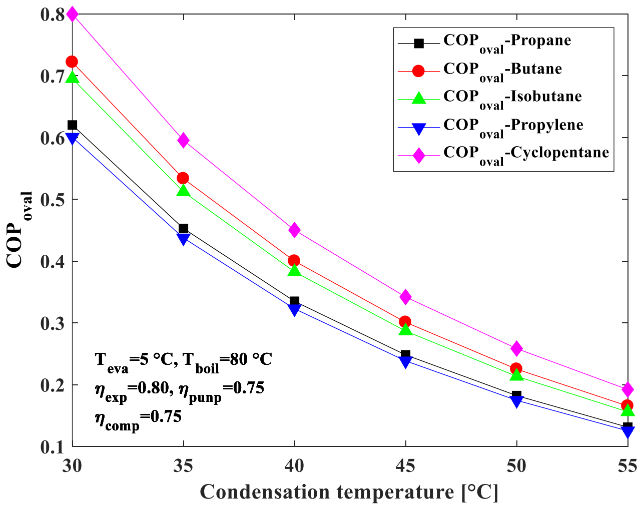


Figure 6. Effect of variation of condensation temperature on COP_{oval}.

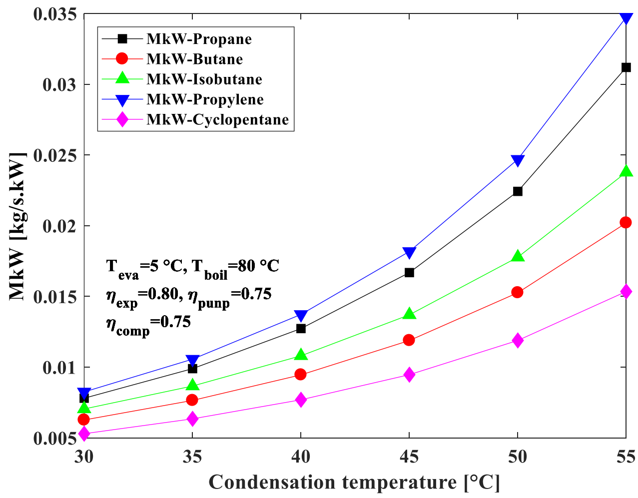


Figure 7. Effect of variation of condensation temperature on MkW.

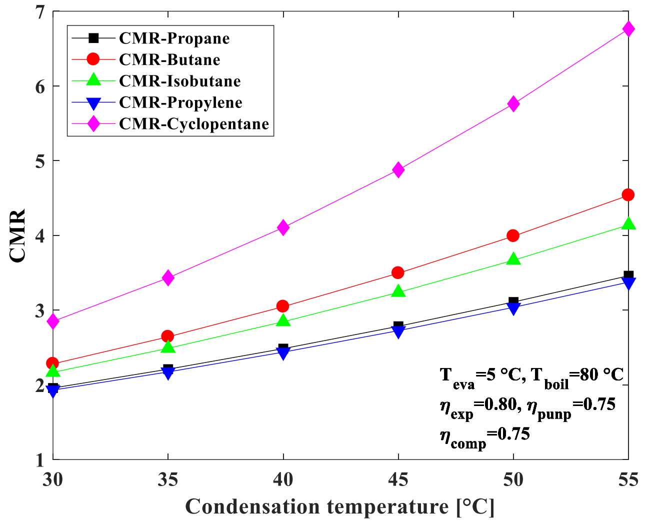


Figure 8. Effect of variation of condensation temperature on CMR.

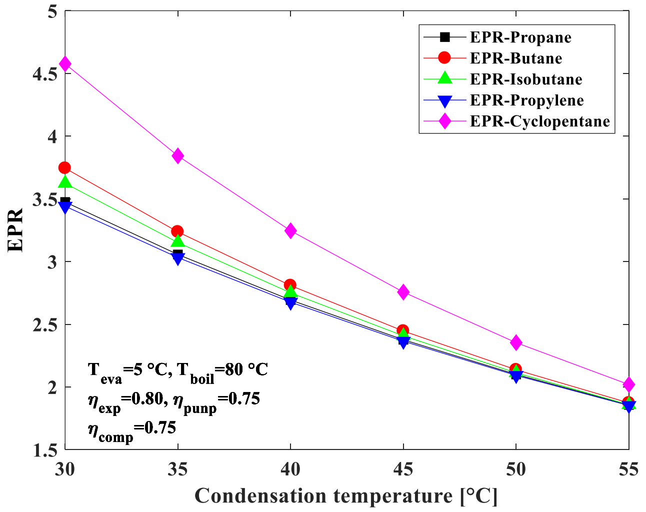


Figure 9. Effect of variation of condensation temperature on EPR.

Figure 6 illustrates the variation of condensation temperature (T_{cond} = 30 to 55 °C) on COP_{oval} of combined power and cooling (ORC–VCRC) system while maintaining the other variables unchanged, using the hydrocarbons: propane, butane, isobutene, propylene, and cyclopentane as working fluids.

As clearly shown in the figure, the results showed that the curves of the variation of the COP_{oval} decreases as condensation temperature increases for all the considered working fluids, where it can be observed that the lower value of condensation temperature (30 °C) led to a higher value of COP_{oval}, however, as the condensation temperature increases (55 °C), the COP_{oval} decreases.

As can be seen from equation (10) the COP_{oval} value of the combined power and cooling (ORC–VCRC) system is found by multiplying the thermal efficiency of ORC and coefficient of cooling performance of the VCRC. As the T_{cond} increases, the COP_{VCRC} value decreases as expected for all working fluids, while the evaporator temperature is constant. On the other hand, the η_{ORC} value decreases since the heat addition to the boiler increases while the condenser temperature is increasing.

According to the figure, the results showed that the COP_{oval} of combined power and cooling (ORC–VCRC) system working with the propylene is lower than that obtained with the other studied working fluids, while for the cyclopentane, the COP_{oval} is higher than the obtained with propane, butane, isobutene and propylene for all the condensation temperature range studied.

The results indicate that at (T_{cond}=30 °C), the COP_{oval} of cyclopentane increase by 16.1 %, 23 %, 46.4 % and 53.4 % compared to those of butane, isobutene, propane and propylene, respectively.

When the condensation temperature increases from 30 to 55 °C, the COP_{oval} of the five working fluids (cyclopentane, butane, isobutene, propane and propylene) decreases from (0.7996 to 0.1922), (0.7218 to 0.1655), (0.6952 to 0.1562), (0.6201 to 0.1313) and (0.6004 to 0.1253), respectively.

Figure 7 represents the effect of variation of condensation temperature on MkW of combined power and cooling (ORC–VCRC) system while maintaining the other variables unchanged, using the hydrocarbons: propane, butane, isobutene, propylene and cyclopentane as working fluids. As can be observed in the figure, the MkW increases when increasing the temperature of the condenser from 30 to 55 °C.

By comparing obtained results of MkW of each working fluids, the figure showed that the MkW of combined power and cooling (ORC–VCRC) system working with propylene is higher than that obtained with the other studied working fluids, while for cyclopentane, the MkW is lower than the obtained with propane, butane, isobutene and propylene for all the condensation temperature range studied.

The results indicate that at (T_{cond}=30 °C), the MkW of cyclopentane reduced by 24.3 %, 35.7 %, 50.9 % and 55.9 % compared to those of butane, isobutene, propane, and propylene, respectively.

When the condensation temperature increases from 30 to 50 °C, the MkW of the working fluids (cyclopentane, butane, isobutene, propane and propylene) increases from (0.0053 to 0.0153), (0.0063 to 0.0202), (0.0070 to 0.0238), (0.0078 to 0.0312) and (0.0082 to 0.0347), respectively.

Figure 8 shows the evolution of the compression ratio in compressor (CMR) of the combined power and cooling (ORC–VCRC) system with various condensation temperatures while maintaining the other variables unchanged, using the hydrocarbons: propane, butane, isobutene, propylene and cyclopentane as working fluids.

From the curves of the variation of the compression ratio in compressor (CMR), it was noticed that the compression ratio in compressor (CMR) keeps increasing as condensation temperature increases for each working fluids, where it can be observed that the lower value of condensation temperature (30 °C) led to a lower value of CMR, however, as the condensation temperature increased (55 °C), the compression ratio in compressor (CMR) increased.

The following provides an explanation of the situation's cause. The compressor saturation pressure rises and the compression ratio in compressor (CMR) value rises as the condenser temperature rises to the constant evaporator temperature (saturation pressure).

It can also be seen that cyclopentane has the highest values of compression ratio in compressor (CMR) compared to the other working fluids.

When the condensation temperature increases from 30 to 55 °C, the compression ratio in compressor (CMR) of the working fluids (cyclopentane, butane, isobutene, propane and propylene) increases from (2.8497 to 6.7639), (2.2810 to 4.5365), (2.1675 to 4.1398), (1.9578 to 3.4606) and (1.9299 to 3.3746), respectively.

The effect of the condensation temperature on the expansion ratio in expander (EPR) of combined power and cooling (ORC–VCRC) system while maintaining the other variables unchanged, using the hydrocarbons: propane, butane, isobutene, propylene and cyclopentane as working fluids is displayed in Figure 9.

As shown in the figure, like the COP_{oval}, the expansion ratio in expander (EPR) decreases as the condensation temperature increases from 30 to 55 °C for all investigating working fluids.

From the curves of the variation of the EPR, it was noticed that the maximum EPR of combined power and cooling (ORC–VCRC) system, which is working with the cyclopentane is better than the obtained with the other working fluids. Additionally, it was found that the values of the EPR of the propane are like propylene for all the condensation temperature range studied.

When the condensation temperature increases from 50 to 60 °C, the EPR of the five working fluids (cyclopentane, butane, isobutene, propane and propylene) decreases from (4.5767 to 2.0190), (3.7443 to 1.8729), (3.6239 to 1.8586), (3.4750 to 3.4606) and (3.4412 to 1.8520), respectively.

4.2.3 Effect of Evaporation Temperature

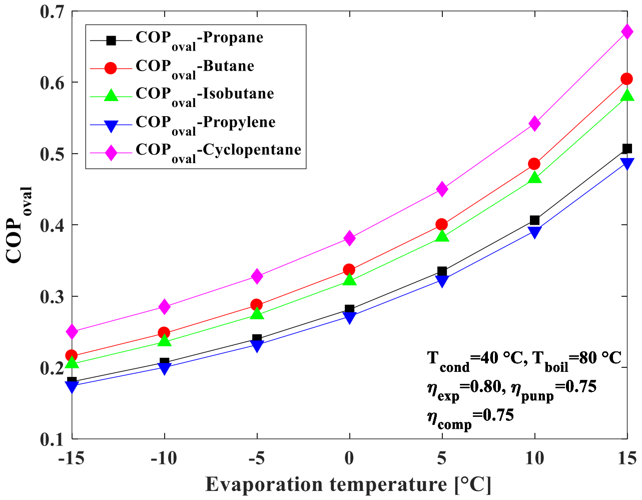


Figure 10. Effect of variation of evaporation temperature on COP_{oval}.

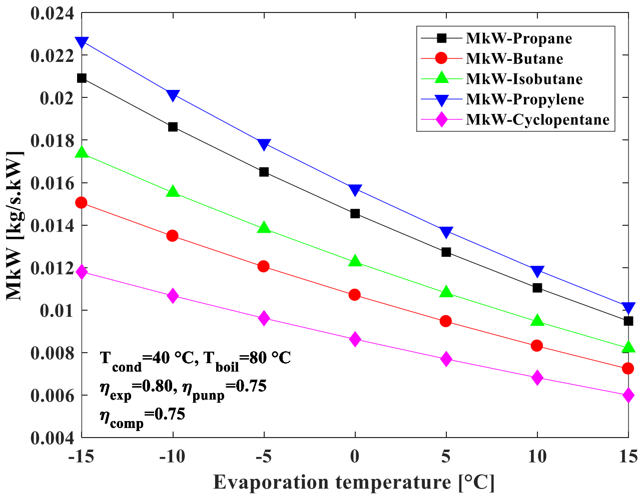


Figure 11. Effect of variation of evaporation temperature on MkW.

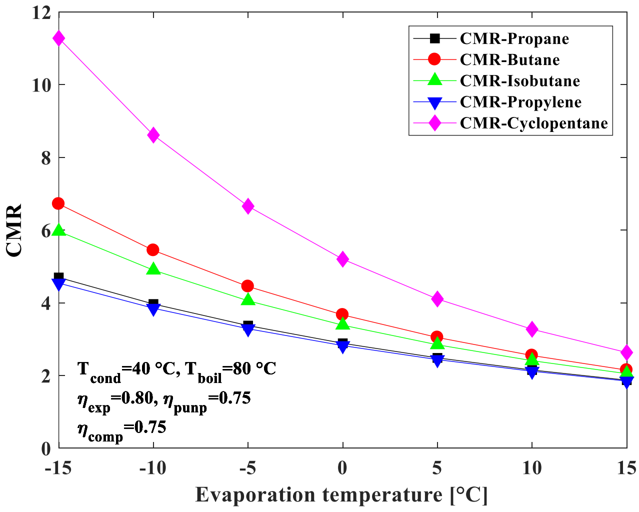


Figure 12. Effect of variation of evaporation temperature on CMR.

The effect of evaporation temperature (T_{eva}=-15 to 15 °C) on the COP_{oval} of combined power and cooling (ORC–VCRC) system while maintaining the other variables unchanged is displayed in Figure 10.

As shown in figure, the curves of the variation of the COP_{oval} keeps increasing as T_{eva} increases for each working fluids, where it can be observed that the lower value of evaporation temperature (T_{eva}=-15 °C) led to a lower value of COP_{oval}, however, as the T_{eva} increased, the COP_{oval} also increased.

The following provides an explanation of the situation's cause. The evaporator capacity increases and the compressor pressure ratio decreases with increased evaporator temperature (saturation pressure) at constant condenser temperature, resulting in a reduction in the compressor's power consumption. This also increases the value of COP_{VCRC} of VCRC system. On the other hand, the η_{ORC} value does not change. Because of these two effects, the COP_{oval} increases as the T_{eva} increases.

On other hand, it was noticed that the best performing working fluid is the cyclopentane. The results indicate that at (T_{eva}=15 °C), the COP_{oval} of cyclopentane increase by 11.1 %, 15.7 %, 32.3 %, and 37.6 % compared to those of butane, isobutene, propane and propylene, respectively.

When the evaporation temperature increases from -15 to 15 °C, the COP_{oval} of the working fluids (cyclopentane, butane, isobutene, propane and propylene) increases from (0.2503 to 0.6710), (0.2161 to 0.6041), (0.2051 to 0.5799), (0.1802 to 0.5071) and (0.1745 to 0.4877), respectively.

Figure 11 gives the effect of variation of evaporation temperature on MkW for the combined power and cooling (ORC–VCRC) system operating with the working fluids (propane, butane, isobutene, propylene, and cyclopentane) while maintaining the other variables unchanged.

From the curves of the variation of the MkW, it was noticed that the MkW keeps decreasing as evaporation temperature increases for each working fluids, where it can be observed that the lower value of evaporation temperature (T_{eva}=-15 °C) led to a higher value of MkW, however, as the T_{eva} increased, the MkW decreased.

The results indicate that at (T_{eva}=15 °C), the MkW of cyclopentane reduced by 16.7 %, 26.9 %, 36.9 %, and 41.2 % compared to those of butane, isobutene, propane, and propylene, respectively.

When the evaporation temperature (T_{eva}) increases from -15 to 15 °C, the MkW of the five working fluids (cyclopentane, butane, isobutene, propane and propylene) decreases from (0.0118 to 0.0060), (0.0150 to 0.0072), (0.0174 to 0.0082), (0.0209 to 0.0095) and (0.0227 to 0.0102) , respectively.

The effect of evaporation temperature on the CMR of the combined power and cooling (ORC–VCRC) system is plotted in Figure 12.

It can be observed from figure that the lower value of evaporation temperature (T_{eva}=-15 °C) led to a higher value of CMR, however, as the evaporation temperature increased, the CMR decreased.

The following provides an explanation of the situation's cause. Increased evaporator temperature (saturation pressure) at the constant condenser temperature causes the compressor pressure ratio to decrease.

When the evaporation temperature (T_{eva}) increases from -15 to 15 °C, the CMR of the five working fluids (cyclopentane, butane, isobutene, propane and propylene) decreases from (11.2766 to 2.6321), (6.7171 to 2.1487), (5.9651 to 2.0511), (4.6958 to 1.8720) and (4.5409 to 1.8481), respectively.

4.2.4 Effect of Expander Isentropic Efficiency

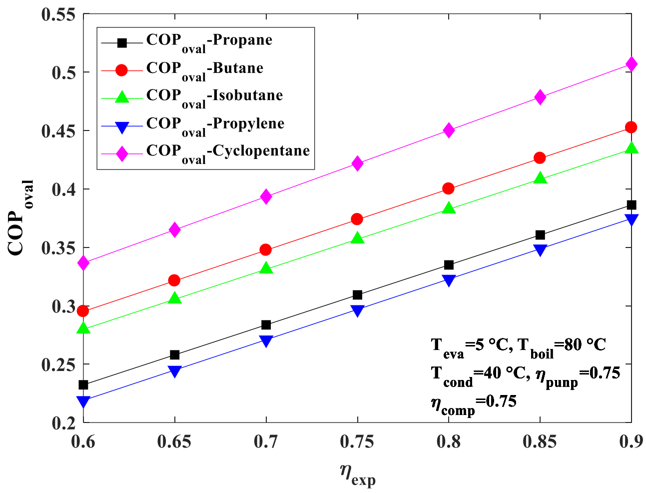


Figure 13. Effect of expander isentropic efficiency on COP_{oval}.

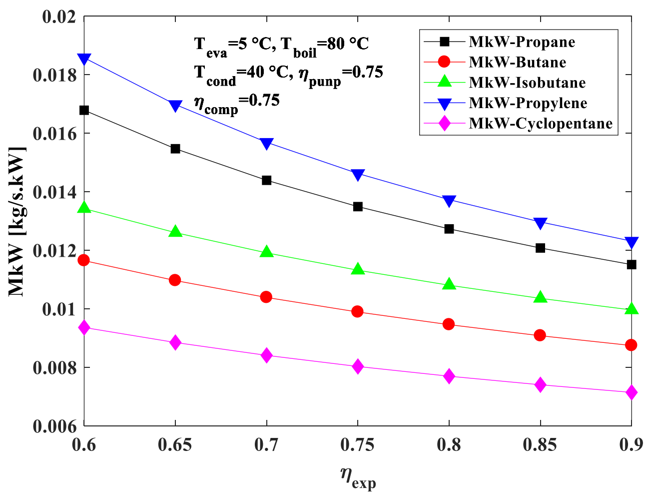


Figure 14. Effect of expander isentropic efficiency on MkW.

Figure 14 reveals the variation of overall coefficient of performance (COP_{oval}) by varying the expander isentropic efficiency (η_{exp}=0.6 to 0.9) for the combined power and cooling system (organic Rankine cycle (ORC)–vapor compression refrigeration cycle (VCRC)) activated by low-temperature renewable energies operating with the investigated working fluids (propane, butane, isobutene, propylene, and cyclopentane).

From the results obtained, it can be observed that an increase in the expander isentropic efficiency leads to a gradual increase in overall coefficient of performance for all investigated working fluids.

The maximum COP_{oval} of system working with the propylene is lower than that obtained with the other studied working fluids, while for the cyclopentane, the COP_{oval} is higher than the obtained with butane, isobutene, propane, and propylene.

The results indicate that at expander isentropic efficiency of 0.9, the overall coefficient of performance of cyclopentane increase by 12.1 %, 16.9 %, 31.1 %, and 35.3 % compared to those of butane, isobutene, propane, and propylene, respectively.

When the expander isentropic efficiency increases from 0.6 to 0.9, the overall coefficient of performance of the working fluids (cyclopentane, butane, isobutene, propane, and propylene) increases from (0.3367 to 0.5070), (0.2951 to 0.4524), (0.2799 to 0.4340), (0.2323 to 0.3868) and (0.2190 to 0.3748), respectively.

The effect of expander isentropic efficiency on MkW of the (ORC–VCRC) system is plotted in Figure 14.

From the simulated results, it can be observed that the MkW curves of the working fluids decreases with the increasing of the expander isentropic efficiency.

The results indicate that at (η_{exp}=0.9), the MkW of cyclopentane reduced by 18.4 %, 29 %, 38.3 %, and 42.3 % compared to those of butane, isobutene, propane, and propylene, respectively.

When the expander isentropic efficiency increases from 0.6 to 0.9, the MkW of the (cyclopentane, butane, isobutene, propane, and propylene) decreases from (0.0094 to 0.0071), (0.0116 to 0.0087), (0.0134 to 0.0100), (0.0168 to 0.0115), (0.0186 to 0.0123), respectively.

4.2.5 Effect of Compressor Isentropic Efficiency

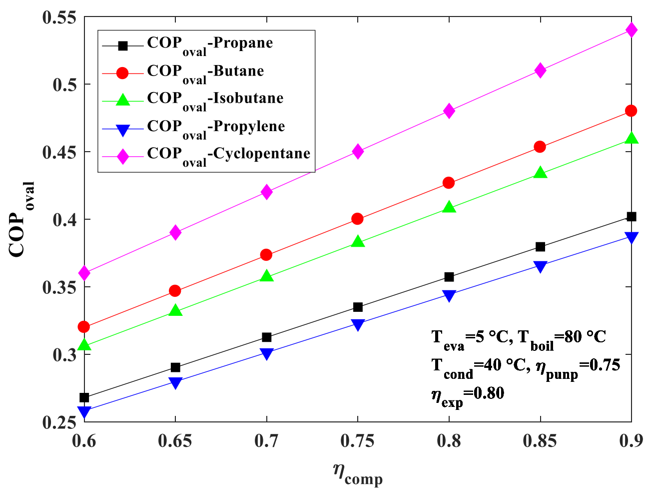


Figure 15. Effect of compressor isentropic efficiency on COP_{oval}.

Figure 15 shows the effect of compressor isentropic efficiency on COP_{oval} for the combined power and cooling (ORC–VCRC) system operating with the working fluids (propane, butane, isobutene, propylene, and cyclopentane).

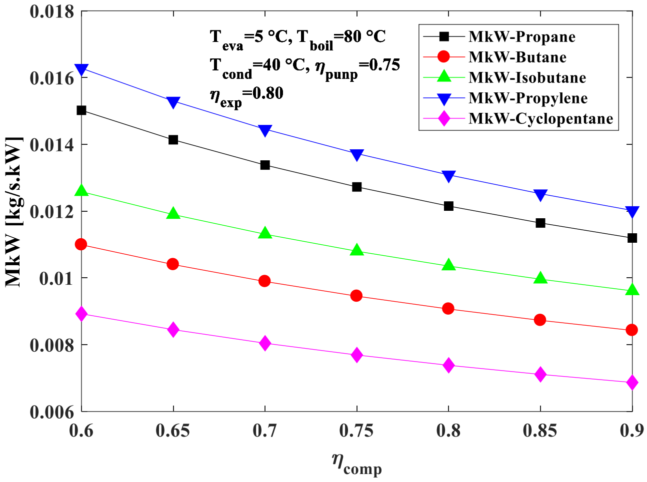


Figure 16. Effect of compressor isentropic efficiency on MkW.

From the simulation results obtained, it is seen that an increase in the compressor isentropic efficiency leads to a gradual increase in COP_{oval} for all working fluids. The results indicate that at (η_{comp}=0.9), the COP_{oval} of cyclopentane increase by 12.6 %, 17.7 %, 34.4 % and 39.5 % compared to those of butane, isobutene, propane and propylene, respectively.

When the compressor isentropic efficiency increases from 0.6 to 0.9, the COP_{oval} of the five working fluids (cyclopentane, butane, isobutene, propane and propylene) increases from (0.3602 to 0.5403), (0.3200 to 0.4800), (0.3061 to 0.4592), (0.2680 to 0.4020) and (0.2583 to 0.3874), respectively.

Figure 16 gives the simulation results of the variation of the MkW values of the (ORC–VCRC) system for working fluids versus the compressor isentropic efficiency (η_{comp}=0.6 to 0.9). As can be observed in the figure, the MkW decreases when increasing the efficiency of the compressor.

The results indicate that at (η_{comp}=0.9), the MkW of cyclopentane reduced by 17.9 %, 28.1 %, 38.4 % ,and 42.5 % compared to those of butane, isobutene, propane and propylene, respectively.

When the compressor isentropic efficiency increases from 0.6 to 0.9, the MkW of the (cyclopentane, butane, isobutene, propane, and propylene) decreases from (0.0089 to 0.0069), (0.0110 to 0.0084), (0.0126 to 0.0096), (0.0150 to 0.0112), (0.0163 to 0.0120), respectively.

5. Conclusions

This study examined the theoretical use of power from ORC using renewable energy sources with low temperature heat sources in VCRC system for cooling using cyclopentane gas as promising working fluid to replace the conventional hydrocarbons (butane, isobutene, propane, and propylene) widely used in (ORC–VCRC) system.

A comparative examination of the performance characteristics of (ORC–VCRC) system between the working fluids are presented. The performance characteristics investigated are (performance indicators (overall coefficient of performance (COP_{oval}) and working fluid mass flow rate of per kW cooling capacity (MkW), expansion ratio in expander (EPR) and compression ratio in compressor (CMR)). Furthermore, the effects of different operating parameters (e.g., boiler, condenser, and evaporator temperatures, isentropic efficiency of expander (η_{exp}), and isentropic efficiency of compressor (η_{comp})) on performance indicators are also examined for each working fluid.

The simulation was realized by adjusting the boiler temperature between 60°C and 90°C, the condenser temperature between 30°C and 55°C, and the evaporator temperature between -15°C and 15°C.

Based on the results obtained from the present study, the main conclusions are listed as follows:

- The COP_{oval} rises as the boiler and evaporator temperatures rises. Conversely, when the condenser temperature rises, the COP_{oval} value falls for all fluids;
- The MkW value falls as the boiler and evaporator temperatures rises. Conversely, when the condenser temperature rises, the COP_{oval} value rises for all fluids;
- The EPR rises as the boiler temperatures rises. Conversely, when the condenser temperatures rises, the EPR value falls for all fluids;
- The CMR rises as the condenser temperatures rises. Conversely, when the evaporator temperatures rises, the CMR value falls for all fluids;
- The COP_{oval} rises as the expander isentropic efficiency rises. Conversely, when this parameter rises, the MkW value falls for all fluids;
- The COP_{oval} rises as the compressor isentropic efficiency rises. Conversely, when this parameter rises, the MkW value falls for all fluids;
- The maximum COP_{oval} of (ORC–VCRC) system is obtained with cyclopentane;
- The minimum COP_{oval} of (ORC–VCRC) system is obtained with propylene;
- The maximum MkW of (ORC–VCRC) system is obtained with propylene;
- The minimum MkW of (ORC–VCRC) system is obtained with cyclopentane;

By analyzing the performance characteristics of studied working fluids, the investigated cyclopentane gas emerges better performances in most of the cases, which confirms that it could be a promising working fluid in terms of performance indicators for (ORC–VCRC) system.

In future works, it would be very interesting to research about a novel design of a power-cooling system using cyclopentane gas as working fluid to improve the performance indicators of the conventional (ORC–VCRC) system.

Nomenclature

Symbols

COP_{VCRC}	:	Coefficient of performance of VCRC
COP_{oval}	:	Overall coefficient of performance of the hybrid ORC–VCRC system
CMR	:	Compression ratio in compressor
EPR	:	Expansion ratio in expander
h	:	Specific enthalpy (kJ kg^{-1})
p	:	Pressure (kPa)
m	:	Mass flow rate (kg s^{-1})
MkW	:	Working fluid mass flow rate of per kW cooling capacity (kg s^{-1} kW^{-1})
s	:	Specific entropy (kJ kg^{-1} K^{-1})
v	:	Specific volume (m^3 kg^{-1})
Q_{boil}	:	Boiler heat input (kW)
Q_{ev}	:	Evaporator cooling capacity (kW)
W_{comp}	:	Compressor work input (kW)
W_{exp}	:	Expander work output (kW)
W_{net}	:	Net work output (kW)
W_{pump}	:	Pump power consumption (kW)
T	:	Temperature (°C)

Greek symbols

η_{comp}	:	Isentropic efficiency of compressor
η_{exp}	:	Isentropic efficiency of expander
η_{pump}	:	Isentropic efficiency of pump
η_{ORC}	:	Power cycle thermal efficiency

Subscripts

cond	:	Condensing process
eva	:	Evaporation process
boil	:	Boiler process
s	:	Isentropic process
ORC	:	Organic Rankine cycle
VCRC	:	Vapor compression refrigeration cycle
1, 2, 3, ...	:	Respective state points in the hybrid (ORC–VCRC) system

Abbreviations

ORC	:	Organic Rankine cycle
VCRC	:	Vapor compression refrigeration cycle
GWP	:	Global warming potential
ODP	:	Ozone depleting potential

References:

[1] P. Gang, L. Jing, J. Jie, "Design and analysis of a novel low-temperature solar thermal electric 465 system with two-stage collectors and heat storage units," *Renew. Energy*., 36, 2324–2333, 2011, doi:10.1016/j.renene.2011.02.008.

[2] M. Ciani Bassetti, D. Consoli, G. Manente, A. Lazzaretto, "Design and off-design models of a 468 hybrid geothermal-solar power plant enhanced by a thermal storage," *Renew. Energy*., 128, 460–472, 2018, doi:10.1016/j.renene.2017.05.078.

[3] H.Cho, A.D, Smith, P .Mago, "Combined cooling, heating and power: a review of performance improvement and optimization," *Applied Energy*., 136, 168–185, 2014, doi:10.1016/j.apenergy.2014.08.107.

[4] H. Chang, Z. Wan, Y. Zheng, X. Chen, S. Shu, Z. Tu, S.H. Chan, "Energy analysis of a hybrid PEMFC-solar energy residential micro CCHP system combined with an organic Rankine cycle and vapor compression cycle," *Energy Conversion and Management*., 142, 374–384, 2017, doi:10.1016/j.enconman.2017.03.057.

[5] C. Yue, F. You, Y. Huang, "Thermal and economic analysis of an energy system of an ORC coupled with vehicle air conditioning," *International Journal of Refrigeration*., 64, 152–167, 2016, doi:10.1016/j.ijrefrig.2016.01.005.

[6] J.M. Calm, "The next generation of refrigerants-historical review, considerations, and Outlook," *International Journal of Refrigeration*., 31, 1123–1133, 2008, doi:10.1016/j.ijrefrig.2016.01.005.

[7] N.Abas, A.R.Kalair, N.Khan, A.Haider, Z.Saleem, M.S.Saleem, "Natural and synthetic refrigerants, global warming: A review.," *Renew. Sustain. Energy Rev*., 90, 557–569, 2018, doi:10.1016/j.rser.2018.03.099.

[8] S.Wang, C. Liu, Q. Li, L. Liu, E. Huo, C. Zhang,"Selection principle of working fluid for organic

Rankine cycle based on environmental benefits and economic performance," *Applied Thermal Engineering*., 178,115598, 2020, doi:10.1016/j.applthermaleng.2020.115598.

[9] J.Bao, L. Zhao, "A review of working fluid and expander selections for organic Rankine cycle," *Renew Sustain Energy Rev*., 24, 325–42, 2013, doi:10.1016/j.rser.2013.03.040.

[10] B. Saleh, "Parametric and working fluid analysis of a combined organic Rankine-vapor compression refrigeration system activated by low-grade thermal energy," *Journal of Advanced Research*., 7, 651–660, 2016, doi:10.1016/j.jare.2016.06.006.

[11] S. Aphornratana, T. Sriveerakul,"Analysis of a combined Rankine-vapour compression refrigeration cycle," *Energy Conversion and Management*., 51, 2557–2564, 2010, doi:10.1016/j.enconman.2010.04.016.

[12] M. Asim, M.K.H. Leung, Z. Shan, Y. Li, D.Y.C. Leung, M. Ni, "Thermodynamic and thermo-economic analysis of integrated organic Rankine cycle for waste heat recovery from vapor compression refrigeration cycle," *Energy Procedia*, 143, 192–198, 2017, doi:10.1016/j.egypro.2017.12.670.

[13] F. Molés, J.Navarro-Esbri, B.Peris, A.Mota-Babiloni, K. Kontomaris, "Thermodynamic analysis of a combined organic Rankine cycle and vapor compression cycle system activated with low temperature heat sources using low GWP fluids," *Applied Thermal Engineering*., 87, 444–453, 2015, doi:10.1016/j.applthermaleng.2015.04.083.

[14] H. Li, X. Bu, L. Wang, Z. Long, Y. Lian, "Hydrocarbon working fluids for a Rankine cycle powered vapor compression refrigeration system using low-grade thermal energy," *Energy Build*., 65, 167–172, 2013, doi:10.1016/j.enbuild.2013.06.012.

[15] X.Bu, L.Wang, H.Li, "Performance analysis and working fluid selection for geothermal energy-powered organic Rankine-vapor compression air conditioning," *Geotherm Energy*., 1,1–14, 2013, doi:10.1186/2195-9706-1-2.

[16] H.Wang, R.Peterson, K.Harada, E.Miller, R.Ingram-Goble, L.Fisher, "Performance of a combined organic Rankine cycle and vapor compression cycle for heat activated cooling," *Energy*., 36, 447–458, 2011, doi:10.1016/j.energy.2010.10.020.

[17] M.O. Nazer, S.M. Zubair, "Analysis of Rankine cycle air-conditioning systems," *ASHRAE Journal*., 88, 332–334, 1982.

[18] A.N. Egrican, A. Karakas, "Second law analysis of a solar powered Rankine cycle/vapor compression cycle," *Journal of Heat Recovery Systems*., 6, 135–141, 1986, doi:10.1016/0198-7593(86)90073-1.

[19] K.H.Kim, H.Perez-Blanco, 2015, "Performance analysis of a combined organic Rankine cycle and vapor compression cycle for power and refrigeration cogeneration," *Applied Thermal Engineering*., 91, 964–974, 2015, doi:10.1016/j.applthermaleng.2015.04.062.

[20] J. Jeong, Y.T. Kang, "Analysis of a refrigeration cycle driven by refrigerant steam turbine," *International Journal of Refrigeration*., 27, 33–41, 2004, doi:10.1016/S0140-7007(03)00101-4.

[21] B. Hu, J.Guo, Y.Yang, Y.Shao, "Performance analysis and working fluid selection of organic Rankine steam compression air conditioning driven by ship waste heat," *Energy Reports*., 8, 194–202, 2022, doi:10.1016/j.egyr.2022.01.094.

[22] S. Khatoon, N.M.A. Almefreji, M.H.Kim, "Thermodynamic study of a combined power and refrigeration system for low-grade heat energy source," *Energies*., 14, no. 2, 410, 2021, doi:10.3390/en14020410.

[23] E. Cihan and B. Kavasogullari, "Energy and exergy analysis of a combined refrigeration and waste heat driven organic rankine cycle system," *Thermal science*., 21, no. 6A, 2621–2631, 2017, doi: 10.2298/TSCI150324002C.

[24] B. Kavasogullari, E. Cihan, H. Demir, "Energy and Exergy Analyses of a Refrigerant Pump Integrated Dual-Ejector Refrigeration (DER) System," *Arabian Journal for Science and Engineering*., 46, 11633–11644, 2021, doi:10.1007/s13369-021-05541-7.

[25] E.O.Küçük and M. Kılıç, "Exergoeconomic and Exergetic Sustainability Analysis of a Combined Dual-Pressure Organic Rankine Cycle and Vapor Compression Refrigeration Cycle," *Sustainability*., 15, no. 8, 6987, 2023, doi: 10.3390/su15086987.

[26] A.K.S. Al-Sayyab, A. Mota-Babiloni, J.Navarro-Esbrí, "Performance Evaluation of Modified Compound Organic Rankine-Vapour Compression Cycle with Two Cooling Levels, Heating, and Power Generation," *Appl Energy*., 334, 120651, 2023, doi: 10.1016/j.apenergy.2023.120651.

[27] R. Zhar, A.Allouhi, M.Ghodbane, A.Jamil, K.Lahrech, "Parametric analysis and multi-objective optimization of a combined Organic Rankine Cycle and Vapor Compression Cycle," *Sustainable Energy Technologies and Assessments*., 47, 101401, 2021, doi:10.1016/j.seta.2021.101401.

[28] Z.Wang, Y.Zhao, X.Xia, S.Zhang, Y.Xiao, X.Zhang, W.Chen, "Experimental study of the thermodynamic performance of the ORC-VCC system with a zeotropic mixture," *Applied Thermal Engineering*., 250, 123534, 2024, doi: 10.1016/j.applthermaleng.2024.123534.

[29] D.M. Ginosar, L.M. Petkovic, D.P.Guillen,"Thermal Stability of Cyclopentane as an Organic Rankine Cycle Working Fluid," *Energy Fuels*., 25, 9, 4138–4144, 2011, doi:10.1021/ef200639r.

[30] Y.Maalem, Y.Tamene, H.Madani, "Behavior of the thermo-physical properties and performance evaluation of the refrigerants blends of (Fluorocarbon/Hydrocarbon) for cooling cycle," *Recueil de Mécanique*., 6, 544-559, 2022, doi:10.5281/zenodo.5918533.

[31] J.S. Lim, J.Y. Park, J.W. Kang, B.G. Lee, "Measurement of vapor–liquid equilibria for the binary systems of propane +1,1,1,2-tetrafluoroethane and 1,1,1-trifluoroethane+propane at various temperatures," *Fluid Phase Equilibria*., 243, 57-63, 2006, doi:10.1016/j.fluid.2006.02.016.

[32] S.Bobbo, R.Stryjek, N.Elvassore, A.Bertucco, “A recirculation apparatus for vapor–liquid equilibrium measurements of refrigerants. Binary mixtures of R600a, R134a and R236fa,” *Fluid Phase Equilibria*., 150, 343-352, 1998, doi:10.1016/S0378-3812(98)00334-3.

[33] Q.N. Ho, B.G. Lee, J.Y. Park, J.D. Kim, J. S. Lim,“Measurement of vapor–liquid equilibria for the binary mixture of propylene (R-1270)+1,1,1,2-tetrafluoroethane (HFC-134a),” *Fluid Phase Equilibria*., 225, 125-132, 2004, doi:10.1016/j.fluid.2004.08.028.

[34] L. Fedele, S. Bobbo, R. Camporese, M. Scattolini, “Isothermal vapour+liquid equilibrium measurements and correlation for the pentafluoroethane+cyclopropane and the cyclopropane+1,1,1,2-tetrafluoroethane binary systems,” *Fluid Phase Equilibria*., 251, 41-46, 2007, doi:10.1016/j.fluid.2006.10.023.

[35] N. Lim, G.Seong, H.Roh, “Vapor-Liquid Equilibria for the 1,1,1,2-Tetrafluoroethane (HFC-134a)+n-Butane (R-600) System,” *J. Chem. Eng. Data*., 52, 1313-1318, 2007, doi:10.1021/je700041v.

[36] Y. Maalem, S. Fedali, H. Madani, Y. Tamene, “Performance analysis of ternary azeotropic mixtures in different vapor compression refrigeration cycles,” *International Journal of Refrigeration*., 119, 139–151, 2020, doi:10.1016/j.ijrefrig.2020.07.021.