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Exergoeconomic and Exergetic Sustainability Analysis of a Combined Dual-Pressure Organic Rankine Cycle and Vapor Compression Refrigeration Cycle

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Abstract: In this paper, an exergoeconomic and exergetic sustainability analysis of a dual-pressure organic Rankine cycle (ORC) and vapor compression refrigeration cycle (VCRC) driven by waste heat is performed for power generation and cooling production. In addition, the most suitable fluid couple among the thirty-five different fluid pairs was investigated for the proposed combined system. The results indicate that the highest energy utilization factor, exergy efficiency, the system coefficient of performance, and net power are calculated for the R123-R141b fluid pair. In terms of exergetic sustainability indicators, the best performance results are obtained for the R123-R141b fluid combination. The minimum unit electricity generation cost and the shortest payback period are calculated as 0.0664 \$/kWh and 2.5 years, respectively, for the R123-R290 fluid pair. The system component with the highest exergy destruction is the boiler, with 21.67%. The result of the parametric analysis showed that the thermodynamic performance parameters increase with the increment of the ORC's boiling temperature. In addition, with the increasing boiling temperature, the environmental effect factor of the system decreases, while the exergetic sustainability index increases. Additionally, as the boiling temperature increases, the total system cost increases, while the unit electricity production cost and payback period decrease. It is suggested to use a R123-R141b fluid couple among fluid pairs created as a result of thermodynamic, exergoeconomic and sustainability analysis.

Keywords: dual-pressure ORC; refrigeration cycle; fluid pair; exergetic sustainability; combined cycle



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1. Introduction

The consumption of fossil fuels, which has led to increased greenhouse gas emissions in recent years, has increased significantly due to worldwide industrialization and population explosion. This situation has led us to pay attention to energy efficiency and energy recovery in thermal systems. For this reason, alternative systems using less energy or renewable energy sources have become widespread. One method of utilizing renewable energy sources and waste heat is an organic Rankine cycle (ORC), a suitable technology for converting low-grade heat sources into power. It is completely identical to a Rankine cycle, except that an organic fluid is used instead of water as the working fluid. Since the boiling temperatures of organic fluids are low, geothermal reservoirs, solar energy, biomass combustion, and industrial waste heat can be used as heat sources in the boiler of the ORC [1–4].

ORC has attracted considerable attention from many researchers working on ORC system-based design, optimization and operating conditions in the literature [5–8]. Higher thermal efficiency can be achieved by using different ORC configurations in energy conversion systems. Therefore, several alternative system architectures have been proposed as enhancements to the basic ORC [9–13]. In addition to that, a large number of working fluids have been analyzed by several researchers for several ORC configurations [3,14–16].

Globally increasing cooling demands consumes a considerable amount of primary energy sources. A major problem with sustainability is the extraordinary global rise in cooling needs [17]. Cooling and refrigeration requirements are mostly handled by mechanical compression systems (VCRs). While the vapor-compression refrigeration cycle provides high cooling power with relatively lower initial and operational costs, it also has a small structure with high efficiency. This makes it preferable to other cooling methods [18]. However, these systems are driven by primary energy. A potential method that intends to achieve refrigeration by using the shaft power of a turbine as a compressor's input power is the integration of ORC with VCR. Combined ORC-VCR systems have been the subject of numerous research projects. The ORC can be integrated with a VCRC to generate electricity and cooling. Recently, improvements of the system performance for the combined ORC-VCRC based on working fluid selection, parameter optimization, and development of different cycle configurations have received increasing attention [5,19]. Some researchers used common condensers and working fluids for integrated ORC-VCRC in their studies. For example, Bu et al. [20] performed an ORC-VCRC system using waste heat from hot springs for air conditioning. They selected the refrigerants R123, R134a, R245fa, R600a, R600, and R290 to find the best one for the ORC-VCRC. Their results show that R600a is the most suitable working fluid. Aphornratana and Sriveerakul [21] performed a thermodynamic analysis using R134a and R22 fluids, developing a new combined system in which two cycles are connected by a device called an expander-compressor unit. The results obtained showed that the coefficient of performance (*COP*) value for the combined system ranged from 0.1 to 0.6. Additionally, the system with R22 provided better performance than the system with R134a. Cihan and Kavasoğulları [22] determined the most suitable fluid using R123, R600, R600a, R245fa and R141b working fluids in the combined ORC-VCRC system. They also investigated the change in the overall system performance coefficient depending on the changing boiling, evaporation and condensation temperatures. As a result of the energy and exergy analysis of the proposed system, it was determined that the most suitable organic fluid was R141b. Saleh and Aly [15] selected fourteen different working fluids for the energy and exergy analysis. They investigated the influences of the boiler, condenser, evaporator temperatures, and compressor and expander efficiencies on ORC-VCRC system performance. Jeong and Kang [23] designed three systems: basic ORC-VCRC, ORC-VCRC with a recuperator; and ORC-VCRC with a recuperator, reheater, and economizer. The fluids used in the combined system were R123, R134a, and R245fa. According to the results obtained, R245ca was the most suitable refrigerant considering the system efficiency and environmental issues. In addition, the maximum *COP* was calculated for ORC-VCRC with a recuperator, reheater, and economizer among the configurations. Kim and Perez-Blanco [24] investigated the effects of turbine inlet pressure, turbine inlet temperature, and the flow division ratio on the system performance for eight refrigerants. Pektezel and Acar [25] reported a comparative study on the combined ORC cycle with either single or dual evaporator VCR cycles. They concluded that the combined ORC-single evaporator VCR shows better performance than the other one. They also observed that R600a was the best refrigerant for their proposed combined cycles. Graubeger et al. [26] presented an experimental and numerical study on the ORC-VCR combined system using R1234ze(E) as the working fluid. In their study, a prototype ORC-VCR was designed, built, and tested at the standard rating conditions. They observed that the thermal *COP* was 0.66. Graubeger et al. [27] also reported the off-design performance of the their system. They stated that increasing the driving heat source inlet temperature improved the *COP* of the system and the thermal efficiency of the ORC power cycle, while decreasing the efficiency of the compressor and the *COP* of the VCR. Xia et al. [28] reported an advanced exergy analysis of an ORC-VCR combined system. They analyzed the exergy destruction distribution characteristics and improvement potential of the components. They reported that the improvement depends on themselves for the turbine and compressor. However, the improvement depends on both themselves and other components for the generator, condenser and evaporator.

The two-fluid system utilizes different refrigerants in the power cycle and cooling cycle of the combined cycle. According to research conducted by Wang et al. [29], the two-fluid system has several advantages over the single-fluid system. A two-fluid system can provide higher efficiency than a single-fluid cycle of the same weight. It is possible to create the most suitable temperature in condensers by using different fluids in the power and cooling cycle and by being separate systems. This makes the system lighter and more compact. In addition, different lubricants can be used in each section according to the operation conditions of the expander and compressor. There have been some analyses on the energy and exergy analysis for a combined system using ORC modification with a two-fluid system. In the study by Wang et al. [29], the impacts of various cycle configurations on system performance were investigated. Analysis results showed that a combined cycle with sub-cooling and cooling recuperation resulted in a 22% improvement in overall COP compared to the basic combined cycle. In addition, both adding a secondary recuperator to the ORC and using different working fluids in the power and cooling cycles provided higher COP and practical advantages. Toujeni et al. [30] developed a new combined ORC-VCRC configuration. The designed system includes two regenerator and evaporator-condenser systems. They selected R600a and ammonia for the ORC and the VCRC systems, respectively. Their results showed that the use of two regenerators improved the system's performance. It has been determined that this system's performance depends on the temperature of the heat source and the cold temperature planned to be produced. Wang et al. [31] developed two configurations of combined power and cooling systems, which are basic ORC-VCRC and dual-pressure ORC-VCRC. They chose R245fa and butane as the working fluids for the power and cooling cycle, respectively. Their results demonstrate that the dual-pressure ORC-VCRC system can achieve a 7.1% increase in thermal efficiency and a 6.7% increase in exergy efficiency over the basic ORC-VCRC. Molès et al. [32] examined a low-grade thermal energy-activated ORC-VCR system using two different GWP working fluids for cooling and power cycles. Their results show that the working fluid selection in the refrigeration cycle is less effective on the system efficiency. HFO-1234ze(E) and HFO-1336mzz(Z) offer a higher system performance for cooling and power cycles, respectively. Wang et al. [33] investigated a theoretical and experimental evaluation of a combined ORC-VCRC to produce cooling based on the first and second laws of thermodynamics. They developed a prototype system with a nominal cooling capacity of 5 kW. The dual fluid system was chosen because of utilizing the smaller and lighter portable system. R245fa and R134a were used as the working fluid for the power cycle and the refrigerant for the cooling cycle, respectively. In the study published by Özdemir Küçük and Kiliç [34], three different combined system configurations: basic ORC-VCRC, two-fluid basic ORC-VCRC and ORC-VCRC with an internal heat exchanger (IHE) and liquid-vapor heat exchanger (LVHE) were developed and compared. According to the results obtained, the best performance results were achieved with the third design, which is the dual fluid configuration with IHE and LVHE. In addition, thirty different fluid pair combinations were evaluated for this configuration and the most suitable fluid pair was determined.

The non-combustion-based renewable electricity production methods were evaluated using data gathered from the literature and a variety of sustainability parameters. The cost of the power generated, environmental impacts, life cycle analysis (LCA), the availability of renewable or waste energy sources, and the effectiveness of energy conversion were the metrics used to evaluate each technology [35]. It was discovered that there was a very broad range in the price of power, greenhouse gas emissions, and the efficiency of electricity generation for each technology [36]. The indicators were then compared to renewable energy technologies, with the assumption being that they are all equally important for sustainable development [37]. The large initial investment cost is frequently a barrier to the combined systems' widespread adoption. Depending on a variety of conditions, the economy of coupled systems may fluctuate significantly. Hence, in addition to energy-environmental considerations, economic feasibility assessments are important and can offer investors financial rewards. Specific exergy cost (SPECO) methodology is one of the exergy-

based economics approaches [38,39]. Exergy analysis is another tool used to evaluate and assess how sustainable an energy system is [40]. More environmentally friendly technology is possible when thermal system efficiency gains are realized and thermodynamic losses at the plant are decreased [41,42]. Exergy efficiency, waste exergy ratio (WER), exergy destruction factor (EDF), environmental effect factor (EEF), and exergetic sustainability index (ESI) are systems' exergetic sustainability metrics [43].

It can be seen from the current literature summarized above that the determination of working fluids used in the specific configuration of a combined organic Rankine cycle and vapor compression refrigeration cycle system has an essential influence on the thermodynamic performance of the ORC and VCRC. It can be observed that integrating ORC cycles with a VCRC has led to a significant increase in the energy and exergy performance of the basic system. In light of the above considerations, the novelty and main contribution of the present work are as follows:

First, a novel concept of the dual-pressure organic Rankine cycle integrated into a vapor-compression refrigeration cycle to generate electric power and cooling is considered in this study. Unlike the literature, in the designed combined system, the ORC has two separate expanders operating high and low pressure conditions. The high-pressure expander is used to generate electric power, while the low-pressure one is used to drive the compressor of the VCRC. For this reason, both electric power generation and a cooling effect are produced with the proposed combined system. Additionally, the ORC and VCRC are separate from each other, and the working fluids circulating in the power and cooling cycles are different.

As a second contribution to the current literature, the exergetic sustainability indicators are employed to evaluate the environmental impact of the proposed system.

Third, an exergoeconomic analysis method is introduced to evaluate the economic parameters with respect to the energy and exergy concept of the suggested system.

Finally, the exergoeconomic and the exergetic sustainability analyses of the combined ORC-VCRC system are performed to evaluate the performance of thirty-five different fluid pairs comparatively. Then, a parametric study is carried out to investigate the effect of the boiler temperature on the performance of the combined dual-pressure ORC-VCRC system by using the selected fluid pairs.

2. Materials and Methods

2.1. System Description and Assumptions

The novel combined power and cooling system mainly consists of a dual-pressure organic Rankine cycle and vapor compression refrigeration cycle. The schematic diagram of this system is shown in Figure 1. The shaft of the low-pressure expander in the dual-pressure ORC and the shaft of the compressor in the VCRC is directly coupled to reduce two-way energy losses. The components of a dual-pressure organic Rankine cycle (1-2-3-4-5-6-7-8-9-10-11-1) are a boiler, a high-pressure expander, a low-pressure expander, a recuperator, a regenerator, a condenser, and two pumps. A vapor compression refrigeration cycle (16-17-18-19-16) consists of a compressor, an evaporator, a condenser, and a throttling valve. The dual-pressure ORC and VCRC systems use different working fluids to improve the combined system of performance [34,44]. The first part of power generation capacity, which is produced by the high-pressure expander, is used for electricity generation in the combined system. The second power production, which is used to activate the compressor, takes place in the low-pressure expander.

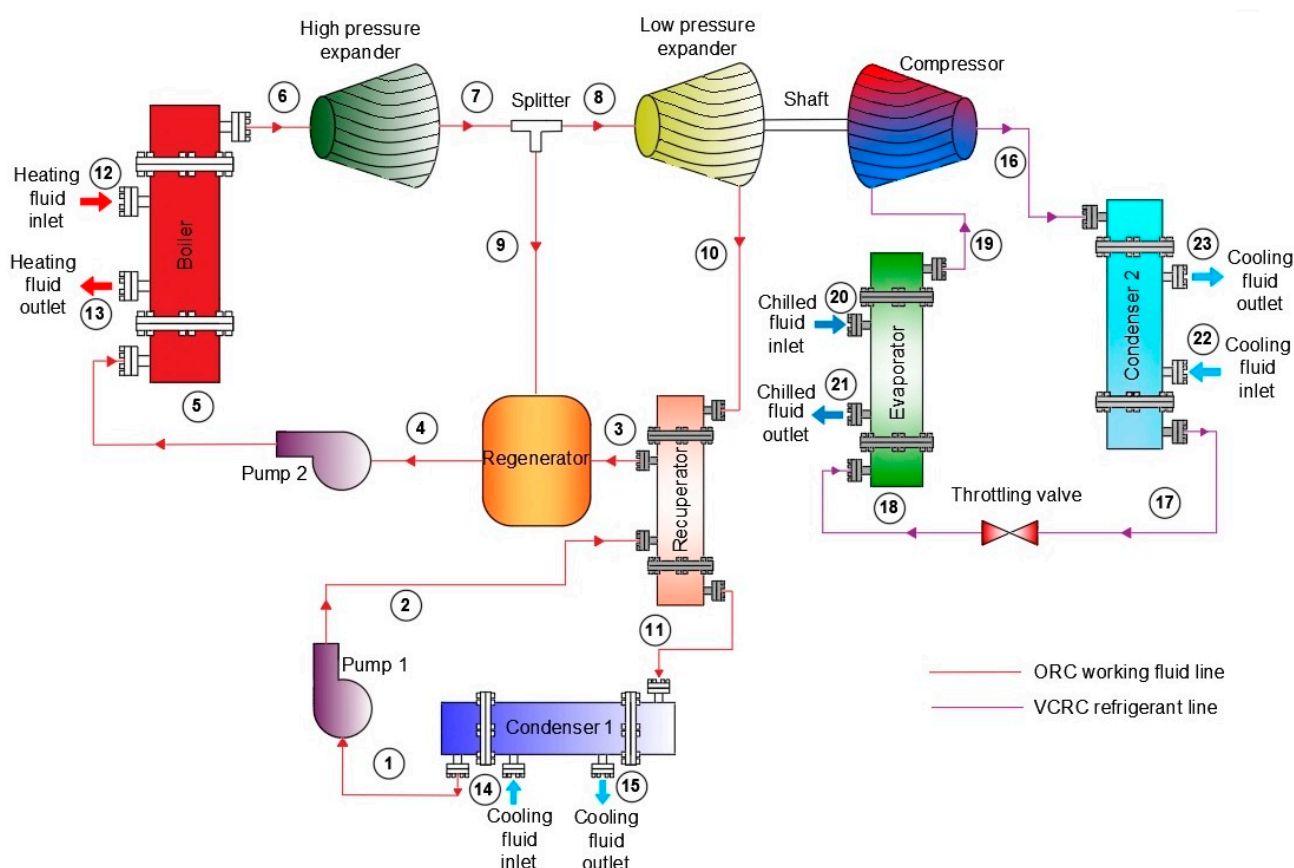


Figure 1. The schematic diagram of the dual-pressure ORC-VCRC.

The working fluid in condenser 1 transfers its heat to cooling fluid, which enters condenser 1 of the ORC at the temperature T_{14} , and leaves condenser 1 at the condenser pressure (state 1). The exiting stream from condenser 1 enters pump 1, in which the pressure is raised to recuperator pressure (state 2) and is routed to the recuperator. The working fluid in the recuperator receives the heat of vapor exiting from the low-pressure expander (state 3), and enters the regenerator. The working fluid leaves the regenerator as a saturated liquid after mixing with the vapor from the high-pressure turbine (state 4). The pressure of the fluid entering pump 2 rises to the boiler pressure (state 5). Then, the hot fluid entering the boiler at the temperature T_{12} transfers its heat to the working fluid. The working fluid leaves the boiler in the form of overheated vapor, and the boiler pressure enters the high-pressure expander (state 6), which expands isentropically to an intermediate pressure (state 7). In the meantime, electricity is generated by the generator connected to the high-pressure expander. The exiting stream from the high-pressure expander is then divided into two separate streams. Part of the working fluid is routed to the regenerator (state 9), while the other part (state 8) enters the low-pressure expander and its pressure drops to the condenser pressure (state 10). The vapor entering the recuperator gives its heat to the organic fluid leaving pump 1 (state 11). The dual-pressure ORC repeats the cycle in this sequence. In the VCRC, which uses the energy produced in the low-pressure expander in its compressor, the refrigerant entering the compressor as evaporator pressure and overheated vapor (state 19) is compressed isentropically to the condenser 2 pressure (state 16). The refrigerant enters condenser 2 as overheated vapor, and transfers its heat to the cooling fluid at temperature T_{22} . The super-cooled working fluid exiting condenser 2 (state 17) is throttled to the evaporator pressure by passing it through a throttling valve at constant enthalpy (state 18). The working fluid enters the evaporator as a low-quality saturated mixture, completely evaporating by absorbing heat from the refrigerated space. It exits the evaporator and re-enters the compressor, completing the cycle (state 19).

The temperature-specific entropy (T-s) diagrams of the dual-pressure ORC and VCRC are presented in Figure 2.

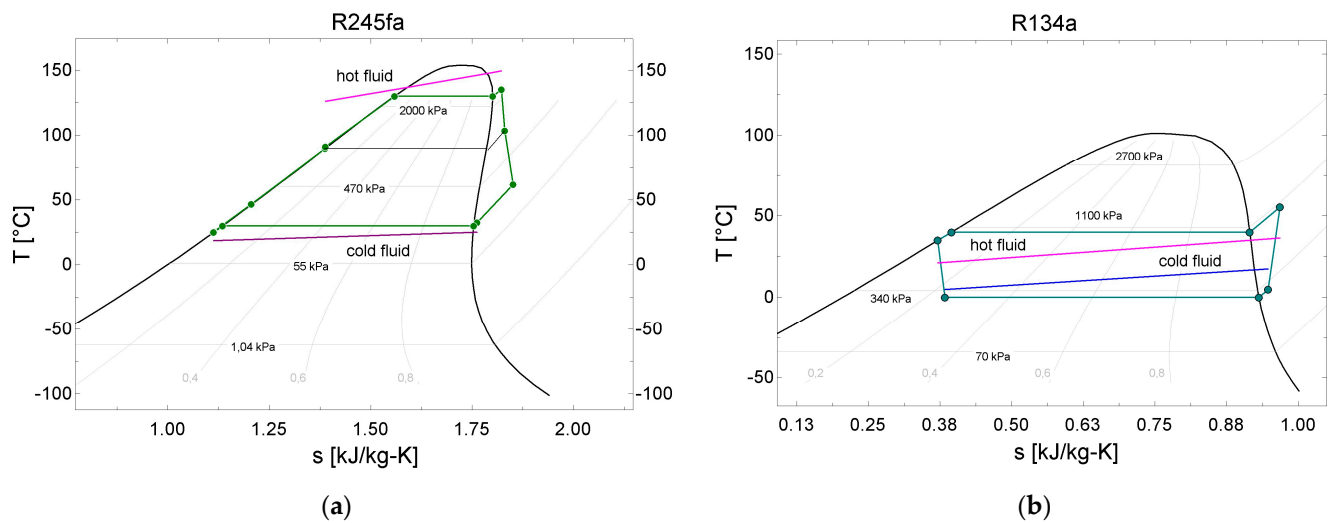


Figure 2. (a) T-s diagram of the dual-pressure ORC. (b) T-s diagram of the VCRC.

The thermodynamic analysis of the dual-fluid ORC-VCRC described is made theoretically in the previous study [34]. To perform the thermodynamic analysis of the ORC-VCRC configuration, the following simplifications and approximations are made:

- The designed combined system works in a steady state.
- There are negligible kinetic and potential energy changes.
- Pressure losses in the heat exchangers are negligible.
- The dead state condition is taken as 101 kPa and 30 °C.
- Expansion and compression processes in the expanders, pumps, and compressor are adiabatic.

The assumptions made for modeling and simulating the combined dual-fluid ORC-VCRC system are given in Table 1.

Table 1. Thermodynamic assumptions for the combined system.

Parameter	Value	Unit
The boiling temperature in the boiler	130	°C
Hot fluid heat rate	200	kW
The evaporation temperature in the VCRC evaporator	0	°C
The condensation temperature in ORC condenser	40	°C
The condensation temperature in VCRC condenser	40	°C
Hot source and cold sink pressures	101	kPa
The mass flow rate of hot fluid in the boiler	20	kg/s
Expanders isentropic efficiency	0.8	
Pumps isentropic efficiency	0.8	
Compressor isentropic efficiency	0.8	
Recuperator effectiveness	0.8	
Pinch-point temperature difference in heat exchangers	5	°C
Overheating in the boiler and evaporator	5	°C
Sub-cooling of the condensers	5	°C

The thermophysical and environmental properties of the analyzed fluids in the combined system are given in Table 2. These selected fluids are among the most used in both separate ORC and VCRC systems and combined ORC-VCRC power plants [14,45,46]. In the analysis, the working fluids used in the ORC system are R114, R123, R600, R600a, and

R245fa. In the subsystem VCRC, R141b, R600a, R290, R134a, R123, R245fa and R143a are applied as a refrigerant.

Table 2. Thermo-physical and environmental properties of the selected fluids.

Fluids	Molecular Mass	Formula	Boiling Temperature	Critical Temperature	Critical Pressure	Critical Density	ODP	GWP 100 yr
	g/mol		°C	°C	MPa	kg/m ³		
R245fa	134.05	C ₃ F ₅ H ₃	15.18	154.05	3.65	519.43	0.00	1030.0
R600	58.12	C ₄ H ₁₀	−0.52	152.01	3.80	228.00	0.00	4.0
R114	170.92	C ₂ Cl ₂ F ₄	3.79	145.68	3.26	579.97	1.00	3.9
R600a	58.12	C ₄ H ₁₀	−11.85	134.70	3.63	225.50	0.00	3.0
R141b	116.95	C ₂ Cl ₂ FH ₃	32.06	204.35	4.21	458.56	0.00	713.0
R290	44.09	C ₃ H ₈	−42.10	96.70	4.25	220.48	0.00	3.3
R123	152.93	C ₂ Cl ₂ F ₃ H	27.78	183.68	3.66	550.00	0.02	77.0
R134a	102.03	C ₂ F ₄ H ₂	−26.09	101.06	4.06	511.90	0.00	1430.0
R143a	84.04	C ₃ F ₃ H ₃	−47.24	72.71	3.76	431.00	0.00	4300.0

2.2. Thermodynamic Analysis

In this section, mass, energy, and exergy balance equations are shared for each element of the designed combined ORC-VCRC system. First, the general conservation of mass equation for each cycle element is expressed by the following equation.

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

where $\dot{m}$ is mass flow rate. The “in” and “out” subscripts indicate the input and output of the system component, respectively. For any system, energy and exergy balance equations can be expressed by the following equations.

$$\dot{Q} - \dot{W} + \sum \dot{m}_{in}h_{in} - \sum \dot{m}_{out}h_{out} = 0 \quad (2)$$

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} + \sum \dot{Ex}_{heat} + \sum \dot{W} - \dot{Ex}_{dest} = 0 \quad (3)$$

where $\dot{Q}$ is heat transfer rate, $\dot{W}$ is power, h is specific enthalpy, $\dot{Ex}$ is exergy rate, and $\dot{Ex}_{dest}$ is the exergy destruction rate. The physical specific exergy of the i th point at a given state is expressed as:

$$ex_i^{ph} = (h_i - h_0) - T_0(s_i - s_0) \quad (4)$$

where h_i and s_i denote specific enthalpy and specific entropy of the i th state, respectively. In addition, h_0 and s_0 are specific enthalpy and specific entropy of the i th state in the ambient environment, respectively.

Chemical exergy becomes important when there is a chemical reaction (such as combustion), a mixture of components or a phase change in the system. The total chemical exergy in a multi-gas mixture is calculated by the following equation [45].

$$ex_i^{ch} = M_i \left[\sum_{i=1}^n x_i \bar{ex}_i^{ch} + RT_0 \sum_{i=1}^n x_i \ln(x_i) \right] \quad (5)$$

where, M_i is the molecular weight of species i , $\bar{ex}_i^{ch}$ is the standard chemical exergy value of species i , x_i is a mole fraction of species i , and R is the universal gas constant. In this study, only physical exergy is considered, ignoring chemical exergy.

The energy and exergy balances for all system components are written in detail and presented in Table 3.

Table 3. Energy and exergy relations for the proposed cycle.

Component	Energy Analysis	Exergy Analysis
Pump 1	$\dot{W}_{p1} = \dot{m}_{lpe}(h_2 - h_1)$	$\dot{E}x_1 + \dot{W}_{p1} = \dot{E}x_2 + \dot{I}_{p1}$
Recuperator	$\dot{m}_{lpe}(h_3 - h_2) = \dot{m}_{lpe}(h_{10} - h_{11})$	$\dot{E}x_2 + \dot{E}x_{10} = \dot{E}x_3 + \dot{E}x_{11} + \dot{I}_{rec}$
Regenerator	$\dot{m}_{ORC}h_4 = \dot{m}_{hpe}h_9 + \dot{m}_{lpe}h_3$	$\dot{E}x_3 + \dot{E}x_9 = \dot{E}x_4 + \dot{I}_{reg}$
Pump 2	$\dot{W}_{p2} = \dot{m}_{ORC}(h_5 - h_4)$	$\dot{E}x_4 + \dot{W}_{p2} = \dot{E}x_5 + \dot{I}_{p2}$
Boiler	$\dot{Q}_b = \dot{m}_{ORC}(h_6 - h_5) = \dot{m}_{12}(h_{12} - h_{13})$	$\dot{E}x_5 + \dot{E}x_{12} = \dot{E}x_6 + \dot{E}x_{13} + \dot{I}_b$
High-pressure expander	$\dot{W}_{hpe} = \dot{m}_{ORC}(h_6 - h_7)$	$\dot{E}x_6 = \dot{E}x_7 + \dot{W}_{hpe} + \dot{I}_{hpe}$
Low-pressure expander	$\dot{W}_{lpe} = \dot{m}_{lpe}(h_8 - h_{10})$	$\dot{E}x_8 = \dot{E}x_{10} + \dot{W}_{lpe} + \dot{I}_{lpe}$
Condenser 1	$\dot{Q}_{c1} = \dot{m}_{lpe}(h_{11} - h_1) = \dot{m}_{14}(h_{15} - h_{14})$	$\dot{E}x_{11} + \dot{E}x_{14} = \dot{E}x_1 + \dot{E}x_{15} + \dot{I}_{cd1}$
Compressor	$\dot{W}_{comp} = \dot{m}_{VCRC}(h_{17} - h_{16})$	$\dot{E}x_{16} + \dot{W}_{comp} = \dot{E}x_{17} + \dot{I}_{comp}$
Condenser 2	$\dot{Q}_{c2} = \dot{m}_{VCRC}(h_{17} - h_{18}) = \dot{m}_{VCRC}(h_{23} - h_{22})$	$\dot{E}x_{17} + \dot{E}x_{22} = \dot{E}x_{18} + \dot{E}x_{23} + \dot{I}_{cd2}$
Throttling valve	$h_{18} = h_{19}$	$\dot{E}x_{18} = \dot{E}x_{19} + \dot{I}_{tv}$
Evaporator	$\dot{Q}_{ev} = \dot{m}_{VCRC}(h_{16} - h_{19}) = \dot{m}_{20}(h_{20} - h_{21})$	$\dot{E}x_{19} + \dot{E}x_{20} = \dot{E}x_{16} + \dot{E}x_{21} + \dot{I}_{ev}$

The net power output and the thermal efficiency of the dual-pressure ORC system are given in Equations (6) and (7), respectively.

$$\dot{W}_{net,ORC} = (\dot{W}_{hpe} + \dot{W}_{lpe}) - (\dot{W}_{p1} + \dot{W}_{p2}) \quad (6)$$

$$\eta_{ORC} = \dot{W}_{net,ORC} / \dot{Q}_b \quad (7)$$

where $\dot{W}_{hpe}$ and $\dot{W}_{lpe}$ represent high-pressure expander power and low-pressure expander power, respectively. $\dot{W}_{p1}$ and $\dot{W}_{p2}$ are the amount of power consumed by pump 1 and pump 2. $\dot{Q}_b$ is the heat transfer rate in the boiler of dual-pressure ORC. The coefficient of performance of refrigeration cycle (COP), which expresses the performance of the cooling cycle, and the overall system coefficient of performance (COP_s), which is the expression of the combined cycle's cooling performance, are calculated with the following equations [21,22].

$$COP = \dot{Q}_{ev} / \dot{W}_{comp} \quad (8)$$

$$COP_s = \eta_{ORC} COP \quad (9)$$

The overall energy efficiency of the combined system is expressed with an energy utilization factor (EUF) because of the presence of more than one product [43,47–49]. The net output power and the thermal efficiency of the dual-pressure ORC-VCRC system are calculated as follows:

$$\dot{W}_{net} = (\dot{W}_{hpe} + \dot{W}_{lpe}) - (\dot{W}_{p1} + \dot{W}_{p2}) - \dot{W}_{comp} \quad (10)$$

$$EUF = \frac{\dot{W}_{net} + \dot{Q}_{cooling}}{\dot{Q}_b} \quad (11)$$

where $\dot{W}_{comp}$ is compressor power. $\dot{Q}_{cooling}$ is the produced cooling provided by the evaporator of the VCRC. Exergy efficiency of the designed system can be calculated as below [24]:

$$\eta_{ex} = \frac{\dot{W}_{net} + \dot{E}x_{cooling}}{\dot{E}x_{in}} \quad (12)$$

where $\dot{E}x_{cooling}$ refers to the exergy rate of the cooling realized by the evaporator in the VCRC of the combined system, while $\dot{E}x_{in}$ refers to the exergy rate of the heat entering the boiler in the power cycle of the combined system. The equations of $\dot{E}x_{cooling}$ and $\dot{E}x_{in}$ parameters are written with Equations (13) and (14) below, respectively.

$$\dot{E}x_{cooling} = \dot{E}x_{21} - \dot{E}x_{20} = \dot{m}_{21}[(h_{21} - h_{20}) - T_0(s_{21} - s_{20})] \quad (13)$$

$$\dot{E}x_{in} = \dot{E}x_{12} - \dot{E}x_{13} = \dot{m}_{12}[(h_{12} - h_{13}) - T_0(s_{12} - s_{13})] \quad (14)$$

2.3. Exergetic Sustainability Indicators

Exergy analysis is also used to determine and analyze the sustainability level of energy systems. Realization of efficiency improvements in a thermal system and reductions in thermodynamic losses in the plant means more sustainable technology. Exergy sustainability indicators of a system are exergy efficiency, waste exergy ratio (WER), exergy destruction factor (EDF), environmental effect factor (EEF) and exergetic sustainability index (ESI) [37,41–43].

Waste exergy ratio (WER) of the combined system can be expressed as the ratio of the total waste exergy to the total inlet exergy, and WER can be calculated by Equation (15).

$$\text{Waste exergy ratio (WER)} = \frac{\text{Total waste exergy out}}{\text{Total exergy inlet}} \quad (15)$$

where total waste exergy out ($\dot{E}x_{we,out}$) is the sum of both the destroyed exergy of the system component and the lost exergy of the system, as shown in Equation (16).

$$\sum \dot{E}x_{we,out} = \sum \dot{E}x_{dest,out} + \dot{E}x_{loss,out} \quad (16)$$

One of the indicators of exergetic sustainability is the exergy destruction factor (EDF), which is calculated by the ratio of total exergy destruction to total exergy input. This parameter is shown in Equation (17).

$$\text{Exergy destruction factor (EDF)} = \frac{\text{Total exergy destruction}}{\text{Total exergy inlet}} \quad (17)$$

Environmental effect factor (EEF) is the ratio of waste exergy to the exergy efficiency. The EEF indicates how much damage to the environment is due to waste exergy output and exergy destruction and is calculated as follows:

$$\text{Environmental effect factor (EEF)} = \frac{\text{Waste exergy ratio}}{\text{Exergy efficiency}} \quad (18)$$

Among exergetic sustainability indicators, the exergetic sustainability index (ESI) is a crucial component for determining the system's sustainability level. The ratio of unity to the environmental effect factor can be used to determine its function. Exergetic sustainability index (ESI) is calculated as given in Equation (19).

$$\text{Exergetic sustainability index (ESI)} = \frac{1}{\text{Environmental effect factor}} \quad (19)$$

2.4. Exergoeconomic Analysis

Specific exergy cost (SPECOC) methodology, which is one of the exergy-based economics approaches, was used in the economic analysis of the developed combined cycle. This method is applied to evaluate and improve various devices such as power plants. In the SPECOC method, the cost balance equation and auxiliary equations for each component of the cycle are defined.

The cost balance equation can be calculated as the cost rate associated with the product of the system ($\dot{C}_P$) equals the total rate of expenditures made to produce the product ($\dot{C}_F$) and the capital investment ($\dot{Z}^{CI}$) and operating and maintenance ($\dot{Z}^{OM}$) costs [40].

$$\dot{C}_{P,tot} = \dot{C}_{F,tot} + \dot{Z}_{tot}^{CI} + \dot{Z}_{tot}^{OM} \quad (20)$$

In exergy costing, a cost is associated with each exergy stream. Thus, for entering and exiting streams of matter with associated rates of exergy transfer $\dot{E}x_{in}$ and $\dot{E}x_{out}$, power $\dot{W}$ and the exergy transfer rate associated with heat transfer $\dot{E}x_q$, respectively, we can write following:

$$\dot{C}_{in} = c_{in}\dot{E}x_{in} \quad (21)$$

$$\dot{C}_{out} = c_{out}\dot{E}x_{out} \quad (22)$$

$$\dot{C}_w = c_w\dot{W} \quad (23)$$

$$\dot{C}_q = c_q\dot{E}x_q \quad (24)$$

where, c denotes the cost per unit of each exergy stream. A cost balance equation can be written as:

$$\sum \dot{C}_{out,k} + \dot{C}_{w,k} = \sum \dot{C}_{in,k} + \dot{C}_{q,k} + \dot{Z}_k \quad (25)$$

The levelized capital investment and operating and maintenance costs ($\dot{Z}_k$) for the k th component can be expressed as:

$$\dot{Z}_k = \frac{PEC_k CRF \varphi}{N 3600} \quad (26)$$

where PEC_k is the purchased equipment cost, CRF is the capital recovery factor, φ is the total operating and maintenance cost factor (1.06), and N is the operating time of the system in hours (7446 h) [50]. The CRF is written as follows:

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (27)$$

In Equation (27), i is the interest rate (10%) and n is the expected life of the system (20 years) [50]. The total system cost rate of the designed system is calculated by summing the total capital investment cost, the total exergy destruction cost and the cost of the lost exergy (Equation (28)) [51].

$$\dot{C}_{sys} = \sum_k \dot{Z}_k + \sum_k \dot{C}_{D,k} + \dot{C}_{L,k} \quad (28)$$

The unit cost of electricity produced by the intended system can be written as follows [52,53]:

$$C_{el} = \frac{PEC_{tot} CRF \varphi}{\dot{W}_{net} N} \quad (29)$$

The payback period of the developed integrated system is calculated with the Equation (30) [54].

$$PB = \frac{\log \frac{(\dot{W}_{net} N c_{el}) - PEC_{top}(\varphi - 1)}{(\dot{W}_{net} N c_{el}) - PEC_{top}(\varphi - 1) - (i PEC_{top})}}{\log(1 + i)} \quad (30)$$

where, c_{el} is the unit price of electricity and it is taken as 0.15 \$/kWh in the analysis [55].

For the exergoeconomic evaluation of the designed dual pressure ORC-VCRC system, the necessary cost balance equations and auxiliary equations for each system component are listed in Table 4. At the same time, the purchased equipment cost function statements of each system component are given in Table 5.

Table 4. The cost equations formulated for each component of the combined system.

Component	Cost Balance Equation	Auxiliary Equation
Pump 1	$c_1\dot{E}x_1 + c_{p1}\dot{W}_{p1} + \dot{Z}_{p1} = c_2\dot{E}x_2$	$c_{p1} = c_{hpe}$
Recuperator	$c_{10}\dot{E}x_{10} + c_2\dot{E}x_2 + \dot{Z}_{rec} = c_{11}\dot{E}x_{11} + c_3\dot{E}x_3$	$c_{10} = c_{11}$
Regenerator	$c_3\dot{E}x_3 + c_9\dot{E}x_9 + \dot{Z}_{reg} = c_4\dot{E}x_4$	
Pump 2	$c_4\dot{E}x_4 + c_{p2}\dot{W}_{p2} + \dot{Z}_{p2} = c_5\dot{E}x_5$	$c_{p2} = c_{hpe}$
Boiler	$c_5\dot{E}x_5 + c_{12}\dot{E}x_{12} + \dot{Z}_{ev} = c_6\dot{E}x_6 + c_{13}\dot{E}x_{13}$	$c_{12} = c_{13}$
High-pressure expander	$c_6\dot{E}x_6 + \dot{Z}_{hpe} = c_7\dot{E}x_7 + c_{hpe}\dot{W}_{hpe}$	$c_6 = c_7, c_7 = c_8, c_7 = c_9$
Low-pressure expander	$c_8\dot{E}x_8 + \dot{Z}_{lpe} = c_{10}\dot{E}x_{10} + c_{lpe}\dot{W}_{lpe}$	$c_{10} = c_8$
Condenser 1	$c_{11}\dot{E}x_{11} + c_{14}\dot{E}x_{14} + \dot{Z}_{c1} = c_1\dot{E}x_1 + c_{15}\dot{E}x_{15}$	$c_{11} = c_1, c_{14} = 0$
Compressor	$c_{16}\dot{E}x_{16} + c_{comp}\dot{W}_{p2} + \dot{Z}_{comp} = c_{17}\dot{E}x_{17}$	$c_{comp} = c_{hpe}$
Condenser 2	$c_{17}\dot{E}x_{17} + c_{22}\dot{E}x_{22} + \dot{Z}_{c2} = c_{18}\dot{E}x_{18} + c_{23}\dot{E}x_{23}$	$c_{17} = c_{18}, c_{22} = 0$
Throttling valve	$c_{18}\dot{E}x_{19} + \dot{Z}_{tv} = c_{19}\dot{E}x_{19}$	
Evaporator	$c_{19}\dot{E}x_{19} + c_{20}\dot{E}x_{20} + \dot{Z}_{ev} = c_{16}\dot{E}x_{16} + c_{21}\dot{E}x_{21}$	$c_{16} = c_{19}, c_{20} = 0$

Table 5. The purchased equipment cost functions of the system components.

Component	Purchased Equipment Cost	Reference
Pump	$PEC_p = 3540 (\dot{W}_p)^{0.71}$	[51]
Evaporator	$PEC_{ev} = 130 (A_{ev}/0.093)^{0.78}$	[51]
Expander	$PEC_{exp} = 4405 (\dot{W}_{exp})^{0.7}$	[51]
Condenser 1	$PEC_{c1} = 2143 (A_{c1})^{0.514}$	[56]
Recuperator	$PEC_{rec} = 2681 (A_{rec})^{0.59}$	[56]
Regenerator	$PEC_{reg} = 280.3 (\dot{m}_{reg})^{0.67}$	[36]
Compressor	$PEC_{comp} = \frac{39.5 \dot{m}_{VCRC}}{0.9 - \eta_{comp}} \left(\frac{P_{17}}{P_{16}} \right) \ln \left(\frac{P_{17}}{P_{16}} \right)$	[57]
Condenser 2	$PEC_{c2} = 268.45 + 516.621 A_{c2}$	[57]
Throttling valve	$PEC_{comp} = 114.5 \dot{m}_{VCRC}$	[57]
Evaporator	$PEC_{ev} = 16,648.3 (A_{ev})^{0.6123}$	[57]

In order to calculate the PEC values of the heat exchangers in the system, the surface areas of the heat exchangers must be calculated. According to the logarithmic temperature difference method, the surface areas are calculated with the following equation.

$$A_k = \dot{Q}_k / (U_k \Delta T_k^{lm}) \quad (31)$$

where $\dot{Q}_k$ is the heat rate of the heat exchanger, U_k is the heat transfer coefficient and ΔT_k^{lm} is the logarithmic mean temperature difference. The accepted heat transfer coefficient values for the heat exchangers of the system are given in Table 6.

Table 6. The total heat transfer coefficient of the heat exchangers.

Component	U (kW/m ² K)	Reference
Boiler	0.9	[49,58]
Condensers	1.1	[49,58]
IHE	0.7	[59]
Evaporator	1.5	[58]

3. Results and Discussion

3.1. Validation of the Present Model

The mathematical model for the proposed combined system is coded in the Engineering Equation Solver (EES, Professional 11), and the set of the defined equations in the previous section is solved in an iterative manner by the EES software. The thermodynamic properties of the working fluids are obtained by using the EES internal libraries. Although there is not any study considering the same configuration with the proposed combined system in this study, there are some studies considering combined ORC-VCRC systems in the current literature. Hence, two recently reported works about the combined systems by Saleh [60] and Graubeger et al. [26] are chosen for the validation of the present model. The configurations and operating conditions of these combined systems were taken as defined in the references. Table 7 shows the comparisons between the present calculation results and the data obtained from the references. It can be seen that the present mathematical model results showed a good agreement with the corresponding results from the works of Saleh [60] and Graubeger et al. [26]. As a result of this, the present mathematical model can be accepted as validated.

Table 7. The validation of the present model calculations.

Parameter	Graubeger et al. [26]	Present Model	Difference (%)	Saleh [60]	Present Model	Difference (%)
VCRC Cooling Rate	300 kW	300 kW	0	136.3 kW	300 kW	0
ORC Efficiency	0.81	0.82	1.2	0.11	0.11	0
VCRC COP	6.32	6.31	0.002	4.84	4.84	0
Turbine Power	50.34 kW	50.84 kW	0.01	28.17 kW	28.17 kW	0
Compressor Power	47.47 kW	47.52 kW	0.001	28.17 kW	28.17 kW	0
Boiler heat rate	458.8 kW	461.3 kW	0.005	245.15 kW	245.15 kW	0

3.2. Evaluation of Energy and Exergy Analysis Results

The simulations are performed by considering the operating conditions given in Table 1. In the analysis, the working fluids used in the ORC system are R114, R123, R600, R600a, and R245fa. In the subsystem VCRC, R141b, R600a, R290, R134a, R123, R245fa and R143a are applied as a refrigerant. Calculations for the proposed combined system to compare the performance parameters are realized for the thirty-five fluid pair combinations under the same operating conditions. The simulation results are presented in Table 8. The evaluation parameters in the thermodynamic analysis are as follows: the overall system coefficient of performance (COP_s), exergy efficiency (η_{ex}), energy utilization factor (EUF), net power ($\dot{W}_{net}$), total exergy destruction rate ($\dot{E}x_{dest,tot}$), evaporator heat rate ($\dot{Q}_{ev}$), the mass flow rate of the working fluid in the ORC ($\dot{m}_{ORC}$) and the mass flow rate of refrigerant in the VCRC ($\dot{m}_{VCRC}$).

Table 8. The thermodynamic results of the proposed system for fluid pairs.

Fluid in ORC	Fluid in VCRC	COP_s	EUF (%)	η_{ex} (%)	$\dot{W}_{net}$ (kW)	$\dot{E}x_{dest,tot}$ (kW)	$\dot{Q}_{ev}$ (kW)	$\dot{m}_{ORC}$ (kg/s)	$\dot{m}_{VCRC}$ (kg/s)
R245fa	R141b	0.7702	46.89	45.96	15.13	35.73	78.65	1.064	0.390
	R600a	0.7313	44.91	45.1	15.13	35.78	74.68	1.064	0.268
	R290	0.7047	43.54	44.53	15.13	35.76	71.96	1.064	0.248
	R134a	0.7165	44.15	44.77	15.13	35.76	73.16	1.064	0.475
	R143a	0.6659	41.57	43.67	15.13	35.76	68.00	1.064	0.495
	R123	0.7563	46.18	45.58	15.13	35.75	77.24	1.064	0.517
	R245fa	0.7470	45.70	45.44	15.13	35.77	76.28	1.064	0.468
R600	R141b	0.7679	47.75	45.39	14.59	36.24	80.91	0.557	0.401
	R600a	0.7291	45.70	44.51	14.59	36.30	76.82	0.557	0.278
	R290	0.7026	44.30	43.92	14.59	36.28	74.02	0.557	0.255
	R134a	0.7143	44.92	44.17	14.59	36.27	75.26	0.557	0.489
	R143a	0.6639	42.27	43.04	14.59	36.27	69.95	0.557	0.509
	R123	0.7541	47.02	45.00	14.59	36.26	79.45	0.557	0.532
	R245fa	0.7447	46.53	44.85	14.59	36.28	78.47	0.557	0.481
R600a	R141b	0.7303	46.97	42.33	13.09	37.92	80.85	0.666	0.401
	R600a	0.6933	44.93	41.45	13.08	37.98	76.77	0.666	0.275
	R290	0.6681	43.53	40.86	13.08	37.96	73.97	0.666	0.255
	R134a	0.6793	44.15	41.11	13.08	37.95	75.21	0.666	0.488
	R143a	0.6313	41.49	39.98	13.08	37.96	69.9	0.666	0.509
	R123	0.7171	46.24	41.94	13.08	37.95	79.4	0.666	0.532
	R245fa	0.7082	45.75	41.79	13.08	37.96	78.41	0.666	0.481
R123	R141b	0.8088	47.90	48.53	16.56	34.63	79.24	1.125	0.393
	R600a	0.7679	45.90	47.67	16.56	34.69	75.24	1.125	0.270
	R290	0.7400	44.53	47.10	16.56	34.67	72.50	1.125	0.250
	R134a	0.7524	45.14	47.34	16.56	34.66	73.71	1.125	0.478
	R143a	0.6993	42.54	46.24	16.56	34.67	68.51	1.125	0.499
	R123	0.7326	43.45	47.7	16.56	32.43	71.58	1.125	0.479
	R245fa	0.7844	46.71	48.01	16.56	34.67	79.24	1.125	0.393
R114	R141b	0.7254	46.91	42.10	12.87	37.94	80.96	1.576	0.402
	R600a	0.6888	44.87	41.21	12.87	38.00	76.87	1.576	0.276
	R290	0.6637	43.47	40.62	12.87	37.98	74.07	1.576	0.255
	R134a	0.6748	44.09	40.87	12.87	37.97	75.31	1.576	0.489
	R143a	0.6272	41.43	39.74	12.87	37.97	70.00	1.576	0.510
	R123	0.7124	46.19	41.71	12.87	37.96	79.50	1.576	0.533
	R245fa	0.7035	45.69	41.56	12.87	37.98	78.52	1.576	0.482

As seen in Table 8, the highest COP_s value is obtained as 0.8088 for the R123 ORC–R141b VCRC fluid pair. Additionally, for each organic fluid used in the ORC system, the maximum COP_s is reached using the R141b fluid in the VCRC. When the EUF , which shows the energy performance of the combined systems, is examined, the maximum value is obtained as 47.90% for the R123–R141b fluid couple. Like COP_s , the highest EUF is achieved for each ORC working fluid by using R141b refrigerant in the cooling cycle. Among the fluid combinations, the lowest COP_s and EUF values are calculated for the R114–R143a fluid pair. Table 8 also shows that the performance parameters for both subsystems employing the same working fluids such as R245fa, R600a, etc. Among the cases in which the same fluid is used in the sub systems, the case of R245fa shows the highest COP_s and EUF values as 0.747 and 45.70%, respectively.

The maximum exergy efficiency of the combined system is calculated as 48.53% for the R123–R141b combination. In addition, as can be seen from Table 8, the highest exergy efficiency values are observed by using R123 fluid in the ORC sub system.

When examining the net power output, the maximum net power 16.56 kW is obtained using R123 in the ORC system. The organic fluids with greater net power to lower ones are

R245fa, R600, R600a, and R114. The lowest total exergy destruction values are also obtained by selecting R123 fluid in the ORC system.

The evaporator heat rate required for the air to reach the desired conditions on the cooling side of the combined system has been calculated for all fluid combinations. It should be mentioned that the compressor power in the VCRC is supplied by the low pressure expander of the ORC. Hence, the cooling rate, $\dot{Q}_{ev}$, depends on the performance of the low pressure expander of the ORC. The results obtained are listed in the eighth column of Table 8. Accordingly, the minimum $\dot{Q}_{ev}$ value is obtained as 68 kW for the R245fa-R143a combination. In addition, as can be seen from the table, the lowest $\dot{Q}_{ev}$ values are obtained by using R143 in the refrigeration cycle for all fluids used in the ORC.

The required mass flow rates of the combined cycle, which will meet the accepted conditions in the energy and exergy analysis performed, are calculated and given in the last two columns of Table 8. Accordingly, the minimum mass flow rate of the organic fluid that will circulate in the ORC system for the air to achieve the desired cooling has been calculated as approximately 0.56 kg/s for the R600 fluid. This is followed by the R600a, R245fa, R123, and R114, respectively. The lowest mass flow rate of the refrigerant in the VCRC is calculated as 0.25 kg/s for the R123-R290 combination. The low mass flow rate of the circulating working fluid is important in terms of the initial investment and operating costs [61].

When all parameters obtained in Table 8 are examined, the best performance results are obtained by using R123 fluid on the ORC side and R141b fluid on the VCRC side of the combined system. It is also observed that there is a notable advantage to employ different working fluids in each subsystem of the combined system. At the same time, it can be said that these two fluids are environmentally friendly fluids due to their zero ODP and very low GWP values.

The thermodynamic properties of air, R123 and R141b for each state point of the designed system shown in Figure 1 were calculated utilizing the governing equations, which are given in Table 3. The results obtained are presented in Table 9.

Table 9. System data, thermodynamic properties, and exergies with respect to state points.

State	Fluid	$\dot{m}$ (kg/s)	T (°C)	P (kPa)	h (kJ/kg)	s (kJ/kgK)	$\dot{E}_x$ (kW)
0	Air		30.0	101.0	401.2	1.673	0.00
0'	R123		30.0	101.0	303.4	6.876	0.00
0''	R141b		30.0	101.0	303.4	0.273	0.00
1	R123	1.03	35.0	154.7	236.6	1.126	1.42
2	R123	1.03	35.1	348.8	236.8	1.126	1.56
3	R123	1.03	50.9	348.8	253.5	1.178	2.33
4	R123	1.14	67.1	348.8	271.1	1.231	4.41
5	R123	1.14	67.1	786.8	271.5	1.231	4.79
6	R123	1.14	105.0	786.8	447.0	1.706	40.95
7	R123	1.14	82.0	348.8	435.6	1.714	25.15
8	R123	1.03	82.0	348.8	435.6	1.714	22.71
9	R123	0.11	82.0	348.8	435.6	1.714	2.43
10	R123	1.14	62.6	154.7	424.1	1.722	8.25
11	R123	1.14	40.3	154.7	407.4	1.671	7.02
12	Air	20.00	112.8	101.0	387.0	7.120	193.80
13	Air	20.00	102.9	101.0	377.0	7.094	153.00
14	Air	34.89	30.0	101.0	303.4	6.876	0.00
15	Air	34.89	35.0	101.0	308.4	6.893	1.44
16	R141b	0.29	5.0	28.1	280.8	1.039	−7.22
17	R141b	0.29	60.4	132.9	321.3	1.064	2.41
18	R141b	0.29	35.0	132.9	79.2	0.292	0.02
19	R141b	0.29	0.0	28.1	79.2	0.301	−0.77
20	Air	11.70	10.0	101.0	283.3	6.808	8.12
21	Air	11.70	5.0	101.0	278.3	6.790	12.84
22	Air	13.07	30.0	101.0	303.4	6.876	0.00
23	Air	13.07	35.4	101.0	308.8	6.894	0.62

3.3. Evaluation of Exergoeconomics and Exergy Sustainability Analysis Results

In the second stage of the analysis, exergetic sustainability indicators named as: waste exergy ratio (*WER*), exergy destruction factor (*EDF*), environmental effect factor (*EEF*), and exergetic sustainability index (*ESI*); and exergetic economic parameters (the total system cost ($\dot{C}_{sys}$), the unit cost of electricity produced (C_{el}), and the payback period (*PB*)) were calculated for the considered cases. The analysis results are given in Table 10.

Table 10. Exergetic sustainability indicators of the proposed system for various fluid pairs.

Fluid in ORC	Fluid in VCRC	<i>WER</i>	<i>EDF</i>	<i>EEF</i>	<i>ESI</i>	$\dot{C}_{sys}$ \$/h	C_{el} \$/kWh	<i>PB</i> Year
R245fa	R141b	0.691	0.542	1.504	0.665	2.014	0.0705	2.67
	R600a	0.692	0.550	1.535	0.651	2.001	0.0698	2.63
	R290	0.692	0.553	1.554	0.644	1.998	0.0697	2.62
	R134a	0.692	0.551	1.545	0.647	2.008	0.0702	2.65
	R143a	0.692	0.560	1.585	0.631	2.004	0.0700	2.64
	R123	0.692	0.545	1.518	0.659	2.020	0.0709	2.68
	R245fa	0.692	0.547	1.523	0.657	2.017	0.0707	2.68
R600	R141b	0.701	0.549	1.545	0.647	2.094	0.0722	2.71
	R600a	0.702	0.556	1.578	0.634	2.082	0.0715	2.68
	R290	0.702	0.560	1.599	0.626	2.079	0.0713	2.67
	R134a	0.702	0.558	1.589	0.629	2.088	0.0718	2.70
	R143a	0.702	0.567	1.631	0.613	2.084	0.0716	2.69
	R123	0.702	0.552	1.559	0.641	2.101	0.0726	2.73
	R245fa	0.702	0.553	1.565	0.639	2.098	0.0724	2.72
R600a	R141b	0.731	0.579	1.727	0.579	2.338	0.0749	2.86
	R600a	0.732	0.587	1.767	0.566	2.326	0.0741	2.83
	R290	0.732	0.591	1.792	0.558	2.323	0.0740	2.82
	R134a	0.732	0.589	1.780	0.562	2.333	0.0745	2.84
	R143a	0.732	0.597	1.831	0.546	2.329	0.0743	2.83
	R123	0.732	0.582	1.745	0.573	2.345	0.0752	2.88
	R245fa	0.732	0.584	1.752	0.571	2.342	0.0751	2.87
R123	R141b	0.666	0.517	1.372	0.729	1.887	0.0672	2.54
	R600a	0.667	0.524	1.399	0.715	1.874	0.0665	2.51
	R290	0.666	0.528	1.415	0.707	1.872	0.0664	2.50
	R134a	0.666	0.526	1.407	0.711	1.881	0.0669	2.53
	R143a	0.666	0.534	1.441	0.694	1.877	0.0667	2.52
	R123	0.669	0.522	1.402	0.713	1.767	0.0691	2.63
	R245fa	0.667	0.522	1.388	0.720	1.890	0.0674	2.55
R114	R141b	0.734	0.581	1.745	0.573	2.599	0.0758	2.86
	R600a	0.736	0.589	1.785	0.560	2.586	0.0751	2.83
	R290	0.735	0.593	1.810	0.553	2.583	0.0749	2.82
	R134a	0.735	0.591	1.798	0.556	2.592	0.0755	2.85
	R143a	0.735	0.599	1.850	0.541	2.589	0.0752	2.84
	R123	0.735	0.584	1.762	0.568	2.605	0.0762	2.88
	R245fa	0.735	0.586	1.769	0.565	2.602	0.0760	2.87

It is seen that the minimum waste exergy ratio (*WER*) values are obtained by using R123 fluid in the ORC system. In other words, it is the R123 fluid that makes more use of the exergy of the hot fluid. Similarly, the minimum exergy destruction factor (*EDF*) values are obtained by using R123 fluid in the power cycle. The lowest *EDF* value is calculated as 0.517 for the R123 ORC-R141b VCRC fluid couple. A higher sustainability index means a lower environmental impact factor. Accordingly, when Table 10 is examined, the lowest *EEF* and, therefore, the highest *ESI* values are obtained using R123 fluid in the power cycle. The minimum *EEF* and maximum *ESI* belong to the R123-R141b fluid pair.

Meanwhile, the exergoeconomic analysis results are given in the last three columns of Table 10. The best results of the $\dot{C}_{sys}$, C_{el} , and PB values are obtained for fluid combinations formed by using R123 in the ORC system. The lowest total system cost was calculated as 1767 \$/h for the R123 ORC-R123 VCRC fluid pair. The minimum value of the unit cost of electricity produced by the combined system, 0.0664 \$/kWh, was obtained for the R123 ORC-R290 VCRC. The fluid combination that allows the combined system to pay for itself in the shortest time ($PB = 2.5$ year) was R123 ORC-R290 VCRC.

The exergy destruction rates for each component of the dual-pressure ORC-VCRC system for the R123-R141b fluid couple are presented in Figure 3. The total exergy destruction under the accepted conditions is 21.70 kW, of which 21.68% is the maximum irreversibility rate donated by the boiler. It is followed by condenser 1, the high-pressure expander, the low-pressure expander and the compressor with 19.15%, 12.79%, 12.27%, and 10.04% ratios, respectively. The rates of exergy destruction in condenser 2 and the evaporator are 8.17% and 7.98%. The exergy destruction values realized in pump 1, pump 2, and the recuperator remained relatively low compared to other components. These results qualitatively agree with the reported data by Javanshir et al. [58]. It should be noted that their cycle configurations and operating conditions were different than the present study. These results indicate that there are higher exergy destructions in components such as the boiler, the expanders, the condenser, and the evaporator, and that these units could be modified to improve the performance and the exergy efficiency of the combined system.

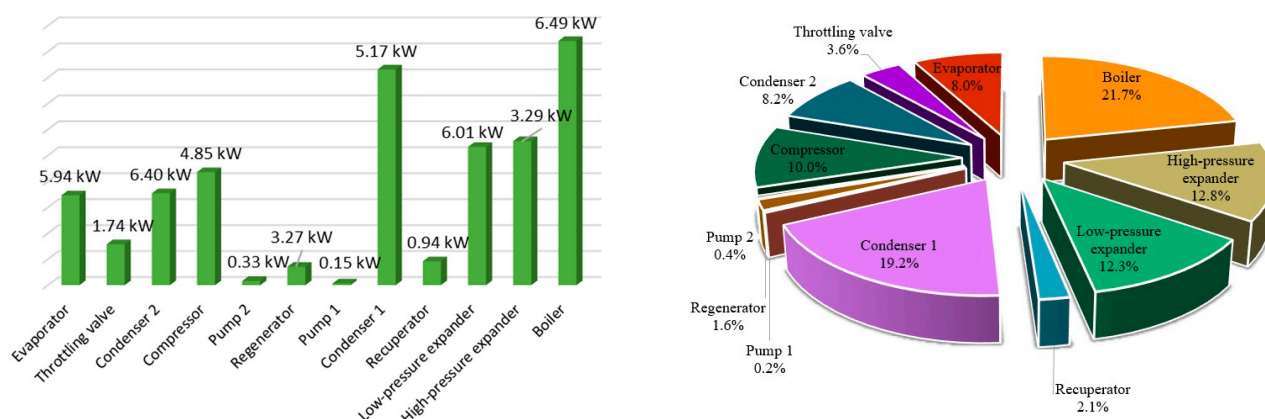


Figure 3. Exergy destruction rates of system components of R123 ORC-R141b VCRC.

3.4. Parametric Analysis

In this section, a parametric study is carried out to determine the effect the boiling temperature (T_b) of the working fluid on the performance indicators of the proposed combined system. The results are presented in Figures 4–16. In the analysis, the boiling temperature is increased from 90 °C to remain below the critical temperature of each fluid evaluated. The first output parameter is the COP_s of the system. From the observation of Figure 4, it can be seen that the COP_s rises dramatically for all of the working fluids with the increasing boiler temperature. These results are in parallel with the reported observations by and Cihan et al. [22], Özdemir Küçük and Kılıç [34], and Saleh [60]. Increasing the evaporation temperature in the boiler results an increase in the inlet pressure of the expander. Hence, the ORC system works under a higher pressure difference, and more power could be produced by the expanders. Additionally, this results in a notable increase on the COP_s . The highest values of the COP_s are obtained for the R123-R141b fluid pair, which has the highest boiling temperature and critical temperature. It is followed by R245fa-R141b, R600-R141b, R600a-R141b, and R114-R141b. Additionally, R245fa and R600 have nearly the same COP_s values in some regions. When the boiler temperature approaches the critical temperature of the fluids, the power production decreases and the

COP_s also decreases. This can occur for the cases that employed R114 and R600a fluids in the ORC, as seen in Figure 4.

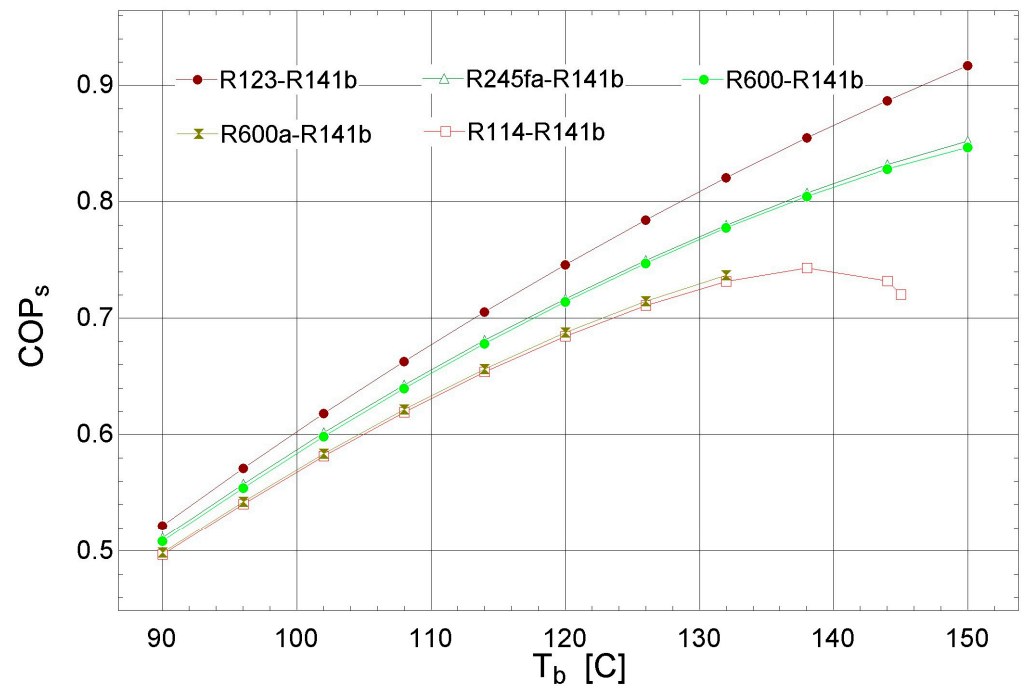


Figure 4. The effect of the T_b on the overall system coefficient of performance (COP_s).

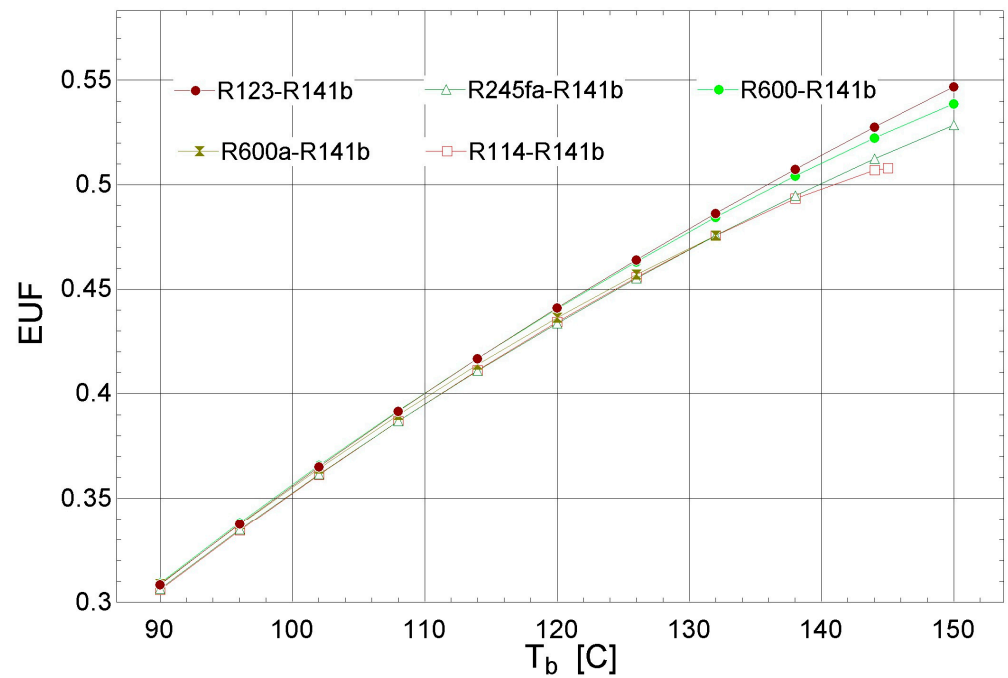


Figure 5. The effect of the T_b on the energy utilization factor (EUF).

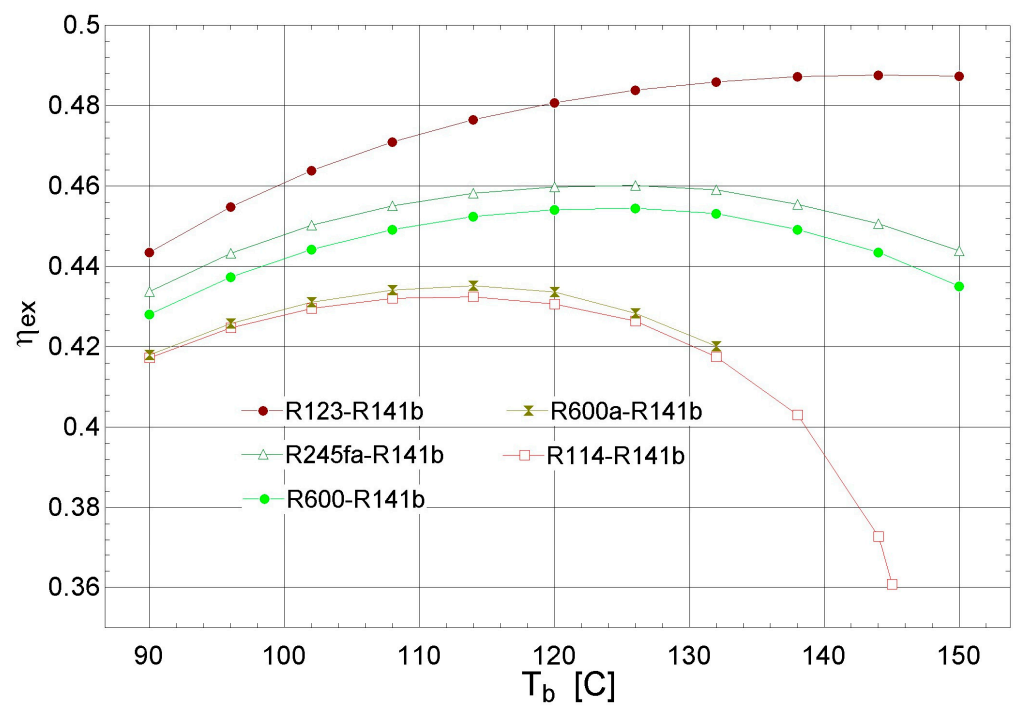


Figure 6. The effect of the T_b on the exergy efficiency (η_{ex}).

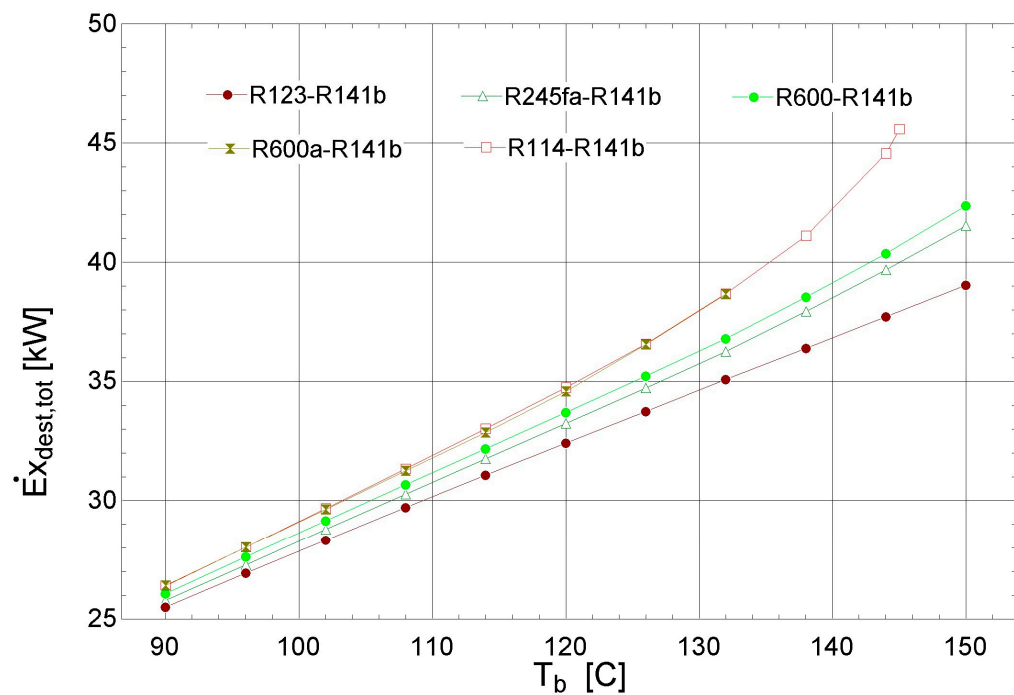


Figure 7. The effect of the T_b on the total exergy destruction ($\dot{E}X_{dest,tot}$).

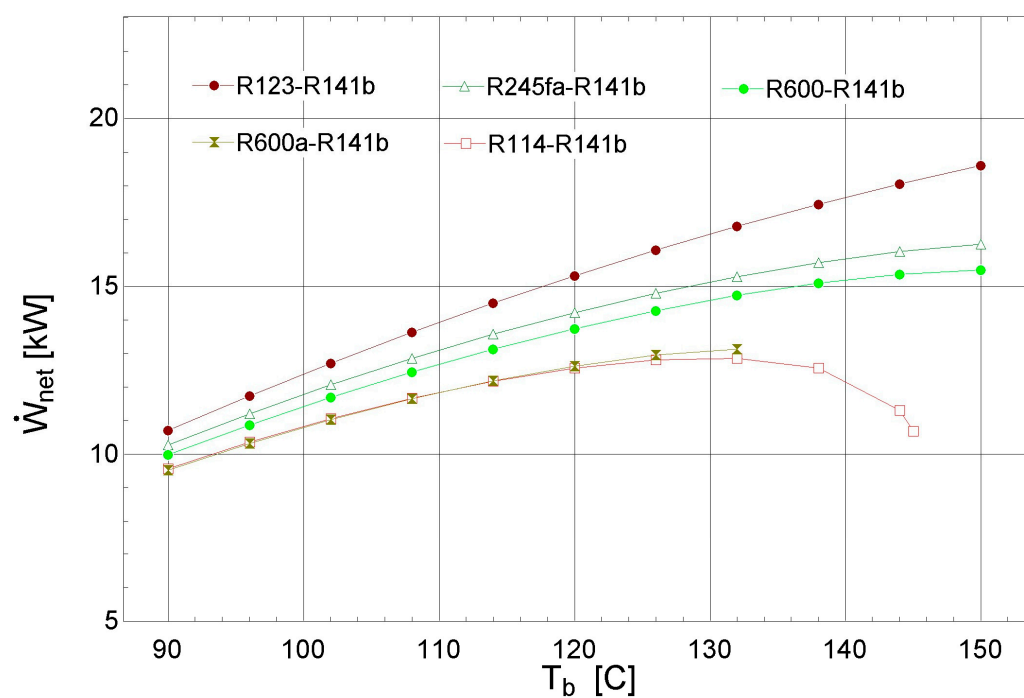


Figure 8. The effect of the T_b on the net power output ($\dot{W}_{net}$).

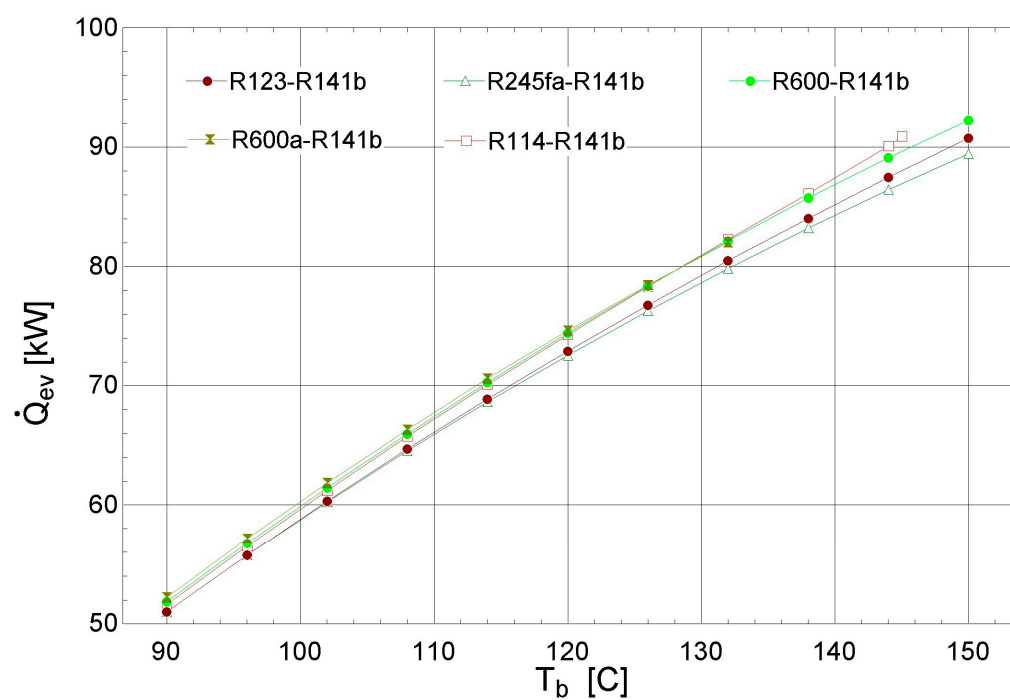


Figure 9. The effect of the T_b on the evaporator heat rate ($\dot{Q}_{ev}$).

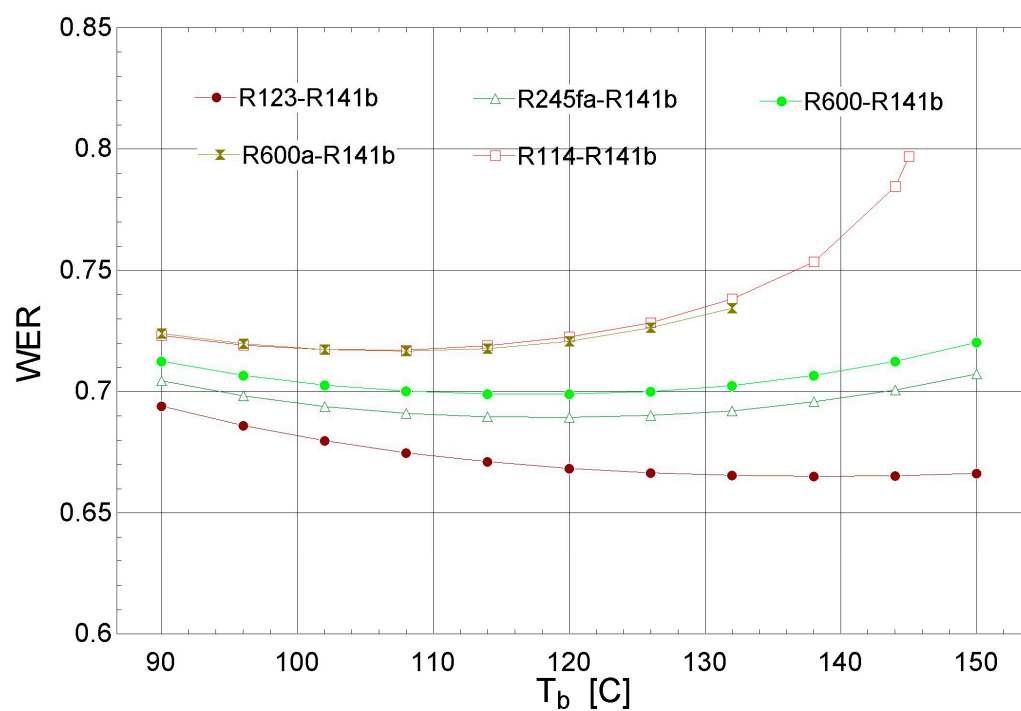


Figure 10. The effect of the T_b on the waste exergy ratio (WER).

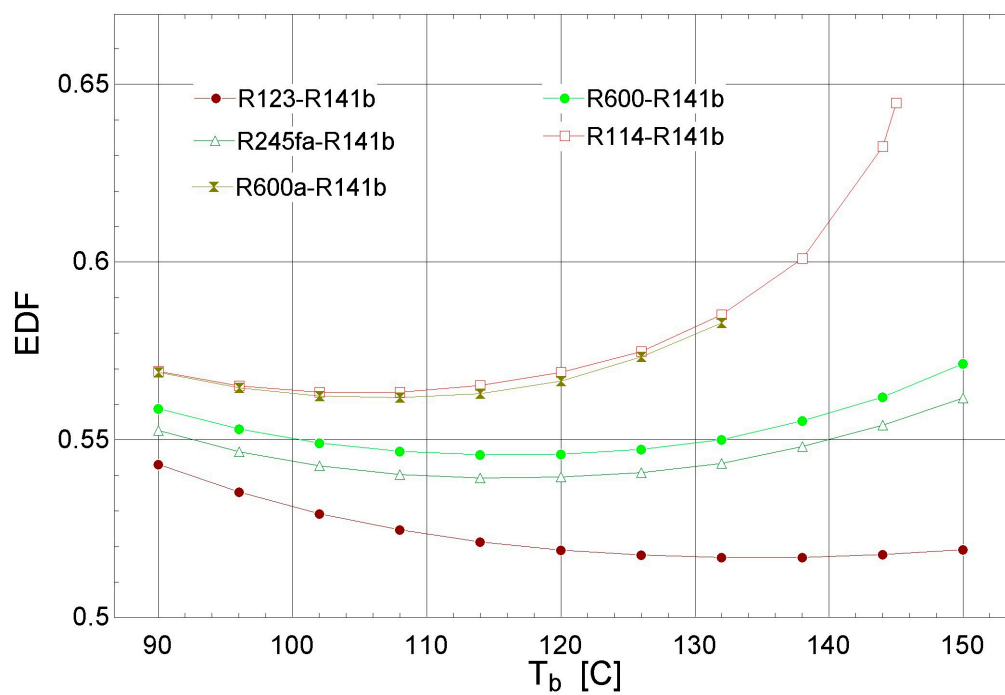


Figure 11. The effect of the T_b on the exergy destruction factor (EDF).

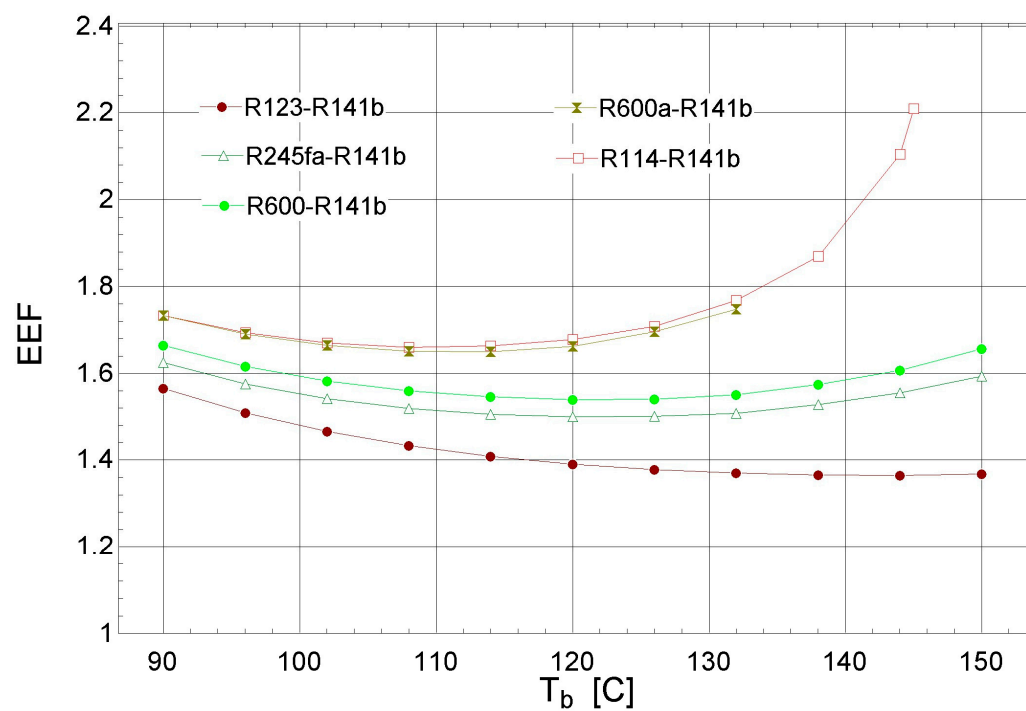


Figure 12. The effect of the T_b on the environmental effect factor (EEF).

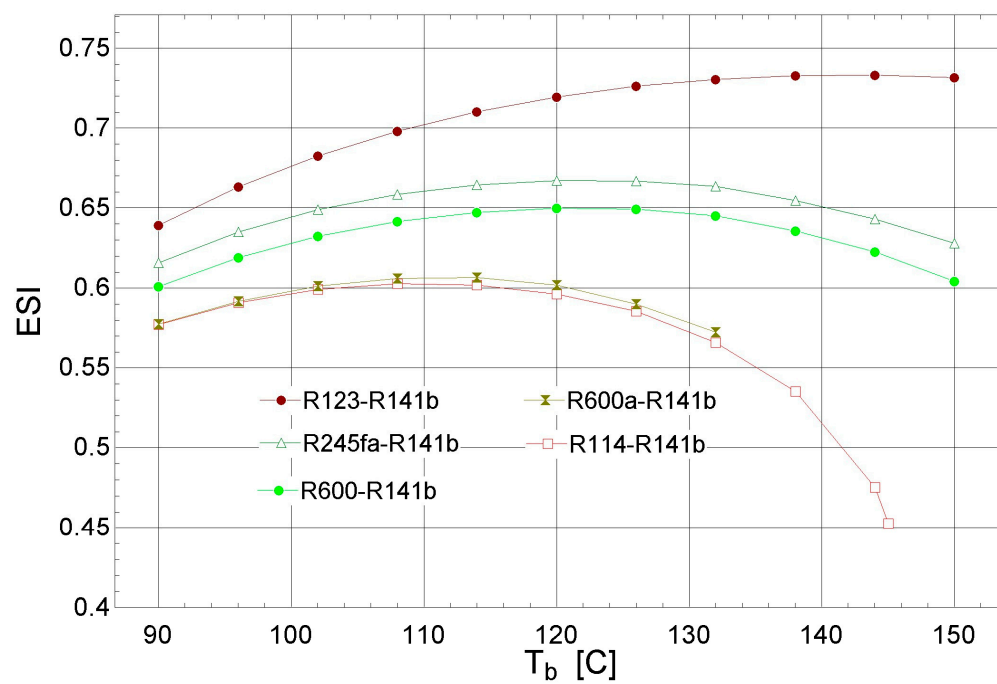


Figure 13. The effect of the T_b on the exergetic sustainability index (ESI).

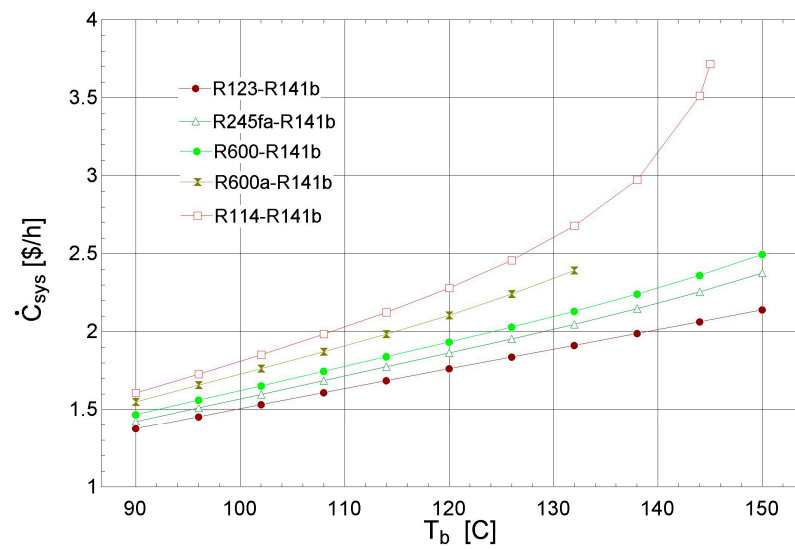


Figure 14. The effect of the T_b on the total system cost ($\dot{C}_{sys}$).

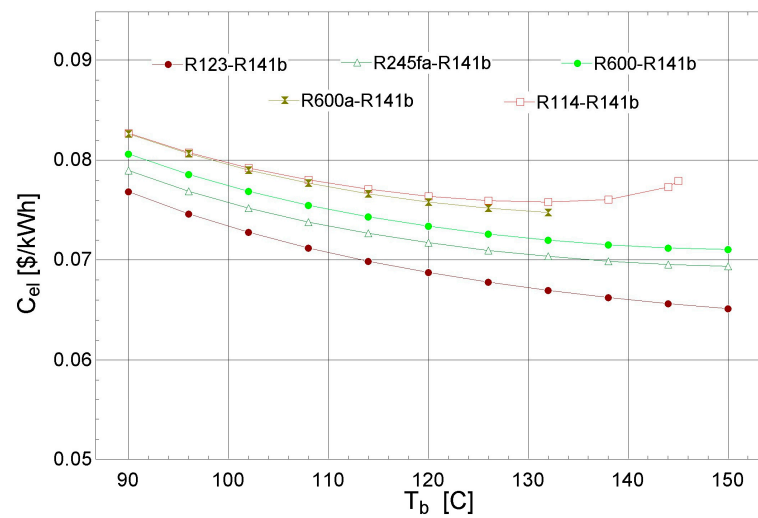


Figure 15. The effect of the T_b on the unit cost of electricity produced (C_{el}).

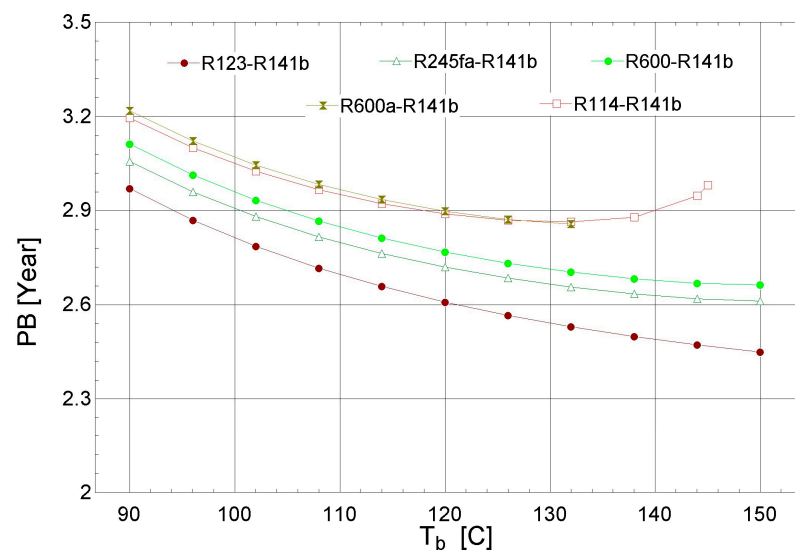


Figure 16. The effect of the T_b on the payback period (PB).

The variation of EUf , which expresses the efficiency of the combined system with the increasing boiling temperature, is given in Figure 5. Similar to the result obtained by Cihan et al. [22], the EUf value for all considered fluid pairs increases as the boiler temperature increases. An increase in the boiler temperature results in an increased pressure level in the boiler and at the inlet of the expander. This means more power could be obtained from the expanders to produce electricity and to supply compressor power to obtain cooling power. Therefore, the EUf values for all the considered fluid pairs grow as the boiler temperature rises. The highest EUf values belong to the R123-R141b fluid couple. Especially at low boiling temperatures, the R123-R141b and R245fa-R141b fluid combinations give almost the same EUf value. As the boiling temperature increased, the maximum EUf values belonged to the R123-R141b fluid couple.

The change in the exergy efficiency of the combined system with increasing boiling temperature was investigated, and the obtained results are presented in Figure 6. Increasing the boiler temperature, the quality of the heat increases, i.e., the supplied exergy to the system increases. Hence, the η_{ex} increases with the rising boiler temperature. However, this increase stops at a specific temperature depending on the fluids' characteristics, and then the η_{ex} decreases for all fluid pairs. Maximum η_{ex} values are obtained for R123-R141b. As can be seen from the figure, the R114-R141b fluid couple has minimum exergy efficiency values.

In Figure 7, the total exergy destruction rates of fluid combinations are presented for different boiling temperatures. Increasing the boiler temperature, the supplied exergy to the system increases. Meanwhile, any loss or irreversibility in the energy conversion process results more exergy destruction in the system. Hence, the $\dot{E}x_{dest,tot}$ increases with the rising boiler temperature. The results show that the $\dot{E}x_{dest,tot}$ rises with the increasing boiling temperature. The minimum and maximum total exergy destruction rates are calculated for the R123-R141b and R114-R141b fluid combinations, respectively.

Figure 8 illustrates the impact of the boiling temperature on the net power generation. An increase in the boiler temperature results in an increased pressure level at the inlet of the expander. This means more power could be obtained from the expanders to produce electricity. The R123-R141b fluid couple achieves the maximum power generation. This fluid pair is followed by R245fa-R141b, R600-R141b, R600a-R141b, and R114-R141b, respectively.

Figure 9 shows the effects of the boiling temperature on the evaporator heat rate. Since, an increase in the boiler temperature results in a rise of power obtained from the expanders for electricity production and compressor driving, the cooling power ($\dot{Q}_{ev}$) increases with the boiling temperature. As can be clearly seen from the figure, the $\dot{Q}_{ev}$ for the five fluid pairs rises with increasing boiling temperature. For the R123-R141b and R245fa-R141b fluid pairs, the evaporator heat rate is calculated at almost the same values at low boiling temperatures. Minimum $\dot{Q}_{ev}$ values at relatively higher T_b temperatures are obtained with the combination of R245fa-R141b fluid.

Thus far, how the increasing boiling temperature affects the thermodynamic performance of the system has been investigated. After that, the change of the sustainability indicators and exergoeconomic parameters of the designed system with increasing T_b was evaluated. Figure 10 shows the effect of increasing T_b on the waste exergy ratio (WER). It is observed that the WER value for all fluid pairs first decreases and then increases with T_b . The minimum WER is obtained for the R123-R141b fluid pair at each temperature level. This is followed by R245fa-R141b, R600-R141b, R600a-R141b, and R114-R141b, respectively.

The variation of the EDF for different boiling temperatures is presented in Figure 11. Accordingly, with the increasing boiling temperature, the EDF factor first shows a decreasing trend and then an increasing trend for all fluid pairs. The order of EDF values from minimum to maximum is as follows, by fluid pair: R123-R141b, R245fa-R141b, R600-R141b, R600a-R141b, and R114-R141b. This ranking is in parallel with the maximum total exergy destruction rate in the system.

The effect of increasing boiling temperature on the environmental effect factor (*EEF*) was calculated for the fluid pairs and the results were plotted in Figure 12. Accordingly, as the boiling temperature increases in the analysis, similarly, the *EEF* first decreases and then increases. The lowest *EEF* values at each temperature level are obtained with the combination of R123-R141b. It can be seen from Figures 11 and 12 that the *EDF* and *EEF* show parallel behaviors. It means that when the exergy destruction increases the effect of the environment increases, more of the available exergy is destroyed by the combined system. It can be concluded that each system must be carefully analyzed to define the optimum operating conditions by using the specific characterization of the configuration and the working fluids.

In Figure 13, the variation of the exergetic sustainability index (*ESI*) depending on different boiling temperatures is given. As can be clearly seen from the figure, the maximum *ESI* value is reached with the R123-R141b fluid coupled. Additionally, the *ESI* for the R245fa-R141b, R600-R141b, R600a-R141b and R114-R141b fluid pairs shows an increasing and then decreasing graph. The minimum *ESI* values are calculated for the R114-R141b combination. As it is defined as the reverse of *EEF*, the same conclusions can be made on the reverse side. It means that when the exergy destruction decreases the effect of the environment decreases, the available exergy is properly utilized by the combined system.

The total system cost ($\dot{C}_{sys}$) was plotted in Figure 14 against varying boiling temperatures. As can be seen from the results, the $\dot{C}_{sys}$ increases with the increasing boiling temperature for all fluid combinations. The reason for the increase in the $\dot{C}_{sys}$ value is the increase in both the initial investment cost and the exergy destruction cost with the increasing boiling temperature. The maximum rate of increase in $\dot{C}_{sys}$ values occurs for the R114-R141b fluid pair. The minimum $\dot{C}_{sys}$ at each level of the boiling temperature is calculated for the R123-R141b combination.

Figure 15 shows the effects of the boiling temperature on the unit cost of electricity produced (C_{el}). According to the results, as the boiling temperature increases in the analysis, the C_{el} value decreases for all fluids. This is because, with the increasing boiling temperature, the purchased equipment cost and net power values of the system increase. However, the C_{el} value decreases according to Equation (29) because the increase rate in net power is higher. The minimum C_{el} value is calculated as 0.065 \$/kWh for R123-R141b fluid at 150 °C boiling temperature.

Figure 16 illustrates the change in the payback period (*PB*) of the system with the increasing boiling temperature. As it can be clearly seen from the figure, the *PB* decreases for almost all fluids as the boiling temperature increases. The fluid couple that pays for itself in the shortest time is R123-R141b. Considering the decrease rates due to the increasing boiling temperature, the maximum decrease is 17.5% for the R123-R141b fluid. The minimum payback period is obtained as 2.45 years with the R123-R141b combination at 150 °C boiling temperature.

4. Conclusions

A cogeneration system that produces both power and cooling using waste heat could be more sustainable compared to separate systems. This study aims to design a novel combined ORC-VCRC system that generates power and cooling energy using the waste heat of low-temperature flue gas and analyze it in terms of exergetic sustainability and exergoeconomic. In the designed combined system, the ORC sub system is dual-pressure and has two separate expanders, a high-pressure expander, and a low-pressure expander. The power generated by the low-pressure expander is used in the compressor of the cooling cycle and cooling is carried out. The working fluids circulating in the ORC and VCRC systems are separate, and thirty-five different fluid pairs were considered in the analysis. According to the results, the highest COP_s and *EUF* values were obtained for the R123-R141b fluid pair as 0.8088 and 47.9%, respectively. The maximum exergy efficiency, 48.53% was achieved with the R123-R141b fluid. The maximum net power generation was

16.56 kW using R123 in the ORC system. The minimum evaporator heat rate of 68.5 kW was performed with R123-R143a.

The result of exergetic sustainability analysis showed that the minimum *WER* value was calculated as 0.666 for R123-R141b, R123-R290, R123-R134a, and R13-R143a among the fluid combinations created. The lowest *EDF* belonged to the R123-R141b. The lowest *EEF* and thus the maximum *ESI* value belonged to the R123-R141b combination as 1.399 and 0.715, respectively.

As a result of the exergoeconomic analysis, the minimum total system cost was calculated as 1.767 \$/h for the R123-R123 fluid pair. The lowest the unit cost of electricity produced and payback period were obtained for R123-R290 as 0.0664 and 2.5 years, respectively.

The exergy analysis shows that the component with the highest exergy destruction was the boiler with 21.68%. This was followed by the condenser of the power cycle, the high-pressure expander, low-pressure expander, and compressor with 19.15%, 12.79%, 12.27%, and 10.04%, respectively.

A comparative parametric analysis was also carried out for five selected fluid pairs to examine the effect of boiling temperature in the boiler on the thermodynamic, exergoeconomic and sustainability parameters of the system. Some of the obtained results are listed below.

- In the analysis, as the boiling temperature rises, generally COP_s , EUF , η_{ex} , $\dot{E}x_{dest,tot}$, $\dot{W}_{net}$, and $\dot{Q}_{ev}$ increase.
- With the increasing boiling temperature, *WER*, *EDF* and *EEF* decrease, while *ESI* increases. Among the five fluid pairs evaluated, the fluid pair with the lowest, *WER*, *EDF* and *EEF* and the highest *ESI* was R123-R141b.
- When the boiling temperature increases, the $\dot{C}_{sys}$ increases, but C_{el} and *PB* decrease. The minimum values for all three parameters are obtained with the combination of R123-R141b.
- Any improvement in efficiency enhances the sustainability of the energy system. However, any rise in the waste energy ratio and the energy destruction factor leads to a rise in the environmental effect factor, which lowers sustainability. These parameters should quantify how the combined ORC-VCRC system becomes more sustainable and environmentally friendly.

It can be concluded that the proposed combined system is very effective to recover the waste energy. Additionally, it is feasible with the economic side. When designing such a system, exergetic sustainability and exergoeconomic parameters should be considered to obtain a more-efficient and less-environmental effects system. Further studies can be conducted on the optimization of the system components operating under flexible conditions.

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Nomenclature

COP	cooling cycle coefficient of performance
COP_s	overall system coefficient of performance
$\dot{E}$	energy rate (kW)
EDF	exergy destruction factor
EEF	environmental effect factor
ESI	exergetic sustainability index
EUF	energy utilization factor
$\dot{E}_x$	exergy rate (kW)
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
s	specific entropy (kJ/kgK)
$\dot{Q}$	heat rate (kW)
$\dot{W}$	power (kW)
WER	waste exergy ratio
T	temperature (K)
η	efficiency

Subscripts

b	boiler
$c1$	condenser 1
$c2$	condenser 2
$comp$	compressor
$dest$	destroyed
ev	evaporator
ex	exergetic
$heat$	heat
hpe	high-pressure expander
in	inlet
lpe	low-pressure expander
ORC	organic Rankine cycle
out	outlet
$VCRC$	vapor compression refrigeration cycle
$p1$	pump 1
$p2$	pump 2
rec	recuperator
reg	regenerator
tot	total
tv	throttling valve
o	dead state condition

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