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Simulation and thermodynamic analysis of a regenerative and recuperative organic Rankine cycle

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Due to the energy shortage, environmental pollutions and climate change, the issues of energy conversion technologies have become more and more significant. In recent years, organic Rankine cycle has become a leading technology for the conversion of heat into useful work or electricity. This promising technology uses an organic fluid which has high molecular mass hydrocarbon compound, low critical temperature, and pressure as a working fluid. In this paper, energy and exergy analysis of a waste air's heat-driven organic Rankine cycle, which has two turbines, two pumps, an evaporator, a condenser, a recuperator, and a feed fluid heater, is performed using R114, R600, R600a and R245fa organic fluids. The organic Rankine cycle's performance parameters are evaluated depending on various evaporation temperatures and the inlet pressure of the high-pressure turbine. The results indicate that R245fa has the highest thermal efficiency also the highest net power is obtained for the R600 working fluid. Also, the thermal efficiency and the net power increase with the increment of the evaporation temperature and these features raise before and then decreases with the increasing high-pressure turbine inlet pressure in the analysis of ORC.

Keywords: Organic Rankine cycle, working fluid, thermodynamic analysis, high-pressure turbine

INTRODUCTION

In recent years, the energy consumption of countries has increased due to the increment of social and economic factors such as population, industrialization, urbanization, technological development, etc. in a globalizing world. Therefore, providing sustainable energy policies which are the primary input of countries' economic development; ensuring the security of energy supply, and diversifying of energy resources have happened vital issues. As a result, the popularity of the organic Rankine cycle (ORC) has increased recently to convert the low-grade heat sources into power.

The organic Rankine cycle (ORC), which uses an organic fluid instead of water as a working fluid, is a power generation cycle from low-grade waste heat [1, 2] and renewable energy sources, such as solar energy [3, 4], biomass energy [5, 6], geothermal energy [7, 8]. This promising technology consists of four phases: pressure increase in the feed pump; isobaric heating, evaporation and overheating of the working fluid in the evaporator; expansion of the vapor working medium in an expansion machine (e.g. a turbine); isobaric heat release, complete condensation and possible under-cooling of the working medium in the condenser.

The slope of saturation vapor curve of a working fluid in T–s diagram is the most crucial feature to determine the fluid applicability, system efficiency, work output and also the overall structure of the system in an ORC. Working fluids for ORC's are categorized in three groups based on their slope of saturation vapor curves in T–s diagram. The fluids having positive slope are dry fluids ($ds/dT > 0$). The fluids having negative slope are wet fluids ($ds/dT < 0$). The fluids having nearly infinitely large slopes are isentropic fluids ($ds/dT=0$) [9]. In the ORC, dry or isentropic fluids are more convenient because they do not require superheating in the evaporator to avoid forming moisture in the working fluid during the expansion process [10, 11].

One of the effects to increase the system efficiency of the ORC is the application of different configurations of ORC which are the double stage ORC, regenerative ORC, recuperative ORC and both regenerative and recuperative ORC. Many studies on energy production from low-grade waste heat and renewable energy resources using ORC configurations have been presented in the literature. For example; Shokati et al. [12] compared the basic, dual-pressure and dual-fluid ORCs and Kalina cycle for power generation from the geothermal fluid reservoir utilizing energy, exergy and exergo-economic viewpoints. Their results show that among the considered cycles, dual-pressure ORC has the maximum value of produced electrical power. This is 15.2%, 35.1% and 43.5% more than the corresponding values for the basic ORC, dual-fluid ORC and Kalina cycle, respectively in optimal condition.

Ayachi et al. [13] analysed the exergetic optimization of single and double stage ORCs for

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waste heat recovery from the one almost dry heat source and highly moist heat source. They examined different combinations of working fluids, such as R1234yf, R245fa (topping cycle) and R245fa, R125, R41 and CO_2 (bottoming cycle). Their results indicate that global exergy efficiency is strongly linked to the critical temperature of the working fluid. They calculated the highest performances in the supercritical operating conditions and estimated that the addition of a low temperature bottoming cycle for recovering the heat during the condensing process offers an efficiency increase potential of about 33%.

Braimakis and Karellas [14] examined three regenerative ORCs which are including an open pre-heater and two additional configurations with closed-type pre-heater regenerative ORC with backward bleed condensate circulation. In the second configuration, the bleed stream is throttled and conveyed to the condenser. In the third ORC configuration, the bleed stream is re-pressurized via a secondary pump and re-circulates into the evaporator. Their results show that recuperative and regenerative ORCs are mostly suitable for dry fluids. Also, simple recuperative ORC has a higher efficiency than the non-recuperative regenerative cycles. The ORC with closed pre-heater and a secondary pump has the highest efficiency, and it is followed by the ORC with open pre-heater and lastly the ORC with a recuperative and open preheater.

Safarian and Aramoun [15] studied a theoretical framework for energy and exergy evaluation of a basic ORC and modified ORC which consider incorporating turbine bleeding, regeneration and both of them. They concluded that the integrated ORC with recuperation and regeneration has the best thermal and exergy efficiencies, equal to 22.8% and 35.5% respectively.

Xi et al. [16] performed the thermodynamic optimization of different ORC system configurations using six different working fluids for low-grade waste heat. The examined configurations are basic ORC, the single-stage regenerative ORC system and double-stage regenerative ORC system. Their results demonstrate that the double-stage regenerative ORC system gives the best thermal efficiency and exergy efficiency under the optimal operating conditions.

Mago et al. [17] evaluated an analysis of regenerative organic Rankine cycle using dry organic fluids, to convert waste energy to power from low- grade heat sources. They selected four dry organic working fluids which are R113, R245ca, R123 and isobutene. Researchers analyzed basic ORC and regenerative ORC using a combined first and second law analysis at various reference temperatures and pressures.

Shokati et al. [18] performed a comparative exergo-economic analysis for heat recovery from gas turbine-modular helium reactor (GT-MHR) using simple ORC, ORC with internal heat exchanger and regenerative ORC (RORC) and compared these combined cycles exergo-economically. The results showed that regenerative ORC has the minimum unit cost of power produced by the turbine and this parameter was the maximum for that ORC with an internal heat exchanger. It was also shown that ORC with internal heat exchanger has the maximum exergy destruction cost rate.

Liu et al. [19] analysed the performance of different ORC plant configurations which are a simple cycle, superheated cycle, recuperated cycle and regenerative cycle respectively, with different working fluids for low temperature binary-cycle geothermal plant. Their results illustrate that despite the slightly higher energetic performance of recuperative and regenerative systems, their higher capital costs inhibited their economic competitiveness and suggested that the standard cycles are more cost-efficient.

Bina et al. [20] evaluated four different ORC configurations, including a standard and a recuperative ORC, along with a regenerative cycle including an open-type pre-heater and a double stage system. They designed these cycles to use the geothermal outlet of the Sabalan flash cycle plant, located in Iran. Five different parameters were used to optimize the systems; the energetic efficiency, the exergetic efficiency, the net power output, the production cost, and the total cost. According to their results, the maximum was calculated for the recuperative. However, when considering the energy production cost and the total energy cost, the regenerative and the standard ORC were the best cycles.

Wang et al. [21] modeled a regenerative organic Rankine cycle for utilizing solar energy over a range of low temperatures, considering flat-plate solar collectors and thermal storage systems. They showed that system performance could be improved, under realistic constraints, by increasing turbine inlet pressure and temperature or lowering the turbine backpressure, and by using a higher turbine inlet temperature with a saturated vapor input. Zare [22] investigated and compared the performance of three configurations of ORC for binary geothermal power plants. The considered configurations are simple ORC, regenerative ORC, and ORC with an internal heat exchanger. His results illustrate that ORC with internal heat exchanger has the best performance from the thermodynamic point of view while simple ORC has the minimum cost among the considered cycles.

Literature summary shows that when a regenerator or a recuperator were separately added the system, the thermal efficiency of system increases. Although many studies have been done in the literature about ORC, there is relatively less research about regenerative and recuperative ORC system. Therefore, in this paper, a new configuration of regenerative and recuperative ORC is analyzed. The energy and exergy analysis of regenerative and recuperative organic Rankine cycle is performed for dry organic fluids. Hence, R245fa, R600, R114, and R600a are selected as working fluids. The organic Rankine cycle's performance parameters are evaluated to identify suitable working fluid which may yield high thermal and exergy efficiencies and net power production depending on varied evaporation temperatures and the inlet pressure of the low-pressure expander.

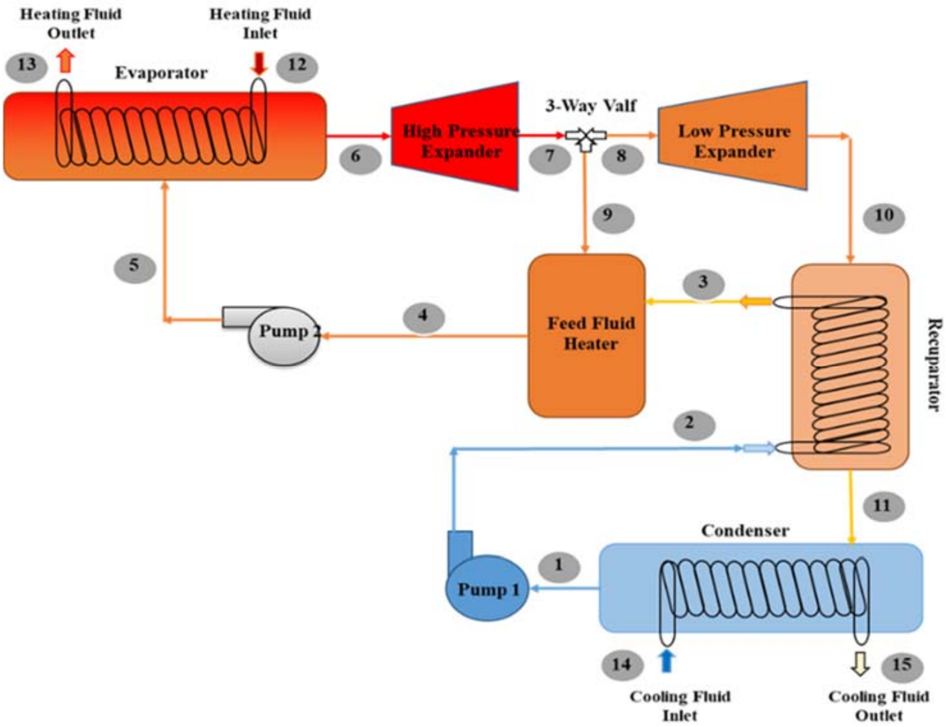

Fig.1. Schematic diagram of the ORC configuration

MATERIAL AND METHOD

The components of the organic Rankine cycle configuration are an evaporator, a high-pressure expander, a low-pressure expander, a recuperator, a feed fluid heater, a condenser, and two pumps system. Fig.1 presents a schematic diagram of the regenerative and recuperative ORC used to obtain energy from waste thermal energy. It is comprised of an evaporator, two expanders, a feed fluid heater (regenerator), a recuperator, a condenser, and two pumps. In the regenerator heat exchanger, heat is transferred between the high-temperature vapor from the high-pressure expander outlet and the low-temperature fluid from the recuperator outlet to avoid energy loss. In a regenerative and recuperative ORC, organic vapor enters the high-pressure expander at the evaporator pressure (6) and expands isentropically to an intermediate pressure (7). Some vapor is extracted at this state and routed to the feed fluid heater (9), while the remaining vapor (8) continues to expand isentropically to the condenser pressure in the low-pressure expander (10). The expanded vapor enters recuperator, and its heat is transferred to working fluid exiting from pump 1. Partially cooled vapor exits from the recuperator and enters the condenser (11). This vapor leaves the condenser at the condenser pressure (1). The condensed fluid enters to the pump 1, in which the pressure is raised to the recuperator pressure (2) and is routed to recuperator (3), and then working fluid enters feed fluid heater where it mixes with the vapor extracted from the high-pressure expander. The mixture leaves the heater as a saturated liquid at the feed fluid heater pressure (4). The pump 2 raises the pressure of the working fluid to the evaporator pressure (5). The cycle is completed by evaporating the working fluid in the evaporator (6) [23].

Dry fluids show better thermal efficiencies because they do not condense after the fluid goes through the expander. Therefore, R245fa, R600, R114, and R600a dry organic fluids are selected as the working fluids in this study. Besides, selected fluids have an ozone depletion potential (ODP) value of 0.0-0.7. The working fluids with lower global warming potential (GDP) to the greater one are R600a, R600, R245fa and R114 [24]. Tab.1 shows the thermo-physical properties of the selected fluids. It can be seen that R245fa has the highest value of critical temperature. It is followed by R600, R114, and R600a respectively.

The analysis of a regenerative and recuperative ORC based on thermodynamic laws and the energy, exergy analyses were performed for the working fluids investigated. For analysed ORC configuration, the considered assumptions and input parameters were made:

- ✓ All processes are operating at steady state.
- ✓ The thermal and friction losses in the pipes are negligible.
- ✓ The kinetic and potential energy changes are negligible.

Table 1. Thermo-physical properties of the selected fluids [25]

Fluids	Molecular mass	Formula	Maximum temperature	Maximum pressure	Critical temperature	Critical pressure	Critical density
	g/mol		K	MPa	K	MPa	kg/m^3
R245fa	134.05	$C_3F_5H_3$	440.00	200.00	427.01	3.65	519.43
R600	58.12	C_4H_{10}	575.00	12.00	425.00	3.80	228.00
R114	170.92	$C_2Cl_2F_4$	507.00	21.00	418.83	3.26	579.97
R600a	58.12	C_4H_{10}	575.00	35.00	407.70	3.63	225.50

- ✓ Pressure drops of working fluid in the evaporator and condenser is neglected.
- ✓ The heat loss from the ORC components is negligible.
- ✓ The isentropic efficiency of expanders η_{exp} and the pumps η_p are 0.80.
- ✓ The effectiveness of recuperator is 0.80.
- ✓ The atmospheric conditions are taken as 100 kPa and 293.15 K.
- ✓ The mass flow rate $\dot{m}_{hf}$ and the pressure P_{hf} of the hot fluid are 10 kg/s and 1 Bar, respectively.
- ✓ The mass flow rate $\dot{m}_{cf}$ and the pressure P_{cf} of the cold fluid are 30 kg/s and 1 Bar, respectively.
- ✓ The overheating in the evaporator and the sub-cooling in the condenser are 5 K.
- ✓ The mass flow rate of working fluid in the low-pressure expander $\dot{m}_{lpe}$ is 1 kg/s.
- ✓ The inlet pressure of low-pressure expander and the inlet pressure of feed fluid heater are equal to the exit pressure of the high-pressure expander in the three-way valve ($P_7 = P_8 = P_9$).
- ✓ The condensing temperature is 30^{o}C.

Engineering Equation Solver software is used to obtain the thermodynamic properties of working fluids and to analyze the regenerative and recuperative ORC system performance. For any steady-state control volume, by neglecting the potential and kinetic energy changes, the thermodynamic expressions of ORC configuration are given below [12-23].

General expression of mass, energy, and exergy balance equations are that;

Mass balance equation:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

Energy balance equation:

$$\dot{E}_{in} = \dot{E}_{out} \tag{2}$$

$$\dot{Q} + \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \tag{3}$$

Exergy balance equation:

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} - \dot{Ex}_d = \Delta \dot{Ex}_s \tag{4}$$

Where for a steady-state system, $\Delta \dot{Ex}_s$ is zero.

$$\dot{Ex}_{in} = \dot{Ex}_{out} \tag{5}$$

$$\dot{Ex}_{heat} + \dot{W} = \dot{Ex}_{out} - \dot{Ex}_{in} + \dot{I} \tag{6}$$

$$\dot{Ex} = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{7}$$

Where, subscripts in and out represent the inlet and exit states, $\dot{Q}$ is heat input, $\dot{W}$ is work input, $\dot{Ex}$ is exergy rate and $\dot{I}$ is the irreversibility rate.

The passed through the high-pressure turbine working fluid separates two parts in the three-way valve. The $\dot{m}_{ff}$ amount of the working fluid enters the feed fluid heater and the $\dot{m}_{lpe}$ amount of the working fluid enters the low pressure expander. These processes are showed with the number 7, 8 and 9 in the Fig.1. When practiced the mass balance equation in this section;

$$\dot{m}_{total} = \dot{m}_{ff} + \dot{m}_{lpe} \tag{8}$$

The streams that are mixing inside the feed fluid heater are at the same pressure. Consequently, since the mixing process occurs at an intermediate pressure level between the condensation and the expander inlet pressures, the regenerative and recuperative ORC requires the addition of an additional pump. Hence, it follows:

$$P_2 = P_3 = P_4 = P_7 = P_8 = P_9 \tag{9}$$

The expander power equals the sum of high pressure and low-pressure expander powers, and it can be defined by Eq. (10).

$$\dot{W}_{exp} = \dot{m}_{total}\eta_{exp}(h_6 - h_{7s}) + \dot{m}_{lpe}\eta_{exp}(h_8 - h_{10s}) \tag{10}$$

The inlet power to the pumps and the heat transfer rate to the working fluid in the evaporator can be calculated following equations.

$$\dot{W}_p = \frac{\dot{m}_{lpe}(h_{2s} - h_1)}{\eta_p} + \frac{\dot{m}_{total}(h_{5s} - h_4)}{\eta_p} \tag{11}$$

$$\dot{Q}_e = \dot{m}_{total}(h_6 - h_5) \tag{12}$$

The net output power from the regenerative and recuperative ORC and the thermal efficiency of ORC are expressed by Eq.(13) and Eq.(14) respectively.

$$\dot{W}_{net} = \dot{W}_{exp} - \dot{W}_p \tag{13}$$

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_e} = \frac{\dot{W}_{exp} - \dot{W}_p}{\dot{Q}_e} \tag{14}$$

The exergy efficiency of regenerative and recuperative ORC system is given by Eq. (15).

$$\eta_{exe} = \dot{W}_{net}/\dot{Ex}_{hf} \tag{15}$$

where, $\dot{Ex}_{hf}$ is the exergy supplied to the system in the evaporator from hot fluid and it can be calculated with Eq.(16).

$$\dot{Ex}_{hf} = \dot{m}_{hf}\left[\left(h_{12} - h_{hf_0}\right) - T_0\left(s_{12} - s_{hf_0}\right)\right] \tag{16}$$

The thermodynamic degree of perfection (TDP) of ORC configuration system and the ratio of the inlet pressure to the outlet pressure in the high-pressure expander and the low-pressure expander are obtained by the following equations.

$$TDP = \eta_{th}\left(1 - \frac{T_L}{T_H}\right)^{-1} \tag{17}$$

$$Rp_{hpe} = \frac{P_6}{P_7} \tag{18}$$

$$Rp_{lpe} = \frac{P_8}{P_{10}} \tag{19}$$

RESULTS AND DISCUSSIONS

In this part, the results of the thermodynamic analysis of regenerative and recuperative ORC system are given using R245fa, R600, R114, and R600a working fluids. Primarily, a comparison of the chosen working fluids utilizing energy and exergy analysis is performed, and the outcomes of calculation are given in Tab.2. The evaporation temperature in the evaporator of ORC configuration is taken as 130°C in the investigation. As seen in Tab.2, R245fa organic fluid has the highest thermal efficiency with approximately 17.4% while the minimum thermal efficiency is calculated for R600a with 16.1% among the all working fluids. R600 and R114 working fluids have a thermal efficiency of about 17.3% and 17.0%, respectively. When this performance parameter is evaluated with the critical temperature of the working fluid, it can be seen that when the critical temperature increases, the thermal efficiency of the regenerative and recuperative ORC improves. In other words, a working fluid with a higher critical temperature exhibits better thermal efficiency.

The highest exergy efficiency values are obtained for R600 and R600a working fluids with about 30.0% and 29.3% respectively. When this result is commented with Eq.(15), R600 working fluid has both the maximum net power and the maximum exergy supplied to the system. The exergy efficiency for R600 is higher because the net power generated is higher than other working fluids. The thermodynamic degree of perfection (TDP) values of the working fluids are obtained with decreasing order as R114, R245fa, R600, and R600a. The maximum value of TDP is calculated for R114 working fluid with about 56.6% at the evaporation temperature of 130°C. This is due to that the minimum inlet temperature of hot fluid with approximately 144°C is determined for R114 for boiling at 130°C in the evaporator. On the other hand, the lowest the total mass flow is obtained for the R600a.

The highest net power about 77.6 kW value is calculated for R600. The other properties, which are the heat input in the evaporator, recuperator heat and exergy supplied to the system by hot fluid, illustrate same order with the net power. For these properties, the fluids are in the following order from large to small values: R600, R600a, R245fa, and R114. The maximum pressure rate in the high and low-pressure expanders are computed for R600a and R245fa respectively among the fluid examined.

The evaporation temperature and the inlet pressure of working fluid to expander affect the performance analysis of an ORC. So, a detailed analysis of the effect of two features on system performance are evaluated in the following section of the paper. Firstly, the evaporation temperature is taken as from 90°C to the critical temperature of each working fluid, and its calculation results are presented in the following figures. The inlet pressure of the low-pressure expander is 1 MPa.

Table 2. The comparison of selected fluids for the regenerative and recuperative ORC

Fluids	η_t	η_{exe}	η_{II}	m_{total}	W_{net}	Q_{ev}	Q_{rec}	Ex_{hf}	Rp_{hpe}	Rp_{lpe}
	%	%	%	kg/s	kW	kW	kW	kW		
R245fa	17.35	18.31	56.62	1.384	41.23	237.6	28.58	225.23	2.339	5.644
R600	17.26	30.01	53.42	1.252	77.59	449.7	57.52	258.56	2.632	3.522
R114	17.01	13.25	57.34	1.314	27.47	161.5	26.67	207.35	2.464	4.003
R600a	16.11	29.28	53.35	1.212	63.51	394.2	45.02	216.92	3.430	2.474

Fig.2 illustrates changing thermal efficiency of regenerative and recuperative ORC system for all of working fluids. According to Fig.2, the thermal efficiency increases with the increment of the evaporation temperature. The maximum value of the thermal efficiency is calculated for R245fa working fluid with about 19% at the evaporation temperature of 153°C. In the lower evaporation temperatures, R600 and R600a working fluids show better thermal efficiency than R245fa and R114.

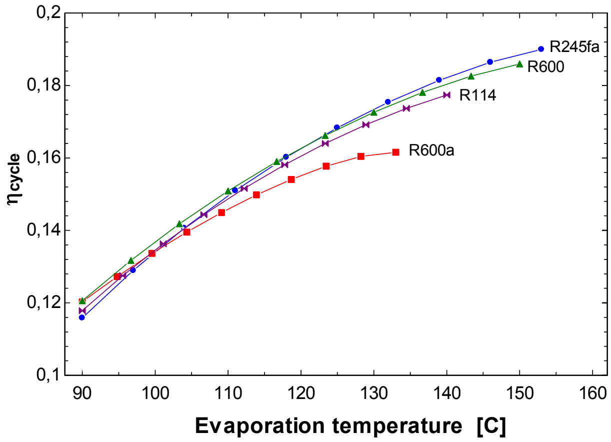

Fig.2. The effect of the evaporator temperature on thermal efficiency of ORC configuration

When examined the effect of the evaporator temperature on the exergy efficiency, which increases before and then decreases with the increase of the evaporation temperature for R245fa, R600 and R600a working fluids (Fig.3). However, this property decreases with increasing evaporation temperature for R114 fluid. Also, the highest exergy efficiency is obtained for the R600 working fluid with about 30.4% at the evaporation temperature of about 115°C. This is followed by R600a, R245fa, and R114, respectively. When this figure and Eq.(15) are commented, it can be shown that the effect of the exergy supplied to the system by hot fluid on the exergy efficiency is greater than the effect of the increment of the net power on the exergy efficiency. For this reason, the exergy efficiency of ORC configuration reduces despite the increment of the net power and exergy supplied to the system by hot fluid.

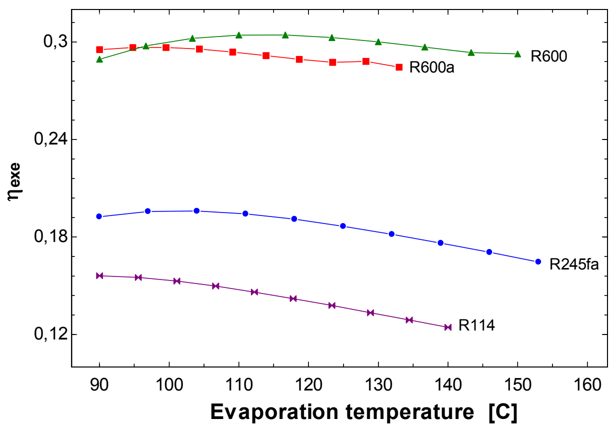

Fig.3. The effect of the evaporator temperature on exergy efficiency of ORC configuration

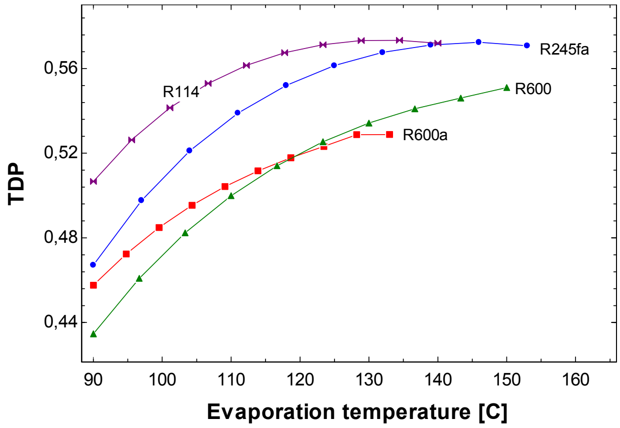

Fig.4. The effect of the evaporator temperature on the thermodynamic degree of perfection (TDP) of ORC configuration

Fig.4 demonstrates the variation of the system the thermodynamic degree of perfection (TDP) with increasing evaporation temperature for selected working fluids in the analysis of regenerative and recuperative ORC. TDP increases and then decreases with the increment of evaporation temperature. The maximum value of this property is calculated for R114 working fluid with about 57.3% at the evaporation temperature of 135°C. As Eq.(17) and Fig.4 are evaluated together, the reason why R114 has the highest TDP value that R114 working fluid requires a lower inlet temperature of hot fluid than the other fluids.

When compared the net power generated of the working fluids investigated in the system, according to Fig.5, the net power rises with the increase of the evaporation temperature, and it exhibits the same trend for all of the working fluids. The maximum net power of regenerative and recuperative ORC is calculated for R600 working fluid with about 85 kW in 150°C and which is followed by R600a, R245fa, and R114.

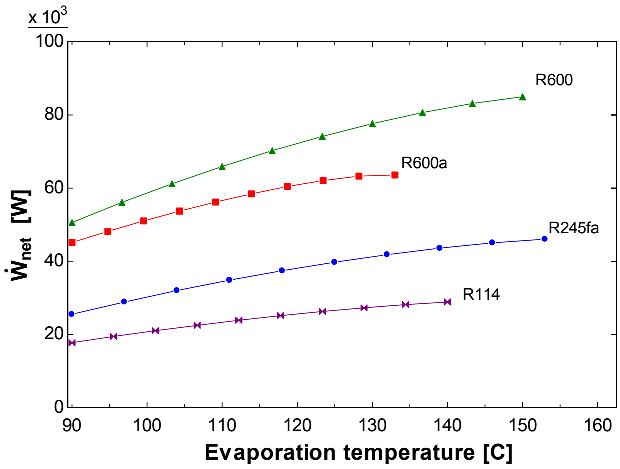

Fig.5. The effect of the evaporator temperature on the net power of ORC configuration

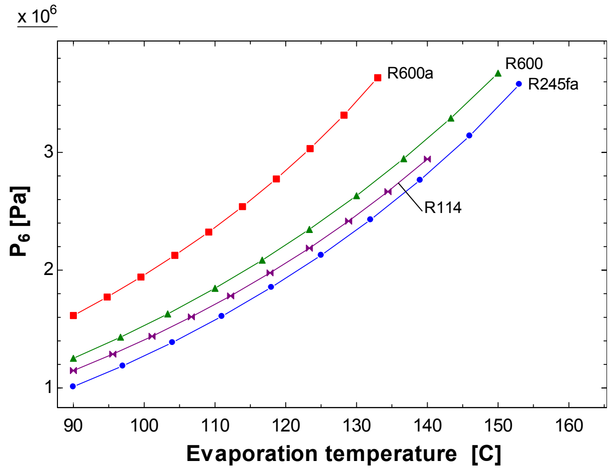

Fig.6. The effect of the evaporator temperature on the P_6 of ORC configuration

The influence of evaporation temperature on the inlet pressure of the high-pressure expander (P_6) are shown in Fig.6. According to the results of ORC configuration's analysis, the inlet pressure of the high-pressure expander rises with the increment of the evaporator temperature because of the increase of saturation pressure for all of the working fluids. It can be seen in the figure; the highest expander pressure value is obtained for R600a working fluid with 3.6 MPa in 133°C evaporation temperature which is followed by R600, R114, and R245fa working fluids. In the same time, rising the inlet pressure of the high-pressure expander will require a larger and more robust expander design.

The increasing evaporation temperature has a considerable effect on the exergy supplied to the system by hot fluid. Therefore, the change of this feature is similar to the net power and inlet pressure of high-pressure expander, raises with the increment of the evaporator temperature. As is seen in Fig.7, maximum exergy entry into the system occurs when R600 working fluid is used in the analyzed system.

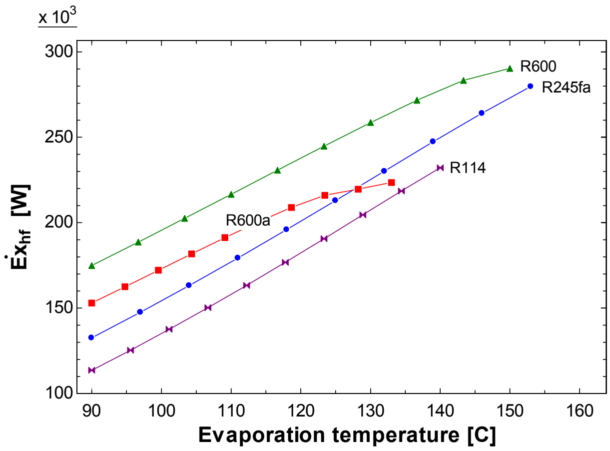

Fig.7. The effect of the evaporator temperature on the exergy supplied of ORC configuration

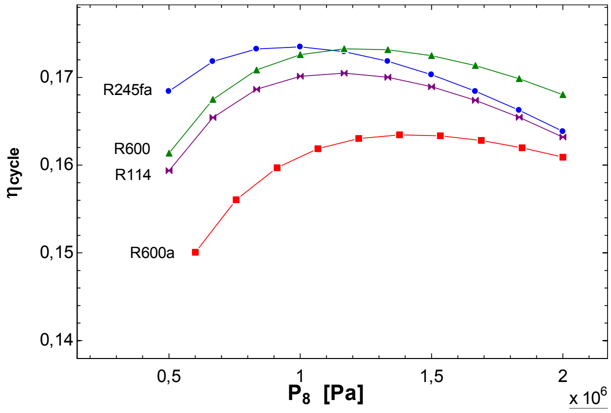

Fig.8. The effect of the inlet pressure of the low pressure expander (P_8) on thermal efficiency ORC configuration

Secondly, the effect of the inlet pressure of the low-pressure expander (P_8) is investigated on the performance parameters of the regenerative and recuperative ORC. The evaporation temperature in the evaporator is assumed as 130°C in the analysis. Moreover, the inlet pressure in the low-pressure expander, which equals to the pressure of feed fluid heater and the outlet pressure of the high-pressure expander, is increased from 0.5 MPa to 2.0 MPa and other performance parameters are calculated.

The Fig.8 presents the change of the thermal efficiency of the system. It can be seen that the thermal efficiency increases up to a maximum value with rising pressure, and then decreases for all of the working fluids in the analysis. Moreover, as the inlet pressure of the low-pressure expander raises, the difference between the calculated thermal efficiency values for the working fluids reduces and approaches each other. The maximum thermal efficiency is performed for R245fa working fluid with 17.4% in the 0.85 MPa of P_8 while the lowest thermal efficiency values belong to R600a.

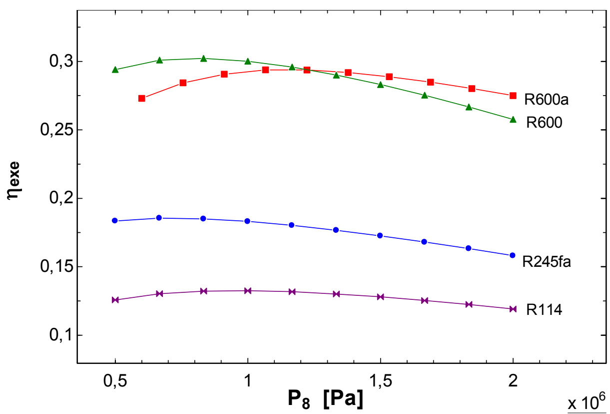

Fig.9. The effect of the P_8 on the exergy efficiency ORC configuration

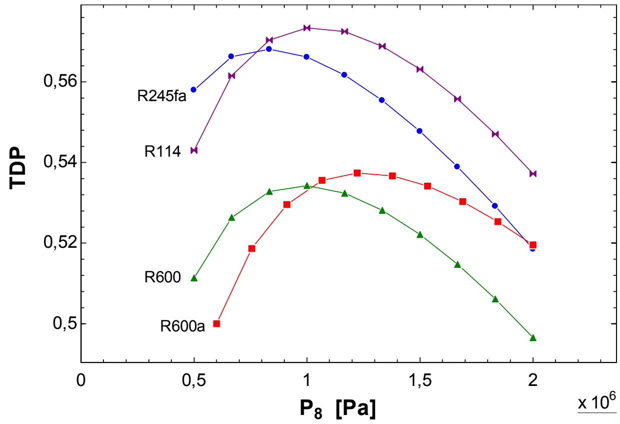

Fig.10. The effect of the (P_8) on the thermodynamic degree of perfection (TDP) ORC configuration

It can be clearly seen from Fig.9 and Fig.10 that the inlet pressure in the low-pressure expander has a positive effect before and then negative effect on exergy and the TDP. The exergy efficiency values of working fluids are calculated with decreasing order as R600a, R600, R245fa, and R114. The maximum value of exergy efficiency is obtained for R600 working fluid with about 30.2% at the 0.83 MPa of P_8. The reason for the decrease of the exergy efficiency is that the exergy supplied to the system in the evaporator increases with the increasing pressure while the net power of the system drops with a very low amount.

According to Fig.10, the maximum value of the thermodynamic degree of perfection (TDP) is 57.3% obtained about the nearly 1.0 MPa of P_8 for R114.

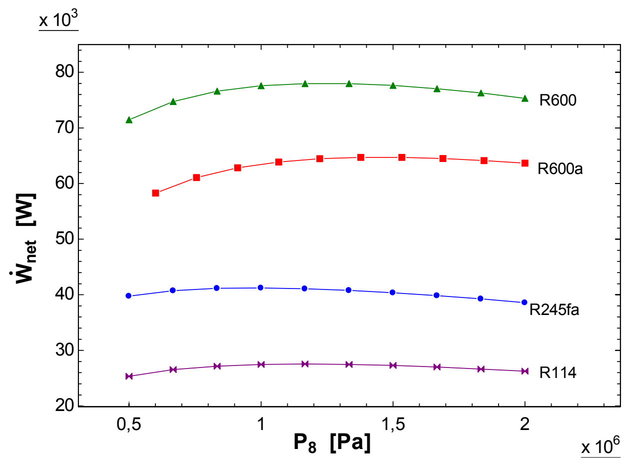

Fig.11. The effect of the (P_8) on the net power ORC configuration

When the effect of the inlet pressure of the low-pressure expander (P_8) on the net power is examined, it is observed that there is very little change in net power with the increment of P_8 (Fig.11). The highest value of the net power is obtained for R600 working fluid. It is calculated as about 78 kW with the inlet pressure of 1.17 MPa. R600a, R245fa, and R114 follow it with decreasing order.

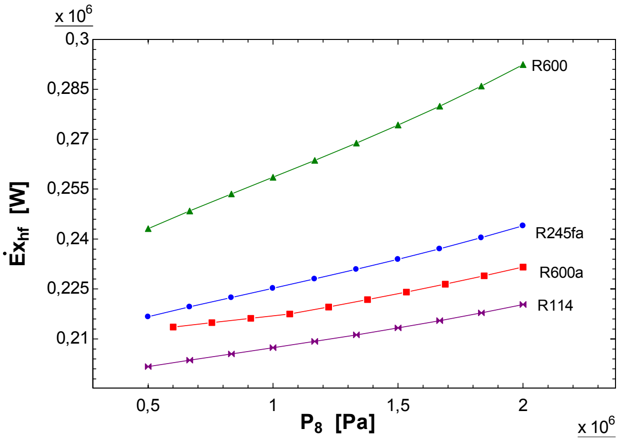

Fig.12. The effect of the (P_8) on the net power ORC configuration

Fig.12 demonstrates the effect of the inlet pressure in the low-pressure expander on the exergy supplied to the system by hot fluid. It can be seen from figure that this property has an inclining trend. Input exergy to the system, which is a measure of how much the hot source has been utilized, has the maximum value with 0.29 MW in the 2.0 MPa of P_8.

CONCLUSIONS

Thermodynamic analysis of a regenerative and recuperative organic Rankine cycle derived by a low grade heat source for power generation is performed in this study. Four dry organic fluid, namely R600a, R114, R600, and R245fa with critical temperature ranging from 134.6°C to 153.9°C, are chosen as working fluid in the analysis. The effects of the evaporation temperature and the inlet pressure of the low-pressure expander on performance parameters are evaluated, and their results are compared for the working fluid examined by using the Engineering Equation Solver model. Primarily, we have compared the selected dry refrigerants' performance parameters in the admitted conditions. The maximum thermal efficiency and thermodynamic degree of perfection (TDP) values are calculated for R245fa and R114 with %17.35 and %57.34, respectively, at the evaporation temperature of 130 °C. R600 refrigerant has maximum exergy efficiency, net power, evaporator, and recuperator heat rate, exergy supplied to the system by hot fluid properties with %30.01, 77.59 kW, 449.70 kW, 57.52 kW, and 258.56 kW, respectively. After that, we have investigated the influence of evaporation temperature on the performance parameters of the ORC configuration. According to analysis, in respect to the thermal efficiency, R245fa organic fluid has the best performance with the about 19% at the evaporation temperature of 153 °C. Besides, the thermal efficiency rises with increasing evaporation temperature for all of working fluids. However, when investigated the effects of the maximum power output, exergy efficiency, and the exergy supplied to the system by hot fluid values, R600 working fluid is the optimal fluid. Also, R600 is the more suitable working fluid for ORC configuration concerning expanders expansion ratio. After this analysis, the inlet pressure of the low-pressure expander has been examined, and the obtained results demonstrate that all analysed parameters have similar effects on the performance of the regenerative and recuperative ORC for all working fluids. In the low-pressure values, R245fa represents the best performance features with the thermal efficiency and thermodynamic degree of perfection (TDP). Similarly, R600 organic fluid has the maximum net power and exergy input from a hot source to the analysed system.

NOMENCLATURE

$\dot{E}$ - energy rate, kW;
$\dot{E}x$ - exergy rate, kW;
$\dot{I}$ - irreversibility rate, kW;
h - specific enthalpy, kJ/kg;
$\dot{m}$ - mass flow rate, kg/s;
P - pressure, Pa;
s - specific entropy, kJ/kgK;
$\dot{Q}$ - heat rate, kW;
$\dot{W}$ - power, kW;
T - temperature, K;
T_L - low temperature of sink, K;
T_H - high temperature of source, K;
η - efficiency;

Subscripts

cycle - cycle
d - destruction
e - evaporator
exe - exergetic
exp - expander
ff - feed fluid
hf - hot fluid
hpe - high pressure expander
in - inlet
lpe - low pressure expander
net - net
o - ambient
out- outlet

p - pump
s - system
th - thermal
total - total

Abbreviations

GWP-global warming potential;
ODP- ozone depletion potential;
ORC- organic Rankine cycle;
TDP- the thermodynamic degree of perfection;

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