Effects of evaporation parameters on recuperative transcritical organic Rankine cycle using binary mixture fluids

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Abstract

In this study, thermodynamic analysis was performed on basic and recuperative transcritical organic Rankine cycles by using five pure and six mixed fluids. The effects of evaporation parameters on the first- and second-law efficiencies ($\eta_I$ and $\eta_{II}$) as well as power output were investigated. The results indicate that a recuperator had a positive effect on the $\eta_I$ and $\eta_{II}$ and negative effects on the specific power. The total irreversibility of the system was improved by the recuperator. However, the total irreversibility considerably increased with an increase in the expander inlet temperature ($T_{\text{exp,in}}$) due to the significant increase in irreversibility in the condenser, particularly for working fluids with low critical temperatures, namely R134a, R1234yf and R290, and low proportions of R245fa and R600a in mixed fluids. For both the pure and mixed fluids, the specific power linearly increased with an increase in the expander inlet pressure ($P_{\text{exp,in}}$) and $T_{\text{exp,in}}$. However, with an increase in $P_{\text{exp,in}}$, the $\eta_I$ and $\eta_{II}$ first increased and then decreased. Finally, for $\eta_I$ and $\eta_{II}$, the effect of the recuperator increased with an increase in $T_{\text{exp,in}}$ even though the recuperator had a relatively small effect on the working fluids with high critical temperature, especially when $P_{\text{exp,in}}$ was high.

Keywords: transcritical organic Rankine cycle, recuperator, binary mixtures, first-law efficiency, second-law efficiency

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1. INTRODUCTION

Due to environmental concerns, engineering industries face the challenge of reducing fossil fuel usage and greenhouse gas emissions. A large quantity of the heat generated in these industries after burning processes is exhausted into the environment. Jouhara et al. [1] indicated that thermal processes account for 72% of the industrial energy use in the United Kingdom. A total of 31% of the heat energy generated in thermal processes is classified as low-grade heat. Therefore, low-grade recovery systems have become one of the key technologies for improving energy efficiency and reducing CO2 emissions.

Several heat-to-power conversion technologies, such as technologies based on the organic Rankine cycle (ORC) [2], transcritical ORC (TRC) [3], trilateral flash cycle and Kalina cycle [4], have been used to enhance energy efficiency in low-grade heat sources. Theoretically, the Kalina cycle can provide 15% more output power than the ORC; however, when the operation conditions related to the cooling source are considered, the difference in performance is only 3% in favour of the Kalina cycle [5]. Systems based on the ORC are simple and require limited maintenance. However, ORCs are characterised by high exergy destruction during heat transfer in the condenser and evaporator. The isothermal irreversibility and exergy loss of the heat source can be reduced using TRCs [6]. Song et al. [7] examined the effects of the dryness and the critical temperature on the TRC performance. Xia et al. [8] found that an increase of the expander inlet pressure can improve the exergy efficiency. Tian et al. [9] proposed a rule for selecting the working fluid under different heat source temperatures. Ya˘glı et al. [10] indicated that the performance of TRCs was higher than that of subcritical ORCs. Zhi et al. [11] found that the energy and exergy efficiencies increase and decrease with an increase in heat source temperature, respectively. Xu et al. [12] investigated the performance of the subcritical and
supercritical cycles with 12 pure working fluids and various heat sources. Their results indicate that a high net shaft power can be obtained when the inlet temperature of the flue gas is higher than the working fluid’s critical temperature. Li et al. [13] stated the effects and suitable conditions for the subcritical and transcritical ORCs with R1234ze. Meng et al. [4] analysed the thermodynamic performance for four cases. Their results indicated that among the four cases, the TRC with reheating had the best thermodynamic performance. Walnum et al. [14] found that both ORC and TRC need an advanced control strategy to avoid non-feasible operation and significant losses in work output.

The efficiency can be improved using an internal heat exchanger (IHE; i.e. a recuperator). Mago et al. [15] found that the recuperative ORC not only increases the first- and second-law efficiencies but also reduces the irreversibility. Vélez et al. [16] indicated that the energy efficiency and heat flow of the evaporator are higher when using an IHE than when not using an IHE at a given pressure. Shu et al. [17] indicated that the system efficiency of recuperative ORCs with mixed fluids substantially increases with an increase in the evaporator temperature. Wang and Zhao [18] indicated that the system efficiency of recuperative ORCs for mixed fluids can be significantly increased by superheating the system.

From the perspective of environmental protection, CO$_2$ is a promising natural working fluid in geothermal systems, power cycle, heating, cooling and refrigeration systems ([19–22]). Therefore, researchers have studied the TRCs of CO$_2$ and its mixtures to investigate how the CO$_2$–mixture system performance is affected by the operating parameters ([23–27]). Vélez et al. [16,28] performed a theoretical examination on a CO$_2$ TRC and found that the irreversibilities of the evaporator and condenser were reduced by an increase in the operating pressure. This result was obtained because the fluid’s pump outlet temperature increased, which caused an increase in the exergy efficiency. For a CO$_2$ TRC performed without a regenerator in a pressure range of 9–15 MPa, Gayer et al. [29] found that the maximal thermal and exergy efficiencies occurred at $\sim$13.5 MPa. The operating conditions affecting the turbine performance of Brayton cycles and CO$_2$ TRCs have been investigated [30–32]. Meng et al. [4] indicated that the economic performance of the CO$_2$ TRCs, which has a higher net power output than the Kalina cycle and ORC, is between that of the Kalina cycle and ORC. Although CO$_2$–based mixtures have been investigated in the literature, the associated system costs are markedly high because the supercritical pressure of CO$_2$ is higher than that of other organic mixtures.

Although R245a and R134a have high global warming potential (GWP), these chemicals are popular and widely used as working fluids in ORC and TRC systems. Hydrocarbon (namely R600a and R290) and hydrofluorocarbons (i.e. R1234yf) refrigerants, which have low GWP and high flammability, are non-ozone-depleting, non-toxic and environmentally friendly materials. From the viewpoint of environmental protection, R600a, R290 and R1234yf are promising working fluids for ORCs and TRCs. However, the performance of these fluids is inferior to that of R245a in low-grade-heat TRCs. In particular, the GWP and properties of organic fluids can be adjusted by varying the mole fractions of the constituents of various working fluid mixtures to improve the environmental friendliness and performance of the system. The objectives of the present study were to examine the first- and second-law efficiencies as well as specific power of basic and recuperative TRCs at expander inlet temperatures ranging from 150 to 200°C. Moreover, both configuration TRC performances of pure and mixed fluids were evaluated under different evaporation pressures.

1.1. Theoretical analysis

The schematics of the basic and recuperative TRC systems are displayed in Figure 1. Two refrigerants, namely R245fa and R600a, were used as working fluids and mixed with R134a, R1234yf and R290 to investigate the effects of the evaporation parameters and recuperator on the TRC performance. Figure 2 displays the T–s diagram for the TRC system. The diagram indicates that temperature glide in the condenser occurred for the working fluid two-phase region. The temperature glide provided a superior temperature match between the cooling source and the working fluid during the condensation process. The irreversibility was low during the condensation of the working fluid. Therefore, the exergy efficiency could be improved. The thermophysical properties of the working fluid were obtained from REFPROP 9.1 [33], and a thermodynamic model was coded in MATLAB software. Table 1 lists the parameters of the TRCs associated with the mixtures and heat source.

The heat transfer rate and irreversibility of the evaporator and condenser as well as the mass flow rate of refrigerants in the mathematical model were different between the basic and recuperative TRCs. Table 2 lists the equations for computing the heat transfer rate and irreversibility for each component of the basic and recuperative TRCs. Moreover, the expander shaft power and pumping power of the basic and recuperative TRCs were estimated using the following equations:

\[ \dot{W}_{\text{pump}} = \dot{m}_f (h_2 - h_1) \]  
\[ \dot{W}_{\text{exp}} = \dot{m}_f (h_4 - h_3) \]

The net power is the shaft power generated by the expander minus the pumping power wasted by the working fluid pump as follows:

\[ \dot{W}_{\text{net}} = \dot{W}_{\text{exp}} - \dot{W}_{\text{pump}} \]

To consider the specific power, we express the net output power per unit heat source mass flow rate as follows:

\[ \zeta = \frac{\dot{p}_{\text{net}}}{\dot{m}_h} = \frac{(\dot{W}_{\text{exp}} \eta_{g,m}) - \dot{W}_{\text{pump}}}{\dot{m}_h} \]

where $\eta_{g,m}$ is the efficiency of electrical–mechanical conversion for the motor of the pump and the generator.
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Figure 1. Schematic of the (a) basic and (b) recuperative TRCs.

Figure 2. T–s diagrams of the (a) basic and (b) recuperative TRCs.

Table 1. Parameters of the TRC.

| Parameter                                         | Value       |
|---------------------------------------------------|-------------|
| Expander isentropic efficiency                    | η_{exp} 0.8 |
| Pump isentropic efficiency                        | η_{pump} 0.65 |
| Recuperator effectiveness                         | ε_{rec} 0.8 |
| Generator/motor efficiency                        | η_{g,m} 0.9 |
| Evaporation temperature pinch point               | ΔT_{pp,eva} 5 K |
| Condense temperature pinch point                  | ΔT_{pp,cond} 5 K |
| Expander inlet temperature                        | T_{exp,in} 150–200°C |
| Expander inlet pressure                           | P_{exp,in} 4.1–6.9 MPa |
| Heat source inlet temperature                     | T_{h,in} 160–210°C |
| Heat source mass flow rate                        | \dot{m}_h 30 kg/s |
| Cooling source inlet temperature                  | T_{c,in} 32°C |
| Ambient temperature                               | T_0 32°C |

The first- and second-law efficiencies of thermodynamics are represented as energy and exergy conversion efficiencies, respectively. The efficiencies are estimated using the following equations:

\[
\eta = \frac{\dot{W}_{net}}{\dot{Q}_{eva}} 
\]

\[
\eta_{II} = \frac{\dot{W}_{net}}{\Delta\dot{E}_h} 
\]

where \(\dot{Q}_{eva}\) is the rate of heat transfer from the heat source to the working fluid in the evaporator, as presented in Table 2. The parameter \(\Delta\dot{E}_h\) can be determined by computing the difference between \(\dot{E}_{h,in}\) and \(\dot{E}_{h,out}\) by using the following equations:

\[
\dot{E}_{h,in} = \dot{m}_h (h_{h,in} - h_0) - \dot{m}_h T_0 (s_{h,in} - s_0) 
\]

\[
\dot{E}_{h,out} = \dot{m}_h (h_{h,out} - h_0) - \dot{m}_h T_0 (s_{h,out} - s_0) 
\]

\[
\Delta\dot{E}_h = \dot{E}_{h,in} - \dot{E}_{h,out} = \dot{m}_h \left[ (h_{h,in} - h_{h,out}) - T_0 (s_{h,in} - s_{h,out}) \right] 
\]

where \(\dot{E}_{h,in}\) and \(\dot{E}_{h,out}\) represent the exergy of the expander inlet and outlet in the evaporator, respectively. The normalised exergy loss is expressed as follows:

\[
\psi_{loss} = \frac{\dot{E}_{h,out}}{\dot{E}_{h,in}} 
\]

To clarify the effects of the evaporation parameters (i.e. the expander inlet pressure and temperature) on each component of the TRC, the equations of normalised irreversibility for the components are listed in Table 2. The normalised total irreversibility of the TRC system is summarised as follows:

\[
\Omega_{total} = \Omega_{pump} + \Omega_{eva} + \Omega_{exp} + \Omega_{cond} + \Omega_{rec} 
\]
Table 2. Mathematical model of the components for the basic and recuperative TRCs.

| Component | Basic | Recuperative |
|-----------|-------|--------------|
| \(\dot{Q}_{eva}\) | \(\dot{Q}_{eva} = \dot{m}_f (h_3 - h_2)\) | \(\dot{Q}_{eva} = \dot{m}_f (h_3 - h_7)\) |
| \(\dot{Q}_{cond}\) | \(\dot{Q}_{cond} = \dot{m}_f (h_4 - h_1)\) | \(\dot{Q}_{cond} = \dot{m}_f (h_4 - h_1)\) |
| \(\dot{Q}_{rec}\) | NA | \(\dot{Q}_{rec} = \dot{m}_f (h_4 - h_8) = \dot{m}_f (h_7 - h_2)\) |
| \(m_f\) | \(m_f = \frac{\dot{m}_h (h_{in} - h_{out})}{h_5 - h_2}\) | \(m_f = \frac{\dot{m}_h (h_{in} - h_{out})}{h_5 - h_2}\) |
| \(\dot{I}_{eva}, \Omega_{eva}\) | \(\dot{I}_{eva} = \dot{m}_f T_s (S_3 - S_2) + \dot{m}_h T_s (S_{h, out} - S_{h, in})\) | \(\dot{I}_{eva} = \dot{m}_f T_s (S_3 - S_2) + \dot{m}_h T_s (S_{h, out} - S_{h, in})\) |
| \(\dot{I}_{exp}, \Omega_{exp}\) | \(\dot{I}_{exp} = \dot{m}_f T_s (S_4 - S_3)\) | \(\dot{I}_{exp} = \dot{m}_f T_s (S_4 - S_3)\) |
| \(\dot{I}_{cond}, \Omega_{cond}\) | \(\dot{I}_{cond} = \dot{m}_f T_s (S_1 - S_4) + \dot{m}_h T_s (S_{c, out} - S_{c, in})\) | \(\dot{I}_{cond} = \dot{m}_f T_s (S_1 - S_4) + \dot{m}_h T_s (S_{c, out} - S_{c, in})\) |
| \(\dot{I}_{pump}, \Omega_{pump}\) | \(\dot{I}_{pump} = \dot{m}_f T_s (S_2 - S_1)\) | \(\dot{I}_{pump} = \dot{m}_f T_s (S_2 - S_1)\) |
| \(\dot{I}_{rec}, \Omega_{rec}\) | NA | \(\dot{I}_{rec} = \dot{m}_f T_s (S_7 + S_8 - S_2 - S_4)\) |

The normalised total irreversibility is affected by the irreversibility of the components and the exergy, which is related to the outlet temperature of the heat source. The normalised equation can be used to determine the weight of the irreversibility in the components of the TRC cycle for comparing the pure and mixed working fluids.

To validate the thermodynamic analysis model developed in the present study, the model was solved at the same operating conditions and working fluid [34]. According to the validation results, the relative deviation in the net shaft power, first-law efficiency, second-law efficiency and irreversibility were 1.48%, 0.7%, 1.9% and 0.9%, respectively.

2. RESULTS AND DISCUSSION

In the present study, theoretical analysis was performed to examine the effects of five pure and six mixed fluids (namely R245fa, R600a, R134a, R1234yf, R290, R245fa/R134a, R245fa/R1234yf, R245fa/R290, R600a/R134a, R600a/R1234yf and R600a/R290) on the first- and second-law efficiencies as well as specific power of the basic and recuperative TRC configurations at various operating parameters. The effects of \(P_{exp, in}\) (the expander inlet pressure) on the TRC system performance at expander inlet temperatures \((T_{exp, in})\) of 150°C and 180°C are illustrated in Figures 3 and 4, respectively. The results indicate that R290 and R134a were not drawn at low \(P_{exp, in}\) because these fluids were in the subcritical state, in which the critical pressure of the pure fluid is higher than the expander inlet pressure. For TRCs, the critical point of the working fluid is a crucial property because the fluid must be in the supercritical state before entering the expander. Moreover, the system performance of pure R600a and R245fa could not be evaluated at high inlet pressures and low inlet temperatures (i.e. 150°C and 180°C) because when the critical temperature \((T_{crit})\) approached the inlet temperature, droplets were generated during the expansion process at high expander inlet pressure. To avoid droplet generation due to a drop in pressure during expansion, the expander inlet temperature of the working fluid should be sufficiently higher than the critical temperature for ensuring that the working fluid remains in the superheated state.

Figure 3 indicates that the efficiencies and specific power first marginally increased and then decreased with an increase in \(P_{exp, in}\) at a \(T_{exp, in}\) value of 150°C. For R600a, the highest efficiency and lowest specific power were observed at low \(P_{exp, in}\) values. However, the peak values of \(n_1\) and \(\eta_{II}\) for R134a occurred at a \(P_{exp, in}\) value of ~5.5 MPa when a recuperator was used. At an inlet temperature of 150°C, R134a and R1234yf had high efficiencies and a wide operating range. Moreover, the specific powers of all the working fluids were lower in the recuperative TRC than in the basic TRC because the preheating of the working fluid by the recuperator resulted in a low mass flow rate of the working fluid.

When \(T_{exp, in}\) was increased to 180°C, the second-law efficiency and specific power increased with the \(P_{exp, in}\) value, except in the case of the second-law efficiency for R245fa and R600a, as presented in Figure 4. Figure 5 indicates that the first-law efficiency and specific power dramatically increased with an increase in the \(T_{exp, in}\) value. However, with an increase in \(T_{exp, in}\), the second-law efficiency of the working fluid with low \(T_{crit}\) (R134a, R1234yf and R290) had opposite trend. This result was obtained because the irreversibility in the condenser at high \(T_{exp, in}\) was higher than that at low \(T_{exp, in}\). Compared with the corresponding results obtained at 150°C, the \(n_1\) and \(\eta_{II}\) values were notably improved by the recuperator at a \(T_{exp, in}\) of 180°C. However, the recuperator had a slight effect on the specific power at a given \(T_{exp, in}\). A
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Figure 3. Effects of $P_{\text{exp,in}}$ on the (a) $\eta_I$, (b) $\eta_{II}$ and (c) $\zeta$ values of the pure fluids at a $T_{\text{exp,in}}$ value of 150°C.

Figure 4. Effects of $P_{\text{exp,in}}$ on the (a) $\eta_I$, (b) $\eta_{II}$ and (c) $\zeta$ values of the pure fluids at a $T_{\text{exp,in}}$ value of 180°C.

Comparison of Figures 3 and 4 shows the differences with R245fa between two TRC configurations for the values of $\eta_I$ and $\eta_{II}$ were 2.9% and 6.9%, respectively, at a $P_{\text{exp,in}}$ value of 3.9 MPa. These different values were higher than those obtained at a $P_{\text{exp,in}}$ value of 5.5 MPa (the differences for the $\eta_I$ and $\eta_{II}$ were 1.0% and 1.7%, respectively). The trends of R600a were similar to those of R245fa. In conclusion, the recuperator had a relatively small influence on the working fluids with a high $T_{\text{cri}}$, especially when the expander inlet pressure was high. In contrast to working fluid with high $T_{\text{cri}}$, the recuperator had substantial influences on the efficiencies of the working fluids with a low $T_{\text{cri}}$ (i.e. R134a, R1234yf and R290) at high $P_{\text{exp,in}}$, as displayed in Figure 4a and b.

The trends of the $\eta_I$, $\eta_{II}$ and $\zeta$ at the expander inlet temperature of 200°C were similar to those at $T_{\text{exp,in}}$ of 180°C. Meanwhile, $\eta_I$ and $\eta_{II}$ as well as $\zeta$ of R245fa and R600a drastically increased when $T_{\text{exp,in}}$ increased from 150 to 200°C, as shown in Figure 5.

The efficiencies and specific power of the six mixed fluids with optimal mole fractions at various $T_{\text{exp,in}}$ and $P_{\text{exp,in}}$ values are displayed in Figures 6–8. The results indicate that the system performance was affected by various operating parameters at the optimal mole fractions corresponding to the maximal specific power, except R245fa/R290 and R600a/R1234yf. The optimal mole fraction of the mixed fluids was shown in the legend of the figure. However, the mole fractions of R245fa/R290 and R600a/R1234yf were not equal to the optimal values because the optimal mole fraction of R245fa/R290 only occurred for pure R245fa when $T_{\text{exp,in}}$ was higher than 180°C. Additionally, the optimal mole fraction of R600a/R1234yf varied with $T_{\text{exp,in}}$. The optimal fractions of R600a/R1234yf with both TRC configurations were 0.45/0.55, 0.29/0.71, 0.37/0.63, 0.48/0.52, 0.59/0.41 and 0.78/0.22 at $T_{\text{exp,in}}$ values of 150°C, 160°C, 170°C, 180°C, 190°C and 200°C, respectively. The optimal mole fraction was affected by the $T_{\text{exp,in}}$ value due to the effects of the difference between the critical temperature of the mixed fluid and the expander inlet temperature ($\Delta T_{\text{inlet,cri}}$). As displayed in Figure 6, R245fa and R600a were low proportion in the optimal fraction to remain a suitable
$\Delta T_{\text{inlet,cri}}$ under the effects of condensation temperature glide, when $T_{\text{exp,in}}$ is low. With an increase in $T_{\text{exp,in}}$, the proportion of R245fa and R600a in the mixed fluids increased, as depicted in Figures 6–8.

Figure 6 shows that when the expander inlet pressure was higher than 5.2 MPa, the efficiencies obtained with the recuperator decreased with an increase in the $P_{\text{exp,in}}$ at a $T_{\text{exp,in}}$ value of 150°C. However, with an increase in $T_{\text{exp,in}}$, the degree of decline in the aforementioned efficiencies was reduced and effects of the $P_{\text{exp,in}}$ on the efficiencies were slight, as displayed in Figures 7 and 8. A comparison of the different working fluids indicates that the specific powers of the R245fa- and R600a-based mixed fluids, except R245fa/R290, were higher than those of the pure fluids (as shown in Figures 5c and 8c), especially as the expander inlet temperature at 200°C. Thus, R245fa/R290 is an unsuitable fluid for TRCs.

Table 3 lists the maximal values of $\eta_1$, $\eta_II$ as well as $\zeta$ corresponding to working fluid and operating parameters. The results show that the pure R600a and R245fa had maximal first-law efficiency at $T_{\text{exp,in}} \leq 180$°C and $T_{\text{exp,in}} \geq 190$°C, respectively, at relative low expander inlet pressure. However, the maximal specific power could be observed in R245fa/R134a (0.17/0.83) and R245fa/R1234yf (0.97/0.03) of the mixed fluids at $T_{\text{exp,in}} \leq 180$°C and $T_{\text{exp,in}} \geq 190$°C, respectively. Furthermore, the trend of second-law efficiency was similar to that of specific power. It is worth noting that the operating pressures corresponding to the maximal values of the specific power and the second-law efficiency were higher than that of the first-law efficiency. A comparison of Figure 5 and Table 3 indicates, for $T_{\text{exp,in}} > 190$°C, R245fa and R245fa-based mixture exhibited the maximum efficiency and specific power in both TRC configurations.

To understand how the irreversibility trend of each component was affected by the inlet expander temperature, the irreversibilities of the components in R134a, R1234yf, R290, R245fa and R600a at $P_{\text{exp,in}}$ of 5.5 MPa are presented in Figure 9. As displayed in Figure 9a, b and c, the irreversibility of the condenser for pure working fluid with low $T_{\text{cri}}$ notably increased with $T_{\text{exp,in}}$, however, it can be substantially reduced by the recuperator. Therefore, the total irreversibility could be improved as TRC adopting recuperator.

Figure 10 illustrates the effects of the expander inlet pressure on the total reversibility of the R245fa- and R600a-based mixed fluids. The results indicate that the recuperator had a positive effect on the total irreversibility of the system. Moreover, the total irreversibility of the system could be effectively reduced by decreasing $P_{\text{exp,in}}$, especially as low $P_{\text{exp,in}}$. However, the effects of $T_{\text{exp,in}}$ on irreversibility depended on the mole fractions of the mixtures. When the mole fractions of R245fa/R134a and R245fa/R1234yf were 0.95/0.05 and 0.97/0.03, respectively, they had the lowest total irreversibilities (lower than 0.36) among the mixed fluids at 180°C and 200°C (as shown in Figure 10a and b). For a low proportion of R245fa in the mixed fluids, the irreversibility was increased with an increase in $T_{\text{exp,in}}$ and higher than 0.36. The trends of total irreversibility for the R600a-based mixed fluids were similar to those of the R2545fa-based mixed fluids in both TRC configurations. However, the total irreversibilities of the R2545fa-based mixed fluids, except R245fa/R290, were lower than those of the R600a-based mixed fluids.

Figure 11 shows the effects of the inlet pressure on the irreversibility of the components in R245fa/R1234yf. A comparison of Figures 10 and 11 indicates that the total irreversibility of R245fa/R1234yf and R245fa/R134a increased with an increase in $T_{\text{exp,in}}$ because the irreversibilities of the condenser and recuperator for the basic and recuperative TRCs, respectively, were notably affected by $T_{\text{exp,in}}$. For low $T_{\text{exp,in}}$ values, the total irreversibility first decreased and then increased with an increase in $T_{\text{exp,in}}$ (Figure 10). This result was obtained because although the irreversibilities of heat exchangers (i.e. the evaporator, condenser and
Figure 6. Effects of $P_{\text{exp,in}}$ on the (a) $\eta_I$, (b) $\eta_{II}$ and (c) $\zeta$ values of the mixed fluids at a $T_{\text{exp,in}}$ value of 150$^\circ$C.

Figure 7. Effects of $P_{\text{exp,in}}$ on the (a) $\eta_I$, (b) $\eta_{II}$ and (c) $\zeta$ values of the mixed fluids at a $T_{\text{exp,in}}$ value of 180$^\circ$C.

Figure 8. Effects of $P_{\text{exp,in}}$ on the (a) $\eta_I$, (b) $\eta_{II}$ and (c) $\zeta$ values of the mixed fluids at a $T_{\text{exp,in}}$ value of 200$^\circ$C.
Figure 9. Effects of $T_{\text{exp,in}}$ on the irreversibility of the components in the basic TRC at a $P_{\text{exp,in}}$ value of 5.5 MPa for (a) R134a, (b) R1234yf, (c) R290, (d) R245fa and (e) R600a.
Table 3. The maximal values corresponding to working fluid and operating pressure at different $T_{\text{exp,in}}$ values in recuperative TRC.

| $T_{\text{exp,in}}$ (°C) | Fluid                  | $\eta_{\text{I}}$ (%) | Max. value | $P_{\text{exp,in}}$ (MPa) |
|-------------------------|------------------------|------------------------|------------|--------------------------|
| 150°                   | R600a                  | 14.34                  | 3.7        |                          |
| 160°                   | R245fa/R134a (0.17/0.83)| 60.01                  | 5.3        |                          |
| 170°                   | R245fa/R134a (0.17/0.83)| 40.24                  | 6.7        |                          |
| 180°                   | R245fa/R134a (0.17/0.83)| 15.20                  | 3.8        |                          |
| 190°                   | R245fa/R1234yf (0.97/0.03)| 61.89                  | 6.9        |                          |
| 200°                   | R245fa/R1234yf (0.97/0.03)| 55.13                  | 6.9        |                          |

Figure 10. Effects of $P_{\text{exp,in}}$ on the total irreversibilities of (a) R245fa/R134a, (b) R245fa/R1234yf, (c) R245fa/R290, (d) R600a/R134a, (e) R600a/R1234yf and (f) R600a/R290.
Figure 11. Irreversibility distribution of R245fa/R1234yf at $T_{\text{exp, in}}$ values of (a) 150°C, (c) 180°C and 200°C.

Figure 12. Exergy loss from the heat source: (a) R245fa/R134a, (b) R245fa/R1234yf, (c) R245fa/R290, (d) R600a/R134a, (e) R600a/R1234yf and (f) R600a/R290.
recuperator) were marginally reduced by $P_{\text{exp,in}}$ at high $P_{\text{exp,in}}$, the weights of the expander and pump for the irreversibility notably and linearly increased with an increase in $P_{\text{exp,in}}$, as shown in Figure 11a.

The exergy loss from the heat source for the different mixed fluids affected by varying operating parameters is shown in Figure 12. The results indicate that the exergy loss could be reduced by increasing the $P_{\text{exp,in}}$ and $T_{\text{exp,in}}$ values. However, the exergy loss of the recuperative TRC was increased with an increase in $T_{\text{exp,in}}$ value. The exergy loss of the basic TRC was lower than that of the recuperative TRC because of the low outlet temperature of the heat source in the basic configuration. For R245fa/R1234yf, R600a/R134a and R600a/R1234yf in the basic TRC, $\psi_{\text{loss}}$ notably decreased with an increase in $P_{\text{exp,in}}$. At $P_{\text{exp,in}} > 5.2 \text{ MPa}$, $\psi_{\text{loss}}$ approached $\sim 0.03$. Similarly, the $\psi_{\text{loss}}$ adopting recuperator decreased to a minimal value at high $P_{\text{exp,in}}$. Moreover, the trends of $\psi_{\text{loss}}$ of R245fa/R134a and R245fa/R1234yf at $T_{\text{exp,in}}$ of 190°C and 200°C were different from that at lower than $T_{\text{exp,in}}$ of 180°C. This was because the proportion of R245fa in the optimal mole fraction at $T_{\text{exp,in}}$ of 190°C and 200°C is different under different $T_{\text{exp,in}}$. The values of $T_{\text{cri}}$, which depended on the mole fraction of the mixed fluids, affected the thermal match between the heat source and the working fluid and consequently influenced the outlet temperature of the heat source in the evaporator. Therefore, as displayed in Figure 12, excellent thermal match was achieved in the TRC without the recuperator at high $P_{\text{exp,in}}$ and $T_{\text{exp,in}}$, and working fluid with low $T_{\text{cri}}$.

3. CONCLUSIONS

In this study, thermodynamic analysis was conducted on a basic TRC system and a recuperative TRC system with the pure and mixed fluids. R245fa and R134a, which are popular working fluids in TRCs and ORCs, were mixed with low-GWP working fluids, for example R600a, R290 and R1234yf. The major results obtained here are briefly summarised as follows:

1. Although the recuperator had a positive effect on $\eta_I$ and $\eta_{II}$, the $\zeta$ value of the recuperative TRC was lower than that of the basic TRC because the working fluid mass flow rate in the evaporator was reduced by the recuperator at a given constant heat source mass flow rate.

2. For both pure and mixed fluids, the $\zeta$ and $\eta_{II}$ increased with an increase in $P_{\text{exp,in}}$ and $T_{\text{exp,in}}$. However, $\eta_{II}$ was decreased with an increase of $T_{\text{exp,in}}$ because of the total irreversibility considerably increased with $T_{\text{exp,in}}$, particularly for the working fluids with a low $T_{\text{cri}}$ and low proportion of R245fa and R600a in mixed fluids. Furthermore, the influence of the recuperator on $\eta_{II}$ increased with an increase in $T_{\text{exp,in}}$; however, the recuperator had relatively small effects on working fluids with a high $T_{\text{cri}}$ (i.e. R245fa and R600a), especially when the expander inlet pressure was high.

3. The irreversibilities in the evaporator and condenser were improved by the recuperator, especially for irreversibility of the condenser. Additionally, the irreversibilities of evaporator, condenser and recuperator could be improved by $P_{\text{exp,in}}$. However, irreversibilities of pump and expander were linearly increased with an increase of $P_{\text{exp,in}}$.

4. The exergy losses for basic and recuperative TRCs were decreased and increased with an increase of $P_{\text{exp,in}}$, respectively. However, for both configurations, the exergy loss could be notably decreased by $P_{\text{exp,in}}$.

5. Based on the analysis data, the pure R600a and R245fa had the maximal values of $\eta_{I}$ at $T_{\text{exp,in}} \leq 180^\circ \text{C}$ and $T_{\text{exp,in}} \geq 190^\circ \text{C}$, respectively. However, the maximal values of $\eta_{II}$ and $\zeta$ were appeared at R245fa/R134a (0.17/0.83) and R245fa/R1234yf (0.97/0.03) of the mixed fluids at $T_{\text{exp,in}} \leq 180^\circ \text{C}$ and $T_{\text{exp,in}} \geq 190^\circ \text{C}$, respectively.

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