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Article

Thermal Analysis of the Transcritical Organic Rankine Cycle Using R1234ze(E)/R134a Mixtures as Working Fluids

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Abstract: A R1234ze(E) based mixture was investigated as a promising environmental solution to enhance system performance of a transctitical organic Rankine cycle(TORC). The main purpose of this study is to research the thermodynamic properties of TORC system using R1234ze(E)/R134a mixtures with various mass fraction of R1234ze(E) when recovering engine exhaust heat. R1234ze(E) was selected due to its zero ozone depletion potential, relative lower global warming potential and it can remedy the thermodynamic properties of traditional working fluid R134a. Thermal analysis and optimization about expander inlet temperature and pressure of TORC, mass fraction of R134a in R134a/R1234ze(E) mixtures are carried out. According to the results, the designed parameters have great effect on the performance of TORC system, there is an optimal pressure to maximize the system performance, and the optimal pressure increases as temperature increases. However, when the expander inlet pressure is relatively lower, the TORC system working with pure R1234ze(E) has a best energy and exergy efficiency than working with other mixtures. When the expander inlet pressure is relatively higher, the TORC system working with pure R134a has a maximum energy and exergy efficiency than working with other mixtures. That is to say, the working conditions should be optimized and regulated in actual cycling system so as to recover more engine exhaust heat and obtain maximum expander output power, a best energy and exergy efficiency.

Keywords: Transcritical; organic Rankine cycle; R1234ze(E)/R134a mixture; engine exhaust

Nomenclature:

p	Pressure, MPa
T	Temperature, °C
s	Entropy, $kJ\cdot kg^{-1}\cdot K^{-1}$
h	Enthalpy, $kJ\cdot kg^{-1}$
Q	Heat load, kW
W	Work, kW
I	Exergy destruction rate, kW

E	Total exergy, kJ·kg^{-1}·K^{-1}
$\dot{m}$	Mass flow rate, kg·s^{-1}
η	Efficiency

Subscripts:

con	Condensing
is	Isoenthalpy
evap	Evaporator
1,2,3,4	Cycling state
TORC,en	Thermal efficiency of TORC
TORC,ex	Exergy efficiency of TORC
exh,in	Inlet of engine exhaust
exh,out	Outlet of engine exhaust
water,in	Inlet of water
water,out	Outlet of water
in	Inlet (stage)
out	Outlet (stage)
0	Ambient condition
pump	Pump
exp	Expander

Greek symbols:

Δ	Difference
x	Mass fraction of R1234ze(E)
η	Efficiency
π	Pressure ratio

Abbreviations:

TORC	Transcritical Organic Rankine Cycle
ORC	Organic Rankine Cycle
HFC	Hydrofluorocarbon
GWP	Global Warming Potential

1 Introduction

In recent years, energy crisis and environmental issues are becoming more and more urgent for the shortage of fossil fuels and pollutions, energy saving and energy recovery is being increasingly important. Engines are indispensable to vehicles, however, the fuel utilization efficiency is relative low in the engine [1], only about 30–45% of fuel energy is converted into useful output power whereas the remaining fuel energy is being wasted mainly through engine exhaust, and the engine exhaust gas temperature is usually relative high (about 250–650°C) [2]. Thus, Waste heat recovery is a preferred way to save energy, and it has been a hot topic owing much attention.

Among all the heat recovery methods, organic Rankine cycle (ORC) exhibits special advantages such as compact type, simple system, high efficiency, easy operation and low costs [3,4]. It has been widely used in recovering low-medium temperature heat (about 80–300°C) [5]. While recovering engine exhaust heat, a dual-loop Rankine cycle was prompted. The engine exhaust heat with higher temperature was recovered in the high-temperature loop, after flowing through the high-temperature loop, the engine exhaust with a relative low temperature was recovered again in the low-temperature loop. Always, the low-temperature loop usually adopted an ORC with suitable working fluid [6,7]. The engine gas temperature was relatively low in the low-temperature loop.

In this study, the engine exhaust gas with temperature being 300°C was considered to be recovered employing an ORC. Working fluids is the most important part in ORC. Nowadays, global warming is becoming more and more serious, hydrofluorocarbon (HFC) refrigerants are bound to be phased out owing to their comparatively high Global Warming Potential (GWP) [8]. According to the European Parliament and the Council of the European Union, 2014 [9]. R134a in refrigeration or ORC system will be dropped gradually. Natural refrigerants such as CO_2, NH_3, H_2O, hydrocarbons and unsaturated HFC refrigerants (also known as hydrofluoroolefin, HFOs refrigerants) are possible alternatives. For the existing of carbon-carbon double bond in HFOs molecular, they have relative low GWP and short atmospheric lifetime [10]. Meanwhile, this double bond also brings high flammability. Nearly all of the HFOs are flammable under various degree. Thus, it is much more favorable for HFOs to be employed as working fluid in mixtures than single medium, so as to prevent from flammability issues [11].

For the ORC systems. R134a has been widely used in recovering engine exhaust heat. Kim et al. [12] presented a single-loop ORC with R134a for waste heat recovery from both exhaust gas (high temperature heat source) and engine coolant (low temperature heat source). There exists other different ORC cycle recovering engine exhaust heat with R134a being working fluid [13,14].

Up to now, many efforts has been done to replace R134a used in ORC. Baik et al. [15] proposed R125 as a replacement for R134a in low-temperature geothermal power generation and the exergy efficiency in every components of the cycle was analyzed. Shu et al. [16,17] presented a similar approach for a dual loop ORC using R1234yf and R134a. Boyaghchi et al. [18] used four working fluids including R1234ze(E) and R134a to promote a thermoeconomical approach in a combined heat and power cycle. Liu et al. [19] has investigated eight HFOs working fluids for geothermal power generation and the system efficiency comparison to conventional fluids (R134a et al) was analyzed. Invernizzi et al. [20] investigated the potential replacement of R134a in ORC applications by two low-GWP refrigerant fluids, namely R1234yf and R1234ze(E), considering the direct replacement of the original fluid by the two refrigerants or a completely new design simulation results show a decrease of the net power. Scholars also chose mixtures to remedy the disadvantages of R134a. Abadi et al. [21] investigated the performance using a new zeotropic mixture of R245fa 60%/R134a 40% (molar concentration) in small scale ORC (organic Rankine cycle), the results showed that the proposed zeotropic mixture increases the power output compared to an identical ORC with pure fluid despite working at a lower pressure ratio. Liu et al. [22,23] performed an experimental investigation using CO_2/R134a mixtures to recover waste heat of engine coolant and exhaust gas, the results showed that CO_2/R134a (0.7/0.3) achieved the maximum net power output at a high expansion inlet pressure, while CO_2/R134a (0.6/0.4) behaves better at low pressure.

All of these studies have in common that no direct R1234ze(E)/R134a mixtures were used in recovering engine exhaust heat, but thermal analysis of R1234ze(E) or R134a has been done and shown good performance. Molés et al. [24] has compared the predicted organic Rankine cycle performance of R1234yf and R1234ze, alternatives to R134a over a wide range of evaporating and condensing temperatures for a given thermal power input. The results showed that R1234ze would require 15.7% to 20.2% lower pump power and would enable up to 13.8% higher net cycle efficiencies than R134a. Yang et al. [25] investigated the thermodynamic and economic performances optimization for an ORC system recovering the waste heat of exhaust gas from a large marine diesel engine of the merchant ship using R1234ze, R245fa, R600 and R600a, and the results showed that R245fa performed the most satisfactorily in terms of the optimal economic performance, while R1234ze had the largest thermal efficiency as well as by proposing an ORC system reducing 76% CO_2 emission per kWh.

Based on the review of recent researches, R1234ze possesses zero ozone depletion potential, relative lower global warming potential and it has shown outstanding thermal properties in different ORC systems. Meanwhile, for the reason that R1234ze(E)/R134a mixtures is rarely used in transcritical organic Rankine cycle (TORC) for recovering part of engine exhaust heat with lower temperature. In the TORC, it is essential to analyze the impact of different parameters such as the mass fraction of R1234ze(E), engine exhaust outlet temperature and expander inlet temperature and pressure on the total thermal efficiency of the TORC. Therefore, a thermodynamic model of TORC using R1234ze(E)/R134a mixtures for engine waste heat recovery was established, the thermal and exergy efficiencies of TORC considering different parameters were analyzed.

2 System Description

Fig. 1 shows the schematic diagram of the system in this work for part of engine exhaust heat recovery with lower temperature. The TORC system contains a working fluid pump, a vapor generator, an expander and a condenser.

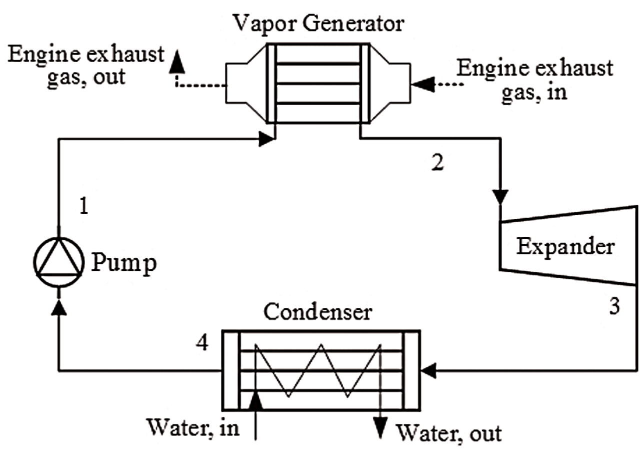

Figure 1: Schematic diagram of the TORC system

R1234ze(E) and R134a are chosen as the working fluids for this work, their thermophysical properties are obtained from Refprop 9.0, and the relative parameters are listed in Tab. 1.

R1234ze(E)/R134a mixtures are employed as the working fluid in TORC for engine exhaust heat recovery to break through the limitations of the thermophysical properties of each single working fluid. Here, the mass fraction of the R1234ze(E) in R1234ze(E)/R134a mixtures are defined as:

$$x = \frac{\dot{m}_{R1234ze(E)}}{\dot{m}_{R1234ze(E)} + \dot{m}_{R134a}} \tag{1}$$

X = 1 stands for the pure R1234ze(E) is cycling in the system, x = 0 stands for the pure R134a is working in the TORC cycle. The working performance of the mixtures with various mass fractions of R1234ze(E) were mainly studied in this study.

Fig. 2 illustrates the temperature-entropy (T-s) diagram of TORC with R1234ze(E)/R134a at the mass fraction of 0.5/0.5.

As is shown in Fig. 2, in the TORC system, the mixed working fluid is superheated by the engine exhaust heat to a supercritical state (point 2). Then, the mixture is expanded in the expander to a lower

Table 1: Thermophysical properties of R1234ze(E) and R134a

Thermal Properties	R1234ze(E)	R134a
Chemical formula	CHF = $CHCF_3$	CH2F-CF3
Critical temperature	109.37°C	101.06°C
Critical pressure	3.636 MPa	4.059 MPa
Normal boiling temperature	–18.97°C	–26.07°C
Molecular mass	114.04	102.03
Safety classification	A2L	A1
ODP(Ozone Depletion Potential)	0	0
100-year GWP(Global Warming Potential)	4	1300

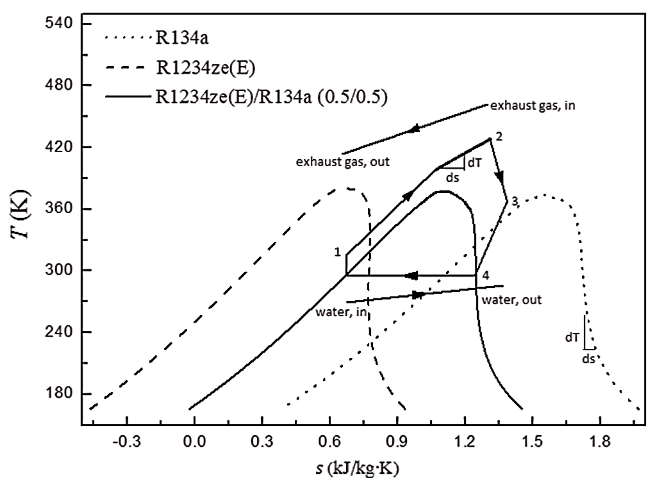

Figure 2: The temperature-entropy (T-s) diagram of TORC with R1234ze(E)/R134a

pressure (point 3), and condensed in a condenser to a liquid state (point 4), then it is pumped to a liquid state with higher pressure (point 1). End here a complete cycle is finished.

In this work, an engine with rated power being 96 kW of a refrigerated truck is analyzed. Considering that the engine has different operating conditions, the representative parameters of the engine and the relative thermodynamic cycle parameters are listed in Tab. 2. [26,27]. Also, the boundary conditions of current TORC model are contained as follows.

3 Thermodynamic Analysis

The following assumptions are made to simplify the analysis of TORC recovering the engine exhaust heat.

(1) The engine exhaust gas consists of 14.83% O_2, 4.36% CO_2, 6.2% H_2O, 74.61% N_2 (mass fraction).

(2) The heat capacity of engine exhaust gas and water is constant.

(3) The isentropic efficiency is set constant for the pump and expander.

(4) The heat and pressure loss in the pipes connecting the components are negligible.

(5) All the kinetic and potential effects are ignored.

(6) The thermal energy from engine exhaust gas is absorbed by the mixture working fluid completely.

Table 2: Exhaust energy parameters and the boundary conditions of TORC

Parameters	Values
Engine exhaust inlet temperature ($T_{exh,in}$)	300°C
Expander inlet temperature (T_2)	200–260°C
Evaporator pressure (P_2)	5–31 MPa
Mass flow rate of engine exhaust heat ($\dot{m}_{exh}$)	0.22 $kg{\cdot}s^{-1}$
Inlet temperature of water ($T_{water,in}$)	30°C
Condensing temperature of TORC (T_{con})	50°C
Pinch temperature for gas-liquid (ΔT_1)	30°C
Pinch temperature for liquid-liquid (ΔT_2)	10°C
Ambient pressure (P_0)	0.1 MPa
Ambient temperature (T_0)	20°C
Pump efficiency (η_{pump})	0.8
Expander efficiency (η_{exp})	0.8

3.1 Energy Analysis

For the evaporator,

The evaporator in TORC system us employed to heat the working fluids, heat from the engine exhaust gas are transferred to the working fluids in TORC system. The engine exhaust heat can be calculated as follows:

$$Q_{exh} = \dot{m}_{exh}\left(h_{exh,in} - h_{exh,out}\right) = \dot{m}_{TORC}(h_2 - h_1) \tag{2}$$

where $\dot{m}_{exh}$ and h_{exh} are the mass flow rate and specific enthalpy of the engine exhaust heat, respectively. The subscript *in* and *out* refers to inlet and outlet. h_1 is the specific enthalpy of the outlet of the pump and h_2 is the specific enthalpy of inlet of the expander. $\dot{m}_{TORC}$ is the inlet mass flow rate of the working fluid in the TORC system.

For the pump,

The pump in TORC system is used to increase the pressure of working fluid, the working fluid in this component is operating in liquid phase. The outlet pressure of the pump can be calculated by

$$P_1 = \pi_{pump} P_4 \tag{3}$$

where P_1, P_4 and π_{pump} are outlet pressure, inlet pressure and pressure ratio of the pump, respectively. The power consumed by pump can be achieved as follows:

$$W_{pump} = \dot{m}_{TORC}(h_1 - h_4) \tag{4}$$

where h_4 is the specific enthalpy of the inlet of the pump, respectively. The isentropic efficiency of the pump η_{pump} is defined as:

$$\eta_{pump} = \left(h_{1,is} - h_4\right)/(h_1 - h_4) \tag{5}$$

where $h_{1,is}$ is the isentropic specific enthalpy of the pump outlet.

For the expander,

Expander is employed to extract energy from working fluid and output electricity when combined with a generator. The outlet pressure of the expander P_3 can be described as follows:

$$P_3 = P_2/\pi_{\text{exp}} \tag{6}$$

where P_2 is inlet pressure of the expander, π_{exp} is pressure ratio of the expander. The power generated in the turbine $\dot{W}_{\text{exp}}$ can be expressed as follow,

$$W_{\text{exp}} = \dot{m}_{\text{TORC}}(h_2 - h_3) \tag{7}$$

where h_3 is the specific enthalpy of the outlet of the expander. The isentropic efficiency of the expander is calculated by

$$\eta_{\text{exp}} = (h_3 - h_2)/(h_{3,\text{is}} - h_2) \tag{8}$$

where $h_{3,\text{is}}$ is the isentropic specific enthalpy of the expander outlet.

For condenser,

Condenser in this system is used to cool the working fluid in TORC to an appropriate temperature, the energy balance equation is:

$$Q_{\text{con}} = \dot{m}_{\text{water}}(h_{\text{water,in}} - h_{\text{water,out}}) = \dot{m}_{\text{TORC}}(h_3 - h_4) \tag{9}$$

where $\dot{m}_{\text{water}}$ and h_{water} are the mass flow rate and the specific enthalpy of cooling water, respectively.

The thermal efficiency of the cycle can be calculated as:

$$\eta_{\text{TORC,en}} = \frac{W_{\text{exp}} - W_{\text{pump}}}{Q_{\text{exh}}} \tag{10}$$

3.2 Exergy Analysis

The exergy analysis was based on the second law of thermodynamics, and the exergy efficiency was finally given here. In order to calculate the exergy efficiency of the TORC system, the exergy loss rate in each component in TORC must be analyzed. The mass flow rate of TORC was needed to complete exergy analysis.

Here, the atmospheric state was assumed to be steady. The exergy destruction rate of each component can be obtained through the following equations:

For the expander,

The exergy destruction of the expander is:

$$I_{\text{exp}} = \dot{m}_{\text{TORC}} T_0 (s_3 - s_2) \tag{11}$$

where T_0 refers to the atmosphere temperature, s_3 and s_2 is the specific entropy of outlet and inlet of the expander.

For the condenser,

The exergy destruction of the condenser is:

$$I_{\text{con}} = T_0\left[\dot{m}_{\text{water}}(s_{\text{water,out}} - s_{\text{water,in}}) - \dot{m}_{\text{TORC}}(s_3 - s_4)\right] \tag{12}$$

where s_{water} is the specific entropy of cooling water. The subscript *in* and *out* refers to inlet and outlet. s_4 is the specific entropy of the inlet of the pump.

For the pump,

The exergy destruction of the pump is:

$$I_{pump} = \dot{m}_{TORC} T_0 (s_1 - s_4) \tag{13}$$

where s_1 is the specific entropy of the outlet of the pump.

For the evaporator,

The exergy destruction of the evaporator is:

$$I_{evap} = T_0 \left[\dot{m}_{TORC} (s_2 - s_1) - \dot{m}_{exh} \left(s_{exh,in} - s_{exh,out} \right) \right] \tag{14}$$

where $\dot{m}_{exh}$ and s_{exh} are the mass flow rate and specific entropy of the engine exhaust heat, respectively.

The input and output of TORC system in the evaporation and condensation exergy were given as follows:

Input exergy to the evaporator:

$$E_{evap} = \dot{m}_{exh} \left[\left(h_{exh,in} - h_{exh,out} \right) - T_0 \left(s_{exh,in} - s_{exh,out} \right) \right] \tag{15}$$

Output exergy from the condenser:

$$E_{con} = \dot{m}_{water} \left[\left(h_{water,out} - h_{water,in} \right) - T_0 \left(s_{water,out} - s_{water,in} \right) \right] \tag{16}$$

The exergy balance equation of TORC system was given as follows, which contains exergy destruction in each component in the system.

$$E_{evap} - E_{cond} = W_{exp} - W_{pump} + \sum I \tag{17}$$

Therefore, the exergy efficiency of TORC system can be obtained:

$$\eta_{TORC,ex} = 1 - \frac{\sum I}{E_{evap} - E_{cond}} = \frac{W_{exp} - W_{pump}}{E_{evap} - E_{cond}} \tag{18}$$

4 Model Validation

According to the mathematical model in 3.1 and 3.2, the energy and exergy analysis of the TORC system recovering engine exhaust heat can be obtained. In this study, the simulation is based on Matlab program by linking to the NIST database (REFPROP 9.0), which is developed by National Institute of Standards and Technology [28].

In order to ensure accuracy of simulation code, the HT loop ORC model in literature [27] is selected to verify the correctness of calculation. The same working fluid and boundary conditions are set, and the results comparison are shown in Tab. 3. As shown, the present simulation result has a good agreement with literature [27]. In this way, the Matlab code has enough accuracy for the current study.

5 Results and Discussions

For the evaporating process in the TORC heating by the engine exhaust heat, if the specific heat is considered as constant, the heating line 1–2 shown in Fig. 2 will be straight, at this time, the pinch point must be in the inlet or outlet of evaporator. However, when R134a or R1234ze is working in the TORC

Table 3: Results comparison between present study and literature [27] using R600a/R601a with mass fraction of 0.5/0.5 as working fluid

Working fluid	$T_{g,i2}$ (°C)	m_{ORC} (kg/s)	$W_{t,HT}$ (kW)	$\eta_{ORC,en}$	$\eta_{ORC,ex}$
R600a/R601a [27]	123.71	0.85	31.32	8.35	33.81
R600a/R601a	123.70	0.85	31.34	8.32	33.80
Relative deviation (%)	0.01	0.00	−0.06	0.36	0.03

system, their thermophysical properties will vary highly with temperature changing from liquid state to supercritical state. The specific heat of R134a or R1234ze cannot be considered as constant. Therefore, the location of pinch point of evaporator in TORC should be carefully calculated, which is essential for the result of mass flow of working fluid. In this study, the exhaust outlet temperature and mass flow rate of TORC are optimized meeting the requirement of pinch temperature for gas liquid. Meanwhile, the total heat from engine exhaust, the total power from expander, the energy efficiency and the exergy efficiency of the TORC system can be obtained.

5.1 Engine Exhaust Outlet Temperature and Engine Exhaust Heat

Fig. 3 shows the variation of the outlet temperature of engine exhaust from evaporator under different expander inlet temperature and pressure. As is given in the figure, the expander inlet temperature varies from 200°C to 260°C, some engine exhaust temperature varies as straight line. When the expander has 200°C inlet temperature and 4 MPa inlet pressure, the pinch point almost located in the inlet of evaporator (point 1 shown in Fig. 2). The engine exhaust outlet temperature values vary along with the changes of expander inlet temperature and expander inlet pressure, indicating that the pinch point has shifted from the inlet of evaporator. The pinch point shifting demonstrate that the thermodynamic properties will change in the supercritical state, and the pinch point iteration calculation can successfully find out the pinch point location under different expander inlet temperature and pressure.

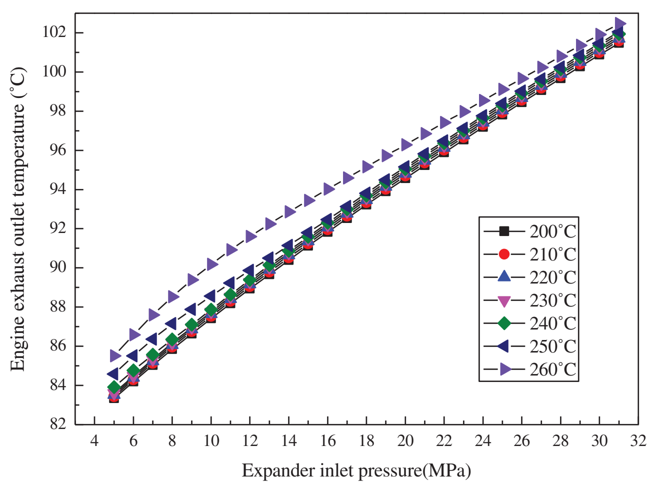

Figure 3: The engine exhaust outlet temperature along with different expander inlet pressure under various expander inlet temperature (200–260°C), mass fraction of R134a/R1234ze(E) is 0.5/0.5

Fig. 4 shows total heat that can be recovered from the engine exhaust. As can be seen, a biggest amount of engine exhaust heat can be recovered under 200°C expander inlet temperature and 4 MPa expander inlet

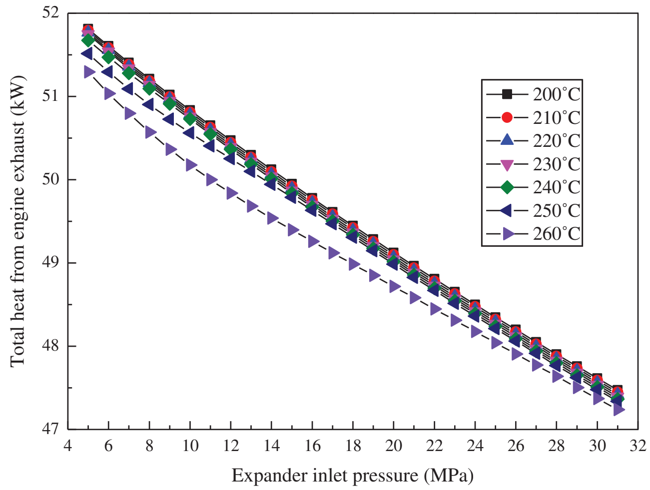

Figure 4: The total heat from engine exhaust along with different expander inlet pressure under various expander inlet temperature (200–260°C), mass fraction of R134a/R1234ze(E) is 0.5/0.5

pressure under a fixed pinch point. The lower the expander inlet temperature and the lower the expander inlet pressure, the higher heat can be recovered from the engine exhaust. When the expander inlet pressure is 5 MPa and expander inlet temperature is 200°C, a maximum heat 51.8118 kW can be obtained from the engine exhaust.

5.2 Mass Flow Rate of R134a/R1234ze(E) (0.5/0.5)

The variations of mass flow rates of R134a/R1234ze(E) (0.5/0.5) mixtures, of which the mass fraction is 0.5/0.5, working in TORC with different expander inlet temperature and pressure is displayed in the Fig. 5. According to Eq. (2), the mass flow rate of R134a/R1234ze(E) (0.5/0.5) mixtures can be calculated according to the recovered heat from engine exhaust and enthalpy difference between point 1and point 2 of R134a/R1234ze(E) (0.5/0.5) mixtures shown in Fig. 1. While the expander inlet pressure is a fixed value, the temperature of point 1 of working fluid shown in Fig. 1 is fixed, along with the increasing of expander inlet temperature, the enthalpy difference between points 1 and 2 of working fluid increases. Then, the

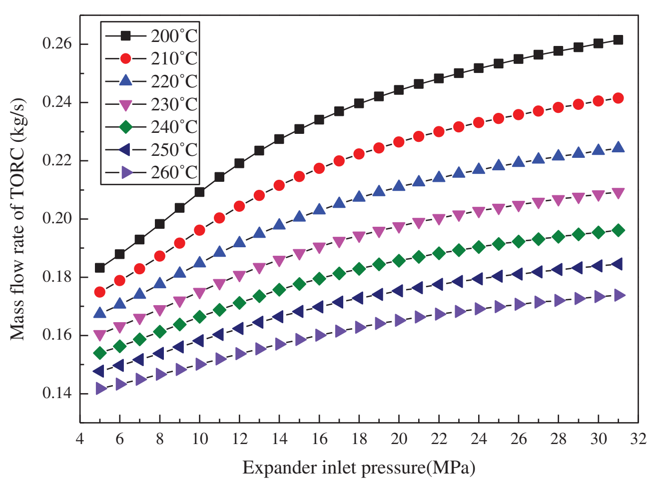

Figure 5: The mass flow rate of TORC along with different expander inlet pressure under various expander inlet temperature (200–260°C), mass fraction of R134a/R1234ze(E) is 0.5/0.5

mass flow rate of TORC will decrease as can be seen in Fig. 5. Also, it can be obtained from Fig. 5 that the mass flow rate of R134a/R1234ze(E) (0.5/0.5) mixtures increases along with expander inlet pressure.

When the expander inlet pressure increases, the pump will require more work, thus, the pump outlet temperature of R134a/R1234ze(E) (0.5/0.5) mixtures will be higher and the enthalpy difference between point 1 and point 2 will be smaller. Simultaneously, the engine exhaust outlet temperature increases with the expander inlet pressure, resulting in a smaller enthalpy difference of the inlet and outlet of engine exhaust, then less heat will be absorbed by the working fluid in TORC. Whereas, the enthalpy difference of working fluid decreases faster than that of engine exhaust. The end result is that the mass flow rate of R134a/R1234ze(E) (0.5/0.5) mixtures increases along with expander inlet pressure.

5.3 Expander Output Power

The variations of expander output power of TORC with different expander inlet temperature and pressure is displayed in Fig. 6. According to Eq. (7), the power output of TORC can be calculated according to the mass flow rate and enthalpy difference between point 3 and point 2 of R134a/R1234ze (E) (0.5/0.5) mixtures shown in Fig. 1. The enthalpy of point 3 is fixed. At the same expander inlet temperature, while the expander inlet pressure is a fixed value, the enthalpy of point 2 of working fluid shown in Fig. 1 is fixed, along with the increasing of expander inlet temperature, the enthalpy difference between point 3 and point 2 of working fluid increases. Then, the mass flow rate of TORC will also increase as can be seen in Fig. 5. In this way, the expander output power will increase along with the increasing expander inlet pressure under a fixed expander inlet temperature.

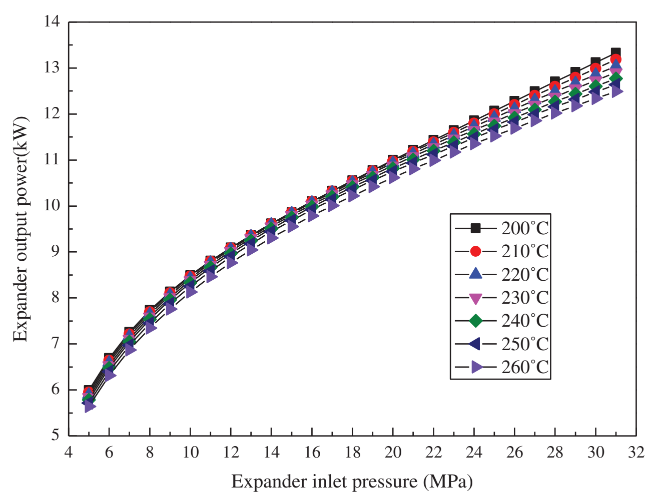

Figure 6: The expander output power along with different expander inlet pressure under various expander inlet temperature (200–260°C), mass fraction of R134a/R1234ze(E) is 0.5/0.5

When the expander inlet pressure is fixed, the increasing expander inlet temperature will cause an increasing enthalpy of point 2. The enthalpy difference between point 3 and point 2 will increase. On the other hand, the higher the expander inlet temperature is, the TORC will recover smaller total engine exhaust heat as is shown in Fig. 4, resulting in that the mass flow rate dropped sharply. However, the enthalpy difference of working fluid increases slower than that of mass flow rate. At last, the expander output power decreases along with the increasing expander inlet temperature under different fixed expander inlet pressure. When the expander inlet pressure is 31 MPa and expander inlet temperature is 200°C, a maximum expander output power 13.3328 kW can be obtained from the TORC system.

5.4 Efficiency Optimization of TORC

Figs. 7 and 8 indicate the variation of energy, exergy efficiency of TORC. As can be seen, for a given turbine inlet temperature, with the increase of turbine inlet pressure, the energy and exergy efficiency firstly increases and then decreases. Namely, there is an optimal turbine inlet pressure to maximize the energy and exergy efficiency of TORC. Meanwhile, the diagram also shows that the higher the expander inlet temperature is, the bigger the optimal expander inlet pressure is. Besides, it can be observed that when the turbine inlet pressure is relatively lower such as 5 and 6 MPa, the higher the turbine inlet temperature is, the less the energy and exergy efficiency are. However, when the turbine inlet pressure is relatively higher such as 14 and 15 MPa, the energy and exergy efficiency increase along with expander inlet temperature. That is because the lower the expander inlet pressure is, the smaller pressure ratio of expander is, that leads a bad match with relatively higher expander inlet temperature and results in a less power output shown in Fig. 6. Contrarily, a high expander inlet pressure has a good match with a high expander inlet temperature, which results in a more mechanical work produced by expander, as can be seen in Fig. 6. In general, when using R134a/R1234ze(E) (0.5/0.5) mixtures as the working fluid for TORC, there exists an optimal expander inlet temperature and expander inlet pressure during which has a higher energy efficiency and exergy efficient. The working conditions should be optimized and regulated in actual cycling system so as to recover more engine exhaust heat and obtain a best energy and exergy efficiency.

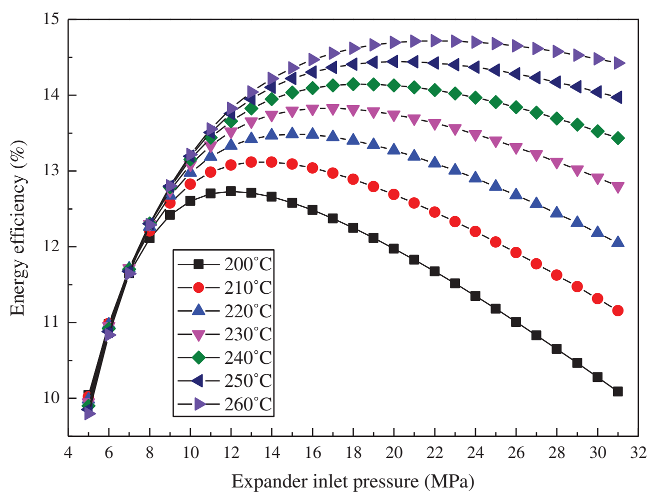

Figure 7: The energy efficiency along with different expander inlet pressure under various expander inlet temperature (200–260°C), mass fraction of R134a/R1234ze(E) is 0.5/0.5

5.5 Effects of Mass Fraction of R1234ze(E) in R134a/R1234ze(E) Mixtures on Performance of TORC

According to the analyzing result stated previously of using R134a/R1234ze(E) (0.5/0.5) in TORC. The thermophysical properties of TROC working with R134a/R1234ze(E) mixture under 260°C expander inlet temperature and different expander inlet pressure are studied, during which the mass fraction of R1234ze(E) various from 0 to 1. Figs. 9–11 present the total output of expander, energy efficiency and exergy efficiency of the system influenced by the mass fraction, respectively. The expander output power will increase along with the increasing expander inlet pressure under different mass fractions. It can be observed that the TORC working with pure R1234ze(E) will obtain a maximum expander output power which is 12.49 kW.

Fig. 10 presents the energy efficiency of TORC system with R134a/R1234ze(E) mixtures under different expander inlet pressure. The energy efficient will increase firstly than decrease along with the increasing expander inlet pressure that means there exists an optimized expander inlet pressure to obtain a maximum energy efficiency. When the expander inlet pressure is relatively lower as 5–10 MPa, R1234ze(E)

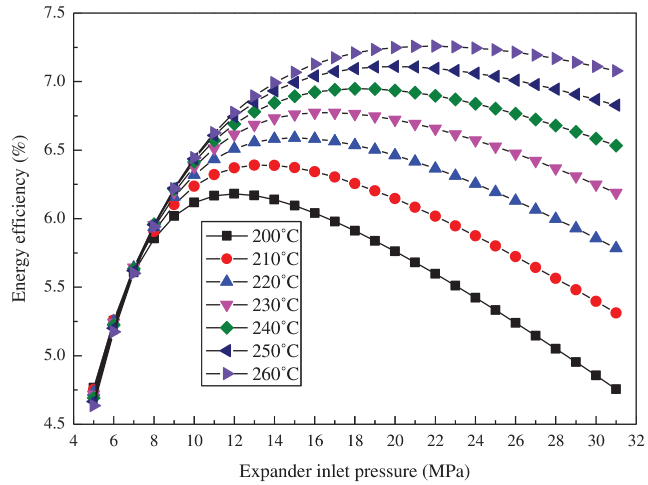

Figure 8: The exergy efficiency along with different expander inlet pressure under various expander inlet temperature (200–260°C), mass fraction of R134a/R1234ze(E) is 0.5/0.5

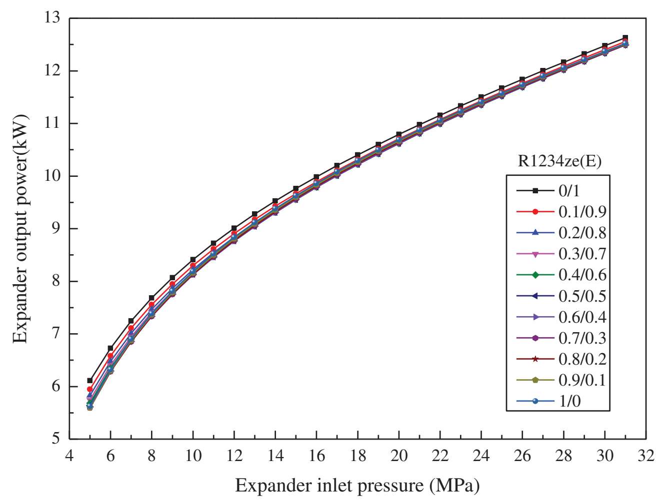

Figure 9: The expander output power along with different expander inlet pressure under mass fraction of R1234ze(E) in R134a/R1234ze(E) mixtures (x = 0–1)

presents a best performance than other R134a/R1234ze(E) mixtures with different mass fraction. When the expander inlet pressure is relatively higher as 11–31 MPa, R134a presents a best performance that other mixtures. This means that R134a and R1234ze(E) both own optimum working condition. In practical usage, the optimizing work should be done before designing the TORC system recovering engine exhaust heat, so as to exquisite highest power output, energy efficiency or exergy efficiency. The optimal working condition with highest energy efficiency are as follows: 20 MPa expander inlet pressure, 260°C expander inlet temperature, exergy efficiency is 14.7922%, the working fluid is pure R134a.

As is shown in Fig. 11, each mixture has its own corresponding optimum expander inlet pressure, under which the exergy efficiency is highest. The optimal working condition with highest exergy efficiency are as follows: 20 MPa expander inlet pressure, 260°C expander inlet temperature, exergy efficiency is 7.3052%, the working fluid is pure R134a.

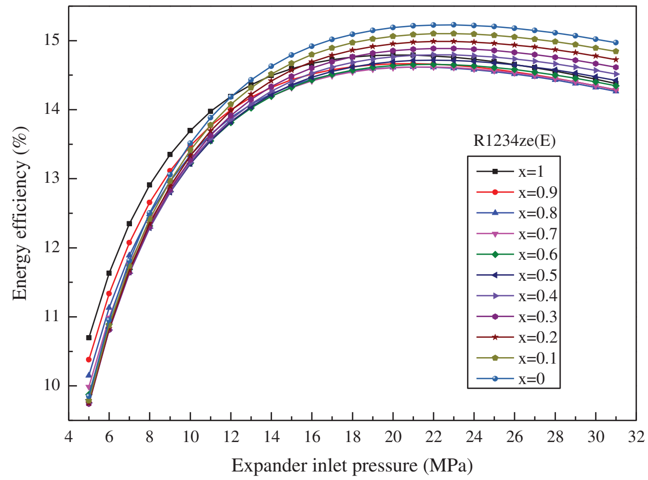

Figure 10: The energy efficiency along with different expander inlet pressure under mass fraction of R1234ze(E) in R134a/R1234ze(E) mixtures (x = 0–1)

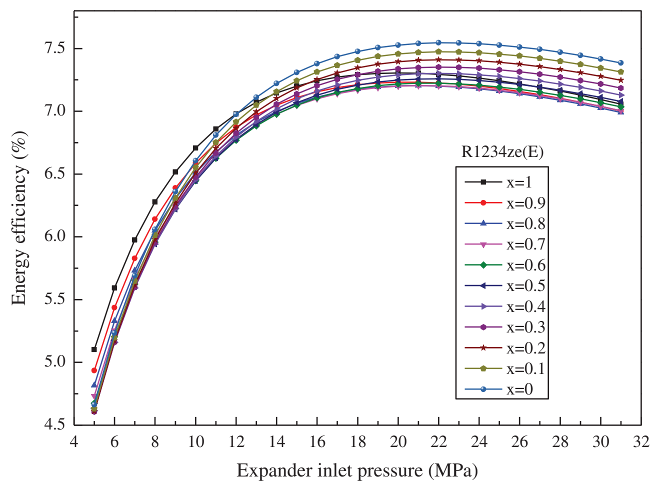

Figure 11: The exergy efficiency along with different expander inlet pressure under mass fraction of R1234ze(E) in R134a/R1234ze(E) mixtures (x = 0–1)

6 Conclusions

In this study, the thermodynamic properties of the TORC system using R1234ze(E)/R134a mixtures to recover the engine exhaust heat was investigated. The expander inlet pressure was optimized, the influences of expander inlet temperature and R1234ze(E) mass fraction on the performance of TORC system were investigated in this work, some conclusions can be obtained as follows:

1. When using R1234ze(E)/R134a (0.5/0.5) mixture as working fluid, the expander output power W_{exp}, thermal efficiency $\eta_{TORC,en}$ and exergy efficiency $\eta_{TORC,ex}$ of the TORC will various along with expander inlet pressure, also, under different expander inlet temperature, there exist an optimized expander inlet pressure under which the system performed best.
2. For a given expander inlet pressure, system performance firstly increases and then decrease with expander inlet pressure. That is to say, there is an optimal pressure to maximize the system performance, and the optimal pressure increases as temperature increases.

3. The TORC system working with pure R1234ze(E)can always obtain a maximum expander outlet power. However, when the expander inlet pressure is relatively lower, the TORC system working with pure R1234ze(E) has a best energy and exergy efficiency than working with other mixtures. When the expander inlet pressure is relatively higher, the TORC system working with pure R134a has a maximum energy and exergy efficiency than working with other mixtures.

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