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Carbon Dioxide as Working Fluids in Transcritical Rankine Cycle for Diesel Engine Multiple Waste Heat Recovery in Comparison to Hydrocarbons

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Abstract: In consideration of the high-temperature characteristic of engine's waste heat and stricter environmental regulations, natural substance, including CO_2 and hydrocarbons, have been treated as promising working fluid for diesel engine waste heat recovery due to its environment friendly and excellent physical and chemical properties. This paper presented a comprehensive performance analysis on transcritical Rankine cycles for diesel engine multiple waste heat recovery using hydrocarbons and CO_2 as working fluid. The optimal turbine inlet pressures corresponding to maximum net power output, maximum exergy efficiency and minimum electricity production cost (*EPC*) were obtained. The effect of working fluid on these optimal pressures has been discussed. For fluids with low critical temperature, the optimal pressure corresponding to maximum net power output is lower than the one for maximum exergy efficiency, while the opposite results can be found for fluid with high critical temperature. Then, the effect of various working fluid properties in transcritical cycle performance is discussed. Comparison results show that CO_2 obtains only more power output than Ethane, Propane and Propene, but CO_2 is capable of absorbing more energy from engine coolant and regeneration heat with comparable total heat transfer areas and has an advantage in turbine size, particularly for hydrocarbons with high critical temperature.

Keywords: transcritical Rankine cycle, natural working fluid, multiple waste heat, performance comparison

1. Introduction

Currently, one of the main contributors on CO_2 emissions is the road transportation. Ministry of Transport of the People's Republic of China reports that freight transportation by heavy-duty truck consumed 49.2% of the total fuel consumption of transportation. The situation of USA is similar to China [1]. The concern about environment problem and the rising price of crude oil have stimulated the technology development for energy conservation and emission reduction. It is well known that more than half of fuel combustion energy is wasted via exhaust gas and engine coolant for typical diesel engine [2]. It is promising to improve diesel engine energy utilization efficiency by waste heat recovery. In existing waste heat recovery technologies, the Rankine cycle technologies are relatively mature, which also are capable of multiple waste heat source recovery from engine [3]. Recent studies showed that waste heat recovery (WHR) prototypes for on-road vehicle have been in testing [4,5].

A large amount of researches have been conducted about Rankine cycle for engine waste heat recovery using refrigerants as working fluid [6]. However, most

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Received: Oct 24, 2018 Corresponding author: TIAN Hua E-mail: thtju@tju.edu.cn refrigerant cannot withstand high temperature exhaust gas from engine due to their relatively low decomposition temperature. In addition, refrigerants, which commonly contain fluorine or chlorine, have significant effect on environment [7]. Previous studies indicated that man-made refrigerants have less potential as promising working fluid for engine waste heat recovery [8]. Hence, it is necessary to explore high temperature environmental-friendly working fluid.

Hydrocarbons as natural substance could be a promising replacement of refrigerants for high temperature waste heat application. Ngoc et al. [9] compared alkanes, aromatic alkanes and linear siloxanes in ORC system with maximum temperature of 250°C and 300°C and revealed that Cyclopentane was the best working fluid for all cased studied. The literature review by Lion et al. [4] indicated that hydrocarbons can be considered as good candidates for high temperature waste heat recovery. Similar conclusion can also be obtained in Ref. [10]. Shu et al. [11] proposed alkane based Organic Rankine cycle system for high temperature exhaust gas recovery and revealed that hydrocarbons are attractive. In automobile field, hydrocarbons or their blends are treated as an alternative to Chlorofluorocarbons (CFCs) or hydrochlorofluorocarbons (HCFCs) for vehicles air-conditioners [12], which indicates the feasibility of hydrocarbons used for engine waste heat recovery.

CO_2 is another working fluid candidate for high temperature heat recovery application due to its safe, cheap and environmental-friendly features. Previous studies showed that the CO_2 transcritical Rankine cycle (CTRC) is a promising technique for waste heat recovery from internal combustion engine [13]. Several CO_2 bottom cycles were presented for waste heat recovery from vehicle exhaust gas by Chen et al. [14] and the comparison results showed that CTRC is suitable for converting small-scale waste heat in vehicle exhaust gas into useful power output. Another research conducted by Wang et al. [13] revealed that the single CTRC has a better thermos-economic performance with exhaust gas temperature range from 300 to 600°C. Preliminary experiment conducted by Echogen Power Systems (EPS) company [15] also indicated the CTRC is a potential waste heat recovery technique in engine exhaust gas temperature range. In addition, CO_2 has been used as working fluid in vehicle air conditioning [16]. The European automotive manufacture Daimler has announced to offer CO_2 air conditioning system in production [17]. Previous researches have been focused on the comparison between CO_2 and R245fa [18], R32 [19], R125 [20,21], Ethane [21], R600a [22] and R601 [22] with single low temperature heat source. Up to now, few studies have conducted the comparison between CO_2 and hydrocarbons with multiple waste heat recovery.

In this paper, six linear hydrocarbons and two cyclic hydrocarbons in total were investigated in comparison to CO_2 for multiple waste heat recovery from diesel engine. The performance indicators considered in this research were: (a) net power output based on the first law of thermodynamics; (b) exergy efficiency based on the second law of thermodynamics; and (c) electricity production cost (*EPC*) as an indicator of the economic performance. The optimal turbine inlet pressures based on above indicators were determined. The effect of working fluid on these optimal pressures was discussed. Finally, a comprehensive comparison between hydrocarbons and CO_2 was conducted based on their thermo-physical properties.

2. System Description

Hydrocarbons are nature substances that consist entirely of hydrogen and carbon atoms. Methane with relatively low critical temperatures is out of consideration. Experiment by Dai et al. [23] showed that hydrocarbons with more than 6 carbon atom have poor thermal stability and are not suitable for supercritical ORCs. Hence, hydrocarbons with 2 to 6 carbons are chosen, including six linear hydrocarbons and two cyclic hydrocarbons. The thermodynamic characteristics of selected hydrocarbons are listed in Table 1. In general, hydrocarbons are technically viable for low-capacity system for automobile [12] because hydrocarbons are treated as an alternative for vehicles air-conditioners and millions of tons of hydrocarbons are used for powering vehicles. CO_2 is another natural substance, which is environmental-friendly, low toxicity, safe and cheap. The thermodynamic characteristic of CO_2 is also listed in Table 1. ODP and GWP mean ozone depletion potential and Global warming potential, respectively.

Table 1 Thermodynamic characteristic of hydrocarbons and CO_2

Name	T_{cri}/°C	P_{cri}/MPa	ODP	GWP [7]
CO_2	30.9	7.38	0	1
Ethane	32.2	4.87	0	5.5
Propene	91.1	4.56	0	1.8
Propane	96.7	4.25	0	3.3
Isobutane	134.7	3.63	0	3
Pentane	196.6	3.37	0	Very low
Hexane	234.7	3.03	0	Very low
Cyclopentane	238.5	4.52	0	Very low
Cyclohexane	280.5	4.08	0	Very low

The objective diesel engine is a typical 9.5 L turbocharged and intercooled diesel engine from the heavy duty truck. The rated power and maximum engine torque of the objective engine are 276.5 kW and 1900

N·m, respectively. The other parameters about this diesel engine can be found in Ref. [3]. The main heat source parameters are listed in Table 2. Exhaust gas and engine coolant are utilized as heat source because they possess almost half of engine combustion energy.

Table 2 Main parameters of the heavy-duty diesel engine

Parameters	Value
Exhaust temperature/°C	519
Exhaust mass flow rate/kg·h^{-1}	1444.1
Engine coolant return temperature/°C	77.9
Engine coolant outlet temperature/°C	88.4
Engine coolant flow rate/kg·min^{-1}	211.3

The structure of the preheating-regenerative Rankine cycle system is shown in Fig. 1. The engine coolant energy is employed to preheat the working fluid. The cycle layout is a regenerative ORC, which has been proved that the addition of regenerator has positive effect on system thermodynamic performance [24]. Finally, transcritical Rankine cycle (Fig. 1(b)) is adopted to avoid the high irreversible losses [25] and matches better with multiple waste heat source [8].

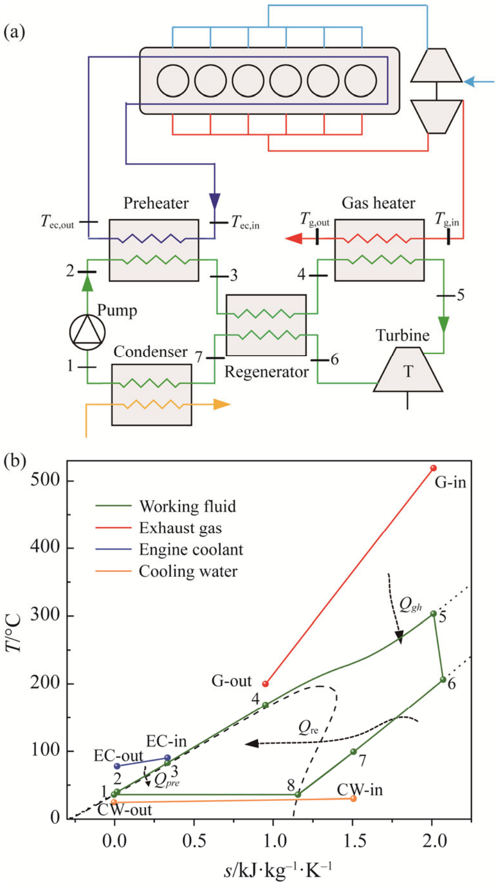

Fig. 1 Preheating-regenerative Rankine Cycle system. (a) Schematic diagram; (b) *T-s* diagram.

3. System Modeling

The main simulation boundary conditions adopted in this study are listed in Table 3. The turbine efficiency and pump efficiency are set as 0.7 and 0.6, respectively. The mass flow rate of working fluid is calculated by the pinch point temperature difference (PPTD) method, which is described in our previous paper [3]. Dai et al. [23] adopted a rapid experimental method to measure the decomposition temperature of several hydrocarbons. The experimental results showed that the decomposition temperature of selected hydrocarbons is about 260-320°C. The auto ignition temperature summarized by Glover et al. [8] presented that most hydrocarbons are safe when the temperature is below 300°C. Hence, the maximum cycle temperature is set as 300°C.

Table 3 Summary of calculation parameter

Parameter	Value
Exhaust gas acid dew point	120°C
Maximum cycle temperature	300°C
PPTD of preheater/gas heater/regenerator/ condenser	5/30/15/5°C
Condensing temperature (P_l>5 kPa)	40/25°C
Inlet temperature of cooling water	30/15°C
Turbine efficiency	0.7
Pump efficiency	0.6
Reduced high pressure $P_{h,r}=P_h/P_{cir}$	1.1,…, 2.5

For most hydrocarbons, condensing temperature of 40°C is set to ensure condensing pressures are above 5 kPa [26]. There are two exceptions: condensing temperature of 25°C is adopted for CO_2 and Ethane in order to keep the condensing temperature lowering than critical temperature.

A basic thermodynamic model is built to obtain the system thermodynamic performance (see our previous paper [24]). The heat utilization rate (*UR*) is an important index to evaluate the multiple waste heat recovery capability of system. In this paper, UR_g and UR_{ec} means the waste heat utilization rates of exhaust gas and engine coolant. The maximum utilization is achieved when the exhaust gas and engine coolant are cooled to acid dew point temperature and engine coolant return temperature, respectively.

$$UR_g=\frac{m_g\cdot C_{p,g}\cdot\left(T_{g,in}-T_{g,out}\right)}{m_g\cdot C_{p,g}\cdot\left(T_{g,in}-T_{g,dew}\right)} \tag{1}$$

$$UR_{ec}=\frac{m_{ec}\cdot C_{p,ec}\cdot\left(T_{ec,in}-T_{ec,out}\right)}{m_{ec}\cdot C_{p,ec}\cdot\left(T_{ec,in}-T_{ec,return}\right)} \tag{2}$$

Besides, a thermo-economic evaluation model is also built and illustrated as following. It should be noted that common manufacturing technique and carbon steel is considered in modeling process. The modeling process can be divided into three steps: firstly, the area of each heat exchanger should be calculated, and then the component capital cost can be obtained based on the components' capability. Finally, electricity production cost (*EPC*) can be determined.

The TEMA-E type shell tube heat exchanger is adopted in this study. Considering the sealing performance, the high pressure working fluid flows in tube side while the heat source (heat sink) flows in shell side. The tube pattern in exchanger is 60°. Tube-side passes and shell-side passes are 4 and 1, respectively. The heat exchange tube is the common stainless steel tube with 10 mm tube outside diameter and 1 mm tube thickness. The pinch between these heat exchange tubes is set as 20 mm for preheater, gas heater and condensers while 15 mm for regenerator. The shell diameters can be calculated by Eq. (3)

$$D_{\text{shell}} = \left(1.1\sqrt{n_{\text{t}}} - 1\right) \cdot st + 1.5 d_{\text{o}} \tag{3}$$

where n_{t} is the tube number and d_{o} is the tube outside diameter. *st* is the spacing between the adjacent tubes. The shell side is provided with the horizontal baffle. The baffle space and baffle cut can be determined as following.

$$Space = 0.5 \cdot D_{\text{shell}} \tag{4}$$

$$Cut = 0.25 \cdot D_{\text{shell}} \tag{5}$$

After determined the geometric parameters of each heat exchange, the heat transfer coefficients are calculated by numerical correlation shown in Table 4. Finally, the total heat transfer coefficient and heat transfer area can be determined by:

$$\frac{1}{U} = \frac{1}{h_{\text{shell}}} + \frac{1}{h_{\text{tube}}} \cdot \frac{d_{\text{o}}}{d_{\text{i}}} + \frac{d_{\text{o}}}{2k} \ln\left(\frac{d_{\text{o}}}{d_{\text{i}}}\right) \tag{6}$$

$$A = \frac{Q}{U \cdot \Delta T \cdot F_t} \tag{7}$$

wherein U is the total heat transfer coefficient; h_{shell} is the heat transfer coefficient in the shell side and h_{tube} is the heat transfer coefficient in the tube side. d_{i} is tube inside diameter. k is the thermal conductivity of tube material. Q is heat flux and ΔT is the log mean temperature difference. F_t is the correction factor. It should be noted that the maximum pressure drops of 10 kPa for both sides are set as the limitation condition in heat exchanger design process.

Table 4 Summary of heat transfer correlation

Heat exchanger	Tube side	Shell side
Preheater /Gas heater /Regenerator	The Krasnoshchekov–Protopopov correlation [27]	Bell–Delaware method [28]
Condenser-1	Petukhov correlation [29]	
Condenser-2	Chen correlation [30]	

Knowing the area of each heat exchanger, the bare component cost of each component can be calculated by the Module Costing Technique, which is described in our previous work [6]. The system capital cost is the sum of all components. Based on that, the electricity production cost, which means the required cost per kilowatt electricity, can be determined as follow:

The capital recovery cost can be calculated as:

$$CRF = i \cdot (1+i)^{\text{time}} / ((1+i)^{\text{time}} - 1) \tag{8}$$

where i is the interest rate with the value of 5%; time is the economic life time with the value of 15 years. Hence, the annuity of the investment is obtained by following equation.

$$A_{\text{nk}} = Cost2014 \cdot CRF \tag{9}$$

where *Cost*2014 is the total system capital cost in the situation of year 2014. Finally, the electricity production cost could be determined by:

$$EPC = \frac{(A_{\text{nk}} + f_{\text{K}} \cdot Cost2014)}{(W_{\text{net}} \cdot h)} \tag{10}$$

where f_{K} is the operation, maintenance and insurance cost factor, whose value is 1.65%; h is the full-load operation hours, which is set as 7500 h.

4. Results and Discussion

In this section, a comprehensive comparison between hydrocarbons and CO_2 is conducted from the thermodynamic and economic aspects. Firstly, the optimal operating pressure is determined and the effect of working fluid on these optimal pressures is discussed.

4.1 Optimal operating pressure for various indictors

The effect of turbine inlet pressure on above three indicators is shown in Fig. 2. For Propane in Fig. 2(a), the net power output and exergy efficiency both increase first and then decrease with increase of turbine inlet pressure. The maximum net power output (27.1 kW) and maximum exergy efficiency (34.8%) are obtained at P_{h}=8.71 MPa and P_{h}=9.78 MPa, respectively. However, the electricity production cost (*EPC*) changes in the opposite trend and the minimum *EPC* is achieved at P_{h}=6.16 MPa. As for hexane in Fig. 2(b), it could be observed that the change trends of net power output and exergy efficiency are same with these of propane while the *EPC* increases constantly with turbine inlet pressure.

It is not surprise that there exist optimal turbine inlet pressure to achieve $W_{\text{net,max}}$, $\eta_{\text{ex,max}}$, and EPC_{min}. Similar result was also got in Ref. [26]. It is worth noting that optimal turbine inlet pressures corresponding to these maximum and minimum values are relatively different. For Propane, the optimal pressure for the maximum net power output P_{h} ($W_{\text{net,max}}$) is lower than the one corresponding to maximum exergy efficiency P_{h} ($\eta_{\text{ex,max}}$), but opposite situation is presented for Hexane. This regularity can be explained by Fig. 3(a). For Propane, the UR_{ec} decreases and UR_g remains almost constant with the turbine inlet pressure increasing. The total heat input declines, so does exergy. Hence, P_h ($\eta_{ex,max}$) would be higher than P_h ($W_{net,max}$) according to the exergy efficiency definition. On the contrary, the total heat input decreases for Hexane.

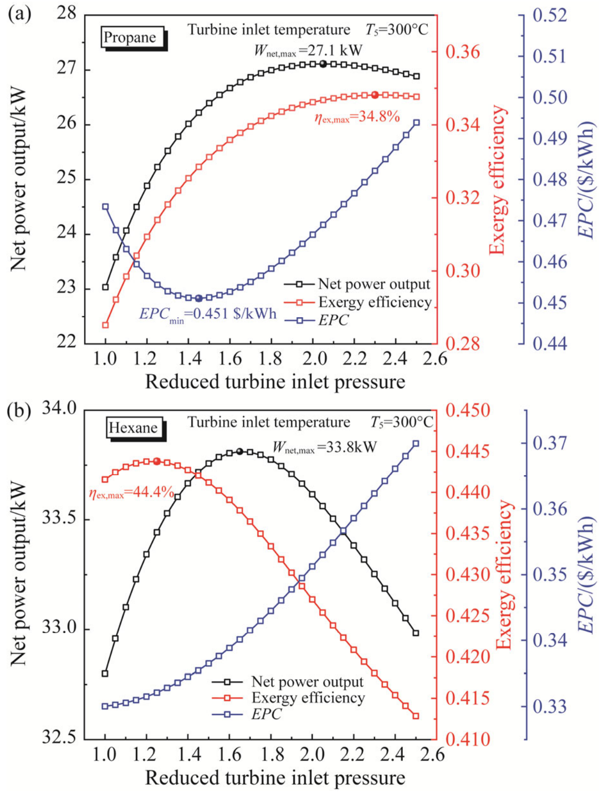

Fig. 2 Effect of turbine inlet pressure on net power output, exergy efficiency and electricity production cost at T_5=300°C. (a) Propane; (b) Hexane.

A deeper explanation is presented from the working fluid specific heat variation as plotted in Fig. 3(b) and (c). The enclosed area between the c_p curve and X axis represents heat absorption amount per mass fluid. It is well known that specific heat c_p will reach the peak at certain temperature for supercritical fluid. The maximum c_p point is the pseudocritical point and the corresponding temperature is called the pseudocritical temperature. The peak c_p would move towards to high temperature direction and its value would decrease with pressure in supercritical region. Depending on the heat sources, the high pressure specific heat curve is divided into three sections. The left section (below T_3) corresponds to the engine coolant waste heat recovery zone; the middle section (range from T_3 to T_4) corresponds to the regenerated heat recovery zone, and the right section (range from T_4 to T_5) corresponds to the exhaust gas waste heat recovery zone. For propane with relatively low critical temperature, it is appreciated that the peak c_p is located in the regenerated heat recovery zone (near the engine coolant waste heat recovery zone). The increase of turbine inlet pressure would lead to: (1) a loss in enclosed area of engine coolant waste heat recovery zone, and (2) a decrease of both the regenerated heat and the enclosed area of regenerated heat recovery zone. Hence, the engine coolant utilization rate decreases while exhaust gas utilization rate remains almost constant with the turbine inlet pressure increasing for Propane. Similar analytical method based on Fig. 3(c) can be also used for Hexane.

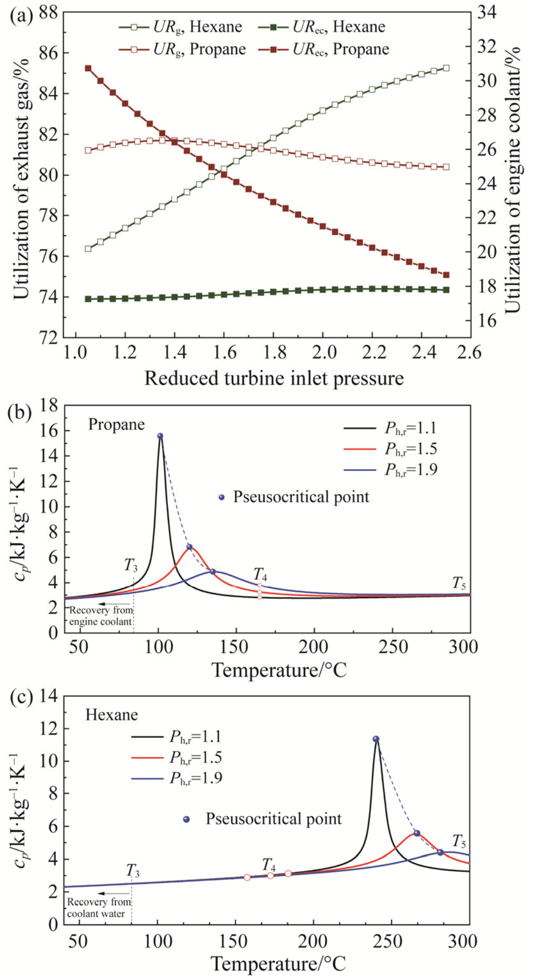

Fig. 3 (a) Utilization rate variation of exhaust gas and engine coolant between Propane and Hexane; (b) Specific heat c_p variation of Propane; (c) Specific heat c_p variation of Hexane.

Having reviewed the detailed findings for Propane and Hexane, more results about the optimal turbine inlet pressures corresponding to $W_{net,max}$, $\eta_{ex,max}$ and EPC_{min} for selected hydrocarbons are listed in Table 5. It is evident that the optimal pressure P_h ($W_{net,max}$) is lower than P_h ($\eta_{ex,max}$) for fluids with low critical temperature (Ethane, Propene, Propane, Isobutane). The opposite results can be found for fluid with high critical temperature (Pentane, Hexane, Cyclopentane, Cyclohexane). This result is consistent with previous observations regarding Propane and Hexane. The minimum *EPC* is obtained at pressure that is always lower than the previous two. Additionally, the optimal pressures using CO_2 are also listed in Table 5. Optimal net power output and *EPC* appear at the same turbine inlet pressure using CO_2 as working fluid. These results are then used in the next section to analyze and compare the system performance between hydrocarbons and CO_2. It should be noted that the optimal performance value is obtained at $P_h=1.1\times P_{cir}$ for fluids who prefer a low turbine inlet pressure.

Table 5 Optimal turbine inlet pressures for maximum net power output, maximum exergy efficiency and minimum *EPC*

	P_h ($W_{net,max}$) /MPa	P_h ($\eta_{ex,max}$) /MPa	P_h (EPC_{min}) /MPa
CO_2	12.91	14.02	12.91
Ethane	8.04	10.96	7.80
Propene	9.11	10.48	6.60
Propane	8.71	9.78	6.16
Isobutane	5.44	7.44	3.99
Pentane	6.57	5.22	↓
Hexane	5.01	3.79	↓
Cyclopentane	6.32	4.97	↓
Cyclohexane	4.48	↓	↓

↓-Lower value preferred

4.2 System performance comparison between CO_2 and hydrocarbons

A comprehensive comparison between CO_2 and hydrocarbons is presented from their properties in this section. The following comparison is conducted under the above-mentioned optimal pressures.

4.2.1 Thermodynamic performance comparison

The exhaust gas utilization rate (UR_g) and engine coolant utilization rate (UR_{ec}) are plotted in Fig.4. It is obvious that all selected working fluid can achieve considerable exhaust gas utilization rate (>60%). However, only CO_2 and Ethane have preferable coupling recovery capability for these two waste heat sources. Compared with CO_2 and Ethane, other hydrocarbons can recover exhaust gas waste heat more efficiently but are significantly weaker in absorbing heat from low-temperature waste heat (engine coolant).

To explain the difference of utilization rates among various working fluids, the specific heat comparison between hydrocarbons and CO_2 is shown in Fig. 5, it is appreciated that the peak specific heat c_p of Ethane is located in the engine coolant recovery zone due to relatively low critical temperature. As a result, Ethane could recover more heat from engine coolant per mass and achieve a higher engine coolant utilization rates. As for exhaust gas, it is well known that the adding a regenerator results in a decrease of exhaust gas utilization rate, which is largely determined by the degree of heat regeneration. As we will show in the next section, fluids with high critical temperature are inclined to achieve higher enthalpy drop during expansion process. Therefore, the heat regeneration degree is small and exhaust gas utilization rate can be improved. For Propane, Propene and Isobutane, the peak specific heats are located in the regenerated heat recovery zone (as shown in Fig. 5) and a higher exhaust gas utilization rate can also be achieved. Comparison between CO_2 and Ethane shows that another advantage of CO_2, in addition to the peak specific heat in engine coolant recovery zone, is the higher working fluid mass rate owing to its low specific heat (shown in Fig. 5). Hence, UR_{ec} and UR_g of CO_2 are both higher than these of Ethane.

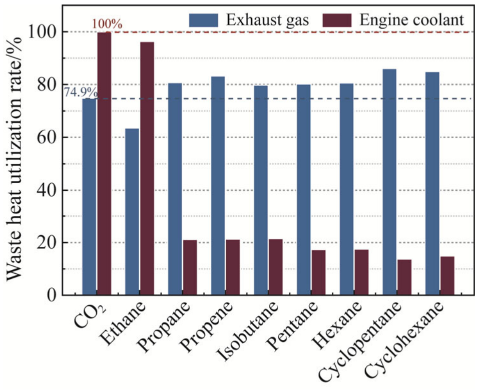

Fig. 4 Waste heat utilization rates of exhaust gas (UR_g) and engine coolant (UR_{ec}) for hydrocarbons and CO_2 at P_h ($W_{net,max}$)

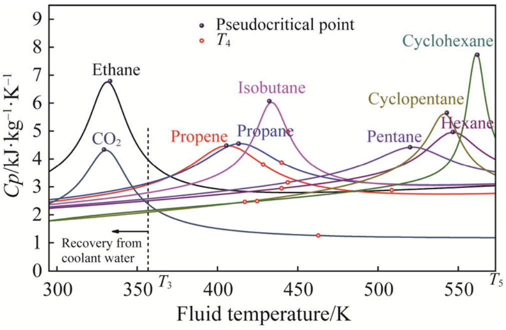

Fig. 5 Variation of specific heat c_p for hydrocarbons and CO_2 at P_h ($W_{net,max}$)

Fig. 6 exhibits the net power output ($W_{net,max}$) comparison between hydrocarbons and CO_2. It is clear that $W_{net,max}$ increases with carbon atoms number for linear hydrocarbons. From Ethane to Hexane, $W_{net,max}$

increases from 24.2 kW to 33.8 kW. Cyclic hydrocarbons are superior to linear hydrocarbon in power output ability. $W_{\mathrm{net,max}}$ of Cyclopentane and Cyclohexane are 35.4 kW and 36.4 kW, respectively. CO_2 yields more net power output than Ethane, Propane and Propene with 28.3 kW.

Few previous researches [31,32] have discussed the relationship between several working fluid thermodynamic properties and power output in subcritical cycle. The follow analysis will provide an insight to the effect of main thermodynamic properties on power output ability of working fluid in transcritical cycle. Fig. 7 depicts the cycle process using different working fluids in *T-s* plot. The area size enclosed by solid line represents the specific net power output. It is clear that the difference in area size among various working fluids is closely related to the condensation latent heat (r), specific heat (c_p), as well as critical temperature (T_c). Fluids with higher condensation latent heat (r) are better suited to obtain more net power output. Concerning specific heat (c_p) and critical temperature (T_c), the slope of isobaric curve in *T-s* graph can be expressed as Eq. (11). In Eq.(11), s is the entropy. It can be known that the slope of isobaric curve increase with the decline of specific heat. Fig. 7 shows that high pressure isobaric curve is always concave due to the peak specific heat at pseudocritical temperature as above-mentioned. For working fluids with low critical temperature like CO_2 and Ethane, high pressure isobaric curves tend to be concave at relatively low temperature, which contributes negatively to power output. Accordingly, the specific net power output of CO_2 and Ethane is obviously smaller than other working fluids. For the specific heat, the specific heat of CO_2 is significantly lower than these of hydrocarbons, which leads to a higher working fluid mass flow. This is the reason why CO_2 obtains more power than Ethane, Propane and Propene in spite of small enthalpy drop.

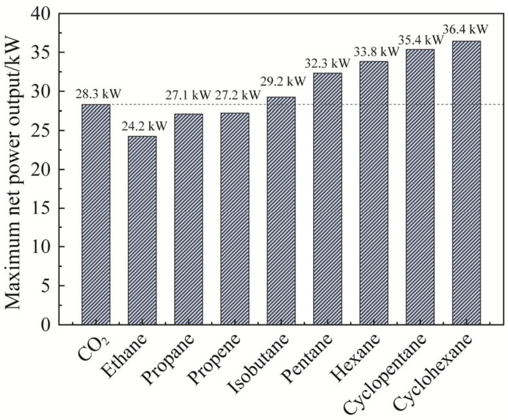

Fig. 6 Maximum net power output comparison of transcritical cycle using hydrocarbons and CO_2

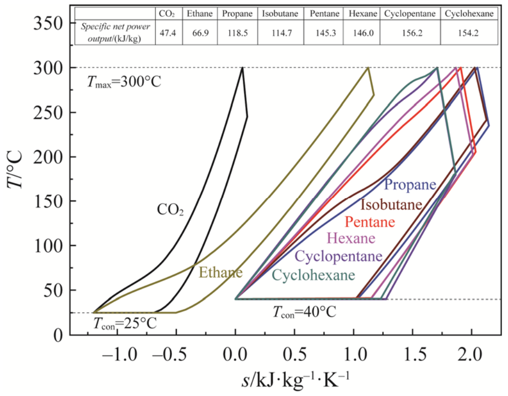

Fig. 7 Comparison of output power capability between hydrocarbons and CO_2

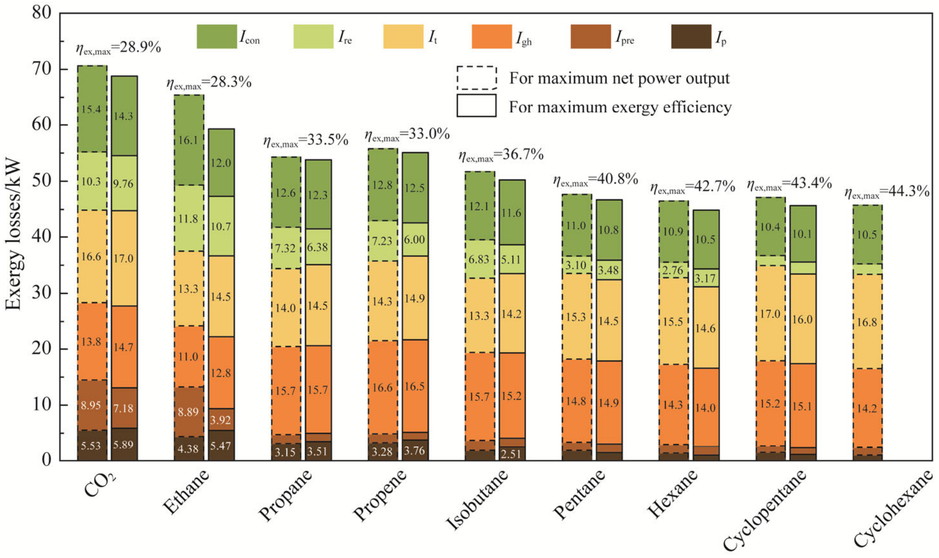

Fig. 8 Component irreversibility distribution of transcritical cycle using hydrocarbons and CO_2

$$\left(\frac{\partial T}{\partial s}\right)_p = \frac{T}{c_p} \tag{11}$$

Fig. 8 gives the exergy destruction and exergy efficiency. I_{con}, I_{re}, I_{t}, I_{gh}, I_{pre} and I_{p} are exergy destruction in the condenser, regenerator, turbine, gas heater, preheater and pump, respectively. For linear hydrocarbons, the maximum exergy efficiency increases from 28.3% to 42.7% with carbon atom number from 2 to 6. Cyclic hydrocarbons obtain higher exergy efficiency and the maximum exergy efficiencies of Cyclopentane and Cyclohexane are 43.4% and 44.3%, respectively. CO_2 has the largest total exergy destructions with exergy efficiency of 28.9%. For exergy losses of each component, exergy losses in the gas heater, condenser and turbine are dominant for system exergy losses for all fluids. However, exergy destructions in preheater, regenerator and pump are apparently different using various working fluids. This is due to differences of engine coolant utilization rate, heat regeneration degree and working fluid mass flow rate based on the above analysis.

4.2.2 Component size comparison

Component size comparison is markedly significant for truck engine waste heat recovery system. Fig. 9 shows the heat transfer areas of each heat exchanger using different working fluids. A_{pre}, A_{gh}, A_{re}, A_{con} are heat transfer areas of the preheater, the gas heater, the regenerator and the condenser, respectively. Among hydrocarbons, Ethane requires the largest total heat transfer areas. Conversely, Propane, Propene and Cyclopentane require the smallest heat transfer areas. The total heat transfer area of CO_2 is almost equal to that of Cyclohexane. As for areas of each heat exchanger, it is normal that heat transfer area of gas heater A_{gh} depends on the exhaust gas utilization rate to a great extent. Fluids with higher exhaust gas utilization rate require larger gas heater. Meanwhile, the condenser size is greatly determined by the cooling load. The major concern of this paper is the preheater and preheater and regenerator. As shown in Table 6, the heat transfer amount in preheater using CO_2 is 572.4% higher than that using cyclohexane (23.2 kW), but preheater area using CO_2 (10.1 m^2) is just 46.4% higher than that of Cyclohexane (6.9 m^2). The heat transfer coefficients using CO_2 are always larger than these using hydrocarbons in preheater and regenerator. The obvious difference of heat transfer coefficient is caused by: (1) CO_2 has relatively small viscosity, which means good fluidity. The biggest viscosity difference between CO_2 and hydrocarbons occurs in the temperature range of 0°C–150°C corresponding to the temperature range in preheater and regenerator. (2) the sudden rise of specific heat near pseudocritical points (shown in Fig. 5) would be beneficial to heat transfer enhancement [33]. Based on above analysis, it can be concluded that CO_2 would be advantageous in preheater and regenerator size.

To evaluate and compare the turbine performance, volumetric flow ratio (*VFR*) and size parameter (*SP*) are adopted in this paper. The *VFR* and *SP* are defined as:

$$SP = \frac{\sqrt{V_{out}}}{\left(\Delta h_{is}\right)^{0.25}} \tag{12}$$

$$VFR = \frac{V_{out}}{V_{in}} \tag{13}$$

where V_{in} and V_{out} are the volume flows at the inlet and outlet of the turbine. Δh_{is} is the isentropic enthalpy difference during the expansion process. For linear hydrocarbons as shown in Fig. 10, it is clear that large turbine size is needed for hydrocarbons with more carbon atoms except Ethane. It is evident that *SP* values of cyclic hydrocarbons are larger than that of corresponding

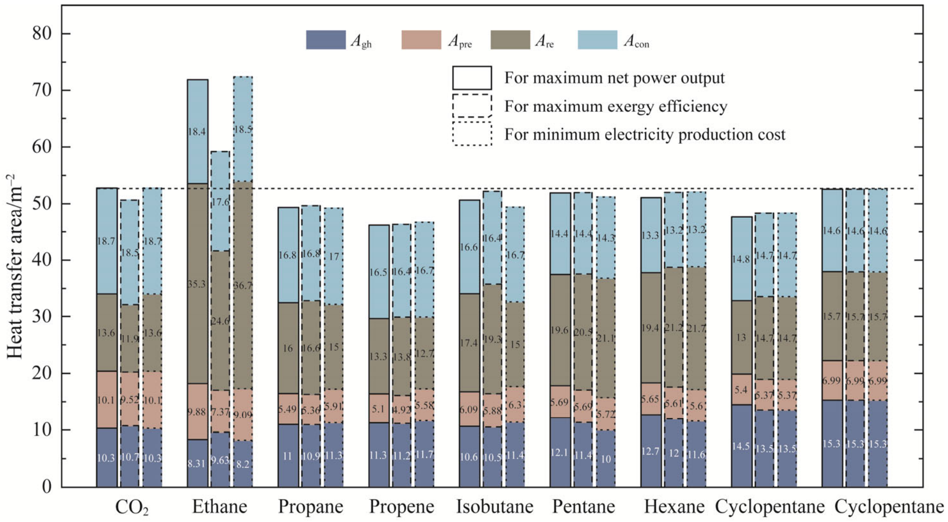

Fig. 9 Heat transfer areas of each heat exchangers in transcritical Rankine cycle for different working fluids

linear hydrocarbons with the same carbon atoms. *SP* value of CO_2 is smaller than that of most of hydrocarbons except Propane and Propene. In general, the working fluid including CO_2 (*SP*=0.042 m), Ethane (*SP*=0.046 m), Propene (*SP*=0.040 m) and Propane (*SP*=0.042 m) require lower *SP*. *SP* depends on working fluid volumetric flow at turbine exit and isentropic entropy drop in turbine. It is appreciated that the gas density of CO_2 is apparently larger than hydrocarbons, which results in a small volumetric flow and then a small *SP* value. As for *VFR*, the comparison result of *VFR* is similar with that of *SP.* It can be observed that the minimum *VFR* of 1.8 is achieved by CO_2. Beside CO_2, the *VFR* values of the Ethane, Propane, Propene and Isobutane are all below 50, which is considered as the criteria of judging whether the turbine efficiency can reach 80%. For high critical-temperature hydrocarbons, due to the higher turbine expansion ratio, multi-stage layout has to be employed with more turbine producing problems, high turbine size and less favorable for turbine manufacture.

Table 6 Heat transfer amounts and heat transfer coefficients in preheater and regenerator for different fluids

	Preheater		Regenerator	
	Q/ kW	*U*/ $W \cdot m^{-2} \cdot K^{-1}$	*Q*/ kW	*U*/ $W \cdot m^{-2} \cdot K^{-1}$
CO_2	156.0	911.7	166.8	400.0
Ethane	149.6	889.2	266.2	319.8
Propane	32.9	446.7	109.9	193.4
Propene	33.1	493.1	100.7	204.7
Isobutene	33.4	372.3	112.3	180.1
Pentane	27.0	321.6	69.2	143.2
Hexane	27.2	317.4	64.8	143.4
Cyclopentane	21.4	267.2	39.4	131.8
Cyclohexane	23.2	216.2	43.5	124.4

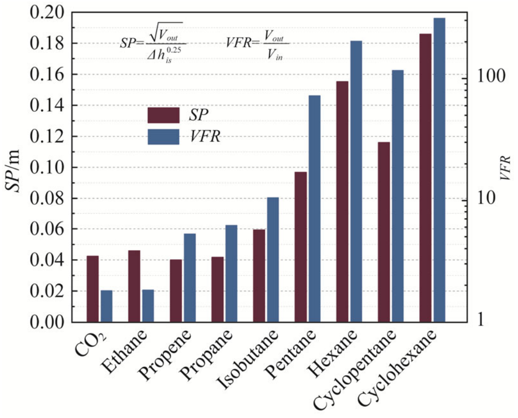

Fig. 10 Comparison of turbine size parameter (*SP*) and volume flow ratio (*VFR*) between hydrocarbons and CO_2

5. Conclusions

This paper presented a comprehensive comparison between hydrocarbons and CO_2 in transcritical Rankine cycles for engine's multiple waste heat recovery. Net power output, exergy efficiency and electricity production cost (EPC) were selected as the system performance indicators. The optimal turbine pressures based on above three indicators were determined and the effect of working fluid on these optimal pressures was discussed. Finally, thermodynamic performance and component size comparison between hydrocarbons and CO_2 was conducted based on their thermo-physical properties. Based on above results and discussion, following conclusions can be drawn:

(1) The optimal turbine inlet pressures for above three performance indicators are different. For fluids with relatively low critical temperature (CO_2, Ethane, Propene, Propane, Isobutane), P_h ($W_{net,max}$) is lower than P_h ($\eta_{ex,max}$). The opposite results are presented for fluid with high critical temperature (Pentane, Hexane, Cyclopentane, Cyclohexane). Optimal pressure for minimum EPC is lower than the previous two.

(2) Great difference exists in the engine coolant utilization rate for various working fluids. Thanks to the pseudocritical temperature matching with engine coolant temperature range, CO_2 and Ethane have preferable coupling recovery ability for exhaust gas and engine coolant.

(3) Compared to hydrocarbons, CO_2 yields more power output than Ethane, Propane and Propene. The main thermodynamic properties affecting working fluid power output ability in transcritical cycle are critical temperature, condensation latent heat and evaporating specific heat.

(4) CO_2 -based cycle can absorb more heat from engine coolant and regeneration heat with comparable total heat transfer areas and has an advantage in turbine size.

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