PERFORMANCE ANALYSIS OF ORC SYSTEM WITH IHE USING THE ZEOTROPIC MIXTURE AND THE PURE WORKING FLUID FOR VEHICLE CNG ENGINE

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ABSTRACT

In order to improve the thermal efficiency of compressed natural gas (CNG) engine, a set of organic Rankine cycle (ORC) system with internal heat exchanger (IHE) is designed to recover exhaust waste heat from the CNG engine. The working fluids under investigation are the pure working fluid R245fa and the zeotropic mixture R416A. Subsequently, the influence of the two different working fluids on performance parameters such as net power output, thermal efficiency, exergy efficiency and output energy density of working fluid are analyzed. The results show that the zeotropic mixture R416A performs better. Finally, a combined CNG engine and ORC system with IHE is defined to evaluate the performance improvement. Results show that compared with the CNG engine, the thermal efficiency of the combined system can be increased by a maximum 7%.

1. INTRODUCTION

The thermal efficiency of most compressed natural gas (CNG) engines are 30% or so and they are lower than that of diesel engines, large amount fuel energy are rejected from CNG engines to the surroundings as waste heat, with a significant fraction through the exhaust. Therefore, recovering the exhaust waste heat from CNG engine so as to improve thermal efficiency and save fuel has become a hot focus of recent research work (Wang et al., 2014).

The organic Rankine cycle (ORC) is a promising method to recover waste heat from internal combustion engines (ICE) exhaust gas (Chiew *et al*., 2011, Wang *et al*., 2011, Zhang *et al*., 2014). Vaja *et al*. (2010) designed a power cycle equipment to match a stationary internal combustion engine, and accordingly chose three pure working fluids to examine three different ORC schemes separately. The analysis demonstrated that a 12% increase in the total efficiency could be achieved with respect to the engine with no ORC cycle. Yang *et al*. (2014) designed a set of dual loop ORC system to recover exhaust energy, waste heat from the coolant system, and released heat from turbocharged air in the intercooler of a six-cylinder diesel engine. R245fa was selected as the working fluid for both loops. The results showed that the dual loop ORC system achieves the largest net power output at 27.85 kW at the engine rated condition. Compared with the diesel engine, the thermal efficiency of the combined system can be increased by 13%.Yu *et al*. (2013) built an ORC system to recover waste heat both from engine exhaust gas and jacket water using R245fa as working fluid. Results indicated that the ORC system performances well under the rated engine condition with expansion power up to 14.5 kW, recovery efficiency up to 9.2% and exergy efficiency up to 21.7%. Combined with bottoming ORC system, thermal efficiency of diesel engine can be improved up to 6.1%.

For the working fluids of ORC system, the match of working fluids with heat source and systems significantly affects system performance. Zeotropic mixtures have a property called "temperature glide" in evaporation and condensation process and this can reduce exergy destruction rate due to heat transfer temperature difference (Wu *et al*., 2012). Radulovic and Castaneda (2014) proposed six zeotropic mixtures for conducting a parametric optimisation of supercritical Rankine cycle powered by low temperature geothermal heat source. And then a comparative analysis between the zeotropic mixtures and pure R-143a was studied. The results showed that the cycle efficiency can be improved by 15% at the same operational conditions. Lecompte *et al*. (2014) examined the thermodynamic performance of non-superheated subcritical organic Rankine cycles (ORCs) with zeotropic mixtures as working fluids based on a second law analysis. The results showed that an increase in second law efficiency in the range of 7.1% and 14.2% is obtained compared to pure working fluids. Furthermore, between optimized ORCs with zeotropic mixtures as working fluid the difference in second law efficiency varies less than 3 percentage points.

According to abovementioned analysis, zeotropic mixtures have bigger potential in optimizing the ORC system performances than pure working fluids, whereas few scholars use zeotropic mixtures for an ORC system in recovering the exhaust energy from a CNG engine.

2. EVALUATION OF EXHAUST WASTE HEAT FROM CNG ENGINE

To design an optimal ORC system that can efficiently recover the exhaust waste heat from CNG engine, it is necessary to analyze the energy distribution in the running process of the engine. In this paper, a six-cylinder four-stroke CNG engine was employed, and the main performance parameters were listed in Table 1. Based on the performance test data of engine experimental system, the running performances and exhaust energy of the CNG engine under various operating conditions were analyzed. During the experiment, the CNG engine speed ranged from 1800 r/min to 2200 r/min at an interval of 200 r/min. The eleven operating condition points were selected and tested under each engine speed.

Table 1: Main technical performance parameters of the CNG engine

Items	Parameters	Units
Cylinder number	6	
Rated power	206	kW
Displacement	8.3	L
Stroke and cylinder bore	135×114	mm
Compression ratio	10.5	
Maximum torque	1050	N.m

The universal characteristic of CNG engine is shown in Figure 1. As seen in this figure, the blue contour lines indicate the variations in the effective power output of the engine under engine various operating conditions. The black contour lines indicate the variations in the brake specific fuel consumption (BSFC) of the engine under engine various operating conditions. At the engine rated condition, the effective power output of the CNG engine is 206.9 kW. In addition, the brake specific fuel consumption (BSFC) is relatively low in the engine's medium speed with medium-high load regions. On the contrary, the brake specific fuel consumption (BSFC) is relatively higher in the engine's low load region, particularly in the high speed region. When the engine speed is 1400 r/min and engine torque is 1050 N.m, the CNG engine achieves the optimal fuel economy, and the BSFC is 199 g/(kW.h).

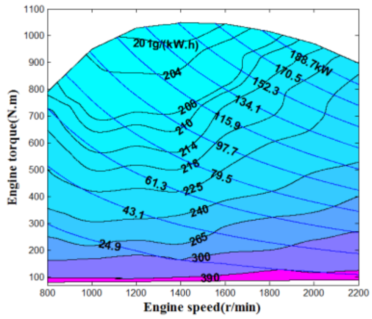


Figure 1: Universal characteristic of CNG engine

Figure 2 shows the thermal efficiency under engine various operating conditions. The thermal efficiency η_{en} can be calculated using the following equation:

$$\eta_{\mathrm{en}} = \frac{\dot{W}_{\mathrm{en}}}{\dot{Q}_{\mathrm{f}}} \tag{1}$$

Where, $\dot{W}_{\mathrm{en}}$ is the effective power output of CNG engine, $\dot{Q}_{\mathrm{f}}$ is the fuel combustion energy which can be calculated using the fuel consumption rate and fuel lower heating value. In this paper, the fuel lower heating value is 50050 kJ/kg.

As seen in the Figure 2, the thermal efficiency increases with the engine torque. On the other hand, the thermal efficiency firstly increases and then decreases with engine torque, whereas the variation trend is not especially obvious in the engine's low load region. When the engine speed is 1400 r/min and engine torque is 1050 N.m, the thermal efficiency of CNG engine can reach up to a maximum of 36.14%.

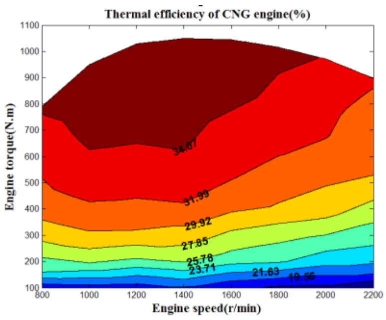


Figure 2: Thermal efficiency of CNG engine

Figure 3 illustrates the variation of exhaust temperature in different engine speed with the engine torque. As shown in Figure 3, exhaust temperature is relatively low under low speed with low torque conditions, and relatively high under both medium-high speed and medium-high torque conditions. But the overall variation trend is complex. From Figure 3 it can be seen that exhaust temperature is in the range of 730 K to 900 K.

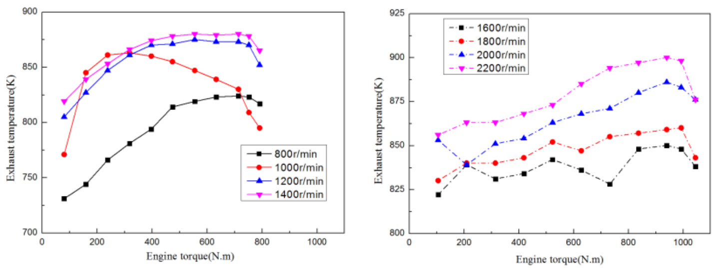


Figure 3: Exhaust temperature of CNG engine

When calculating exhaust energy, methane is assumed to be the natural gas fuel and other substances contained in the natural gas are ignored for this study. Air-fuel ratio of the CNG engine is set to 17.2 (by mass). According to chemical reaction equation for the combustion process, mass fraction of the exhaust components CO_2、H_2O and N_2 is 15.1%、12.4% and 72.5%, respectively. Subsequently, the specific enthalpy of the exhaust under a specific temperature is calculated using the thermodynamic calculation method of ideal gas and REFPROP 9.0 software. In addition, exhaust mass flow rate $\dot{m}_{\mathrm{exh}}$ can be calculated using air-fuel ratio and fuel consumption which measured by experiment. Finally, the exhaust energy $\dot{Q}_{\mathrm{exh}}$ of CNG engine can be calculated using equation (2).

$$\dot{Q}_{\mathrm{exh}} = \dot{m}_{\mathrm{exh}} h_{\mathrm{exh}} \tag{2}$$

Where, h_{exh} is the specific enthalpy of the exhaust.

In practical ORC system, exhaust energy can not be totally absorbed by working fluids (Yu *et al*. 2013). Therefore, in this paper, the exhaust temperature and pressure at the outlet of evaporator are set to 378 K and 97.8 kPa, respectively. Subsequently, specific enthalpy of the exhaust under the set exhaust temperature and pressure is calculated using REFPROP 9.0 software. Finally the available exhaust energy rate can be calculated using equation (3).

$$\dot{Q}_{ava} = \dot{m}_{\mathrm{exh}} (h_{\mathrm{in}} - h_{\mathrm{out}}) \tag{3}$$

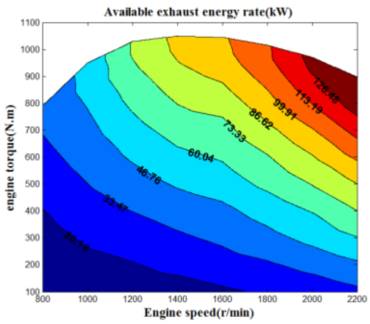


Figure 4: Available exhaust energy rate of CNG engine

According to the performance test data of CNG engine, the variation tendency of available exhaust energy rate under engine various operating conditions is shown in Figure 4. It is evident that available exhaust energy rate increases over the engine whole operating range, achieving 139.58 kW at the engine rated condition.

3. EXHAUST WASTE HEAT RECOVERY SYSTEM BASED ON ORC WITH IHE

3.1 System Description

In order to recover the exhaust waste heat from CNG engine efficiently, the ORC system with IHE is designed as shown in Figure 5. The system consists of an evaporator, a condenser, an expander, a recuperator (namely IHE), a pump, a reservoir and a generator. At first, the exhaust gas exchanges heat with the organic working fluid in the evaporator, then the exhaust gas is released through evaporator into the atmosphere. Meanwhile the organic working fluid turns into high-temperature and high-pressure gas and soon enters expander to generate electricity. Later, the organic working fluid exhausted from the expander goes into the recuperator to exchange heat with the liquid organic working fluid which exported from the pump. Subsequently, the cooled working fluid that is exhausted from recuperator condenses into a saturated liquid state and flows into the reservoir. The organic working fluid is pressurized by using pump and then absorbs heat in the recuperator. Finally, the organic working fluid flows into the evaporator to absorb the heat from engine exhaust.

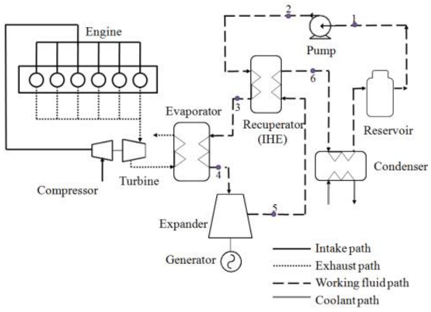


Figure 5: Schematic diagram of ORC system with IHE

3.2 Working Fluid Selection

The selection of working fluids has an important impact on thermodynamic performances of the ORC system. In this paper, there are two kinds of working fluids have been selected to study the performances of ORC system. Therein R245fa has performed well as the working fluid in ORC systems in many studies (Wang *et al*. 2011, Wang *et al*. 2012). In addition, because of the good environment friendliness and safety, zeotropic mixture R416A with property of temperature glide has been selected from the existing serial-numbered refrigerant for this study. The properties of R416A and R245fa are listed in Table 2.

Table 2: Properties of R416A

Parameters	**R416A**	**R245fa**
Components	R134a/R124/R600	~
Composition (mass fraction)	0.59/0.395/0.015	~
Critical temperature (K)	380.23	427.16
Critical pressure (MPa)	3.98	3.651
Glide temperature (K)	1.86	~
Safety	A1	B1
Environment friendliness (Yes/No)	Yes	Yes
Fluid type	Wet	Dry

Glide temperature: standard atmospheric pressure.

3.3 Thermodynamic Model

Figure 6 is the *T-s* diagram of the working fluid. Therein Process 1-2 is the actual pressurization process of pump, process 1-2s is the isentropic pressurization process. Process 2-3 is the isobaric endothermic process of the working fluids in the recuperator. Process 3-4 is the isobaric endothermic process, Process 4-5s is the isentropic expansion process, while the process 4-5 is the actual expansion process. Process 5-6 is the isobaric exothermic process of the organic working fluid in the recuperator. Process 6-1 is the isobaric condensation process. Process $T_{\mathrm{exh\_in}}$-$T_{\mathrm{exh\_out}}$ is the heat transfer process of the engine exhuast in the evaporator, $T_{\mathrm{exh\_in}}$ is the exhaust gas temperaturte at the inlet of the evaporator, $T_{\mathrm{exh\_out}}$ is the exhaust gas temperaturte at the outlet of the evaporator.

Based on the first and second laws of thermodynamics, the performance parameters of ORC system with IHE are calculated using the following equations:
In the process 1-2, the power consumption $\dot{W}_{\mathrm{p}}$ is calculated with the equation below.

$$\dot{W}_{\mathrm{p}} = \dot{m}(h_2 - h_1) = \frac{\dot{m}(h_{2s} - h_1)}{\eta_p} \tag{4}$$

Where, $\dot{m}$ is mass flow rate of organic working fluid, η_p is isentropic efficiency of pump.

In the processes 2-3 and 5-6, the heat transfer rate $\dot{Q}_{\mathrm{r}}$ of the recuperator is calculated using the following equation:

$$\dot{Q}_{\mathrm{r}} = \dot{m}(h_3 - h_2) = \dot{m}(h_5 - h_6) \tag{5}$$

ε is the effectiveness of recuperator, which can be calculated using equation (6):

$$\varepsilon = (T_5 - T_6)/(T_5 - T_2) \tag{6}$$

In the process 3-4, the heat transfer rate $\dot{Q}_{\mathrm{e}}$ of the evaporator is calculated using the following equation:

$$\dot{Q}_{\mathrm{e}} = \dot{m}(h_4 - h_3) \tag{7}$$

In the processes 4-5 and 4-5s, the power output $\dot{W}_{\mathrm{s}}$ of the expander is calculated using the following equation:

$$\dot{W}_{\mathrm{s}} = \dot{m}(h_4 - h_5) = \dot{m}(h_4 - h_{5\mathrm{s}})\eta_{\mathrm{s}} \tag{8}$$

Where, η_{s} is isentropic efficiency of expander.

In the process 6-1, the heat transfer rate $\dot{Q}_{\mathrm{c}}$ of the condenser is calculated using the following equation:

$$\dot{Q}_{\mathrm{c}} = \dot{m}(h_6 - h_1) \tag{9}$$

In the all equations, h and its subscripts represent the specific enthalpy value of each state point in the *T*-*s* diagram.

According to the above analysis, the net power output, thermal efficiency and exergy efficiency of ORC system with IHE are respectively calculated using the following equations:

$$\dot{W}_{\mathrm{n}} = \dot{W}_{\mathrm{s}} - \dot{W}_{\mathrm{p}} \tag{10}$$

$$\eta_{\mathrm{th}} = \frac{\dot{W}_{\mathrm{n}}}{\dot{Q}_{\mathrm{e}}} \tag{11}$$

$$\eta_{\mathrm{ex}} = \frac{\dot{W}_{\mathrm{n}}}{\dot{Q}_{\mathrm{e}}(1 - \frac{T_{\mathrm{L}}}{T_{\mathrm{H}}})} \tag{12}$$

Where, T_{H} is the temperature of the high temperature heat source that can be calculated using equation (13). $T_{\mathrm{exh\_in}}$ can be measured through engine test. $T_{\mathrm{exh\_out}}$ is the exhaust temperature at the outlet of evaporator. T_{L} is the temperature of the low temperature heat source.

$$T_{\mathrm{H}} = (T_{\mathrm{exh\_in}} - T_{\mathrm{exh\_out}}) / \ln(T_{\mathrm{exh\_in}} / T_{\mathrm{exh\_out}}) \tag{13}$$

In addition, output energy density of working fluid ρ is defined to evaluate the capability to produce useful work for per unit of mass of working fluids. Its calculation equation as follows:

$$\rho = \frac{\dot{W}_{\mathrm{n}}}{\dot{m}} \tag{14}$$

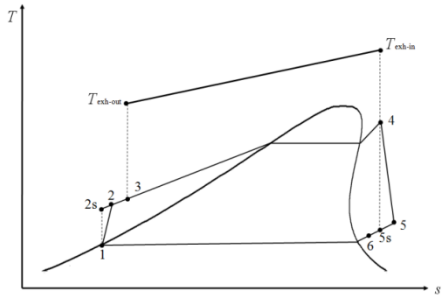


Figure 6:*T*-*s* diagram of ORC system with IHE

4. RESULTS AND DISCUSSION

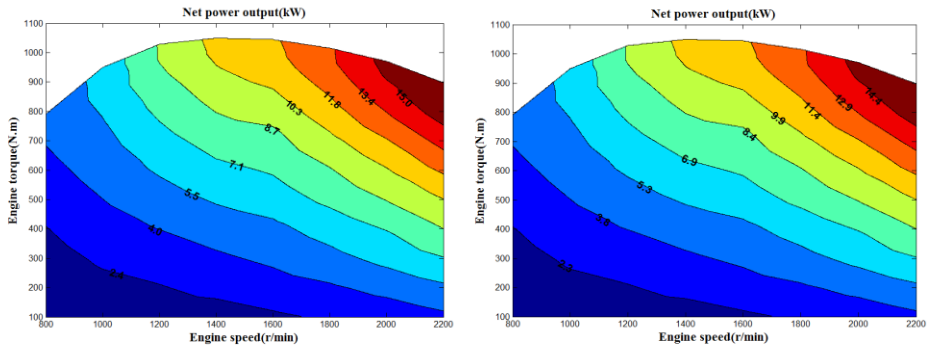
4.1 Boundary Conditions

In order to analyze the effects of R416A and R245fa on the performance of ORC system with IHE, in this paper, the boundary conditions are set as follows:

(1) Pressure drop and heat loss in each component as well as pipelines are neglected.
(2) Evaporation pressure is set to 2.5 MPa.
(3) The degree of superheat and T_L are set to 30 K and 293 K, respectively.
(4) The isentropic efficiencies of the expander and the pump are both set to 0.8.
(5) The working fluid releases heat in the condenser and then turns into a saturated liquid state.
(6) The expansion ratio of expander is set to 4.
(7) The effectiveness of recuperator is set to 0.85.

4.2 Results Analysis

Figure 7 illustrates how the net power output of the ORC system with IHE using the two different working fluids changes over the engine whole operating range. As shown in Figure 7(a) and (b), the variation tendency of the net power output using the two different working fluids are consistent, while the values are different. The net power output of the ORC system with IHE using the two working fluids increases with the increase of the engine's speed and load. At the engine's rated condition, the net power output for R416A reaches a maximum of 16.6 kW and a maximum of 16.0 kW for R245fa. Additionally, comparing Figure 7(a) and (b) shows that the net power output for R416A is larger than that of R245fa for each engine operating conditions. Therefore, by employing zeotropic mixture R416A, the performance of ORC system with IHE has better characteristics of power output.



(a) Net power output of R416A (b) Net power output of R245fa

Figure 7: Net power output of ORC system with IHE

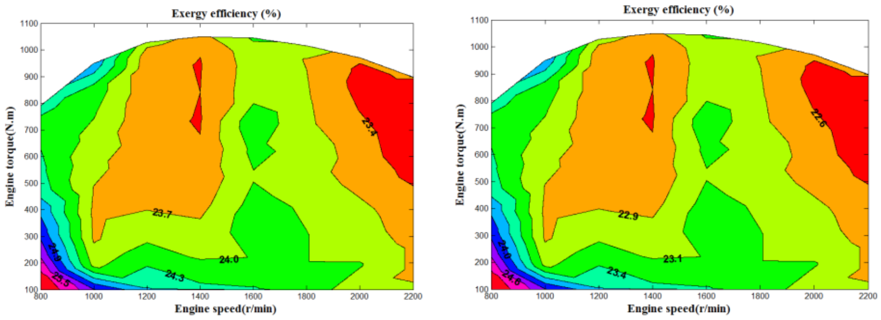
The results of the ORC system with IHE using the two different working fluids are summarized in Table 3. As shown in Table 3, the thermal efficiency and output energy density of working fluid for R416A are all larger than that of R245fa. For the same net power output of ORC system with IHE, the higher output energy density of working fluid, the less mass flow rate of working fluid is required for ORC system that can obviously reduce the working fluid mass filled in the ORC system. The total weight of the ORC system not only can be reduced but also the risk of environmental pollution can be significantly decreased.

Table 3: The results of two different working fluids

Performance parameters	**Working fluids**	
	R416A	R245fa
Thermal efficiency (%)	11.9%	11.4%
Condensing temperature (K)	299.7	344.1
Output energy density of working fluid (kJ/kg)	22.8	21.8

On the other hand, as listed in Table 3, we can see that the condensing temperature for R416A is lower than that of R245fa. As we all know, condensing temperature is one of the key factors which can influence the running performance of ORC system. Although the lower condensing temperature is beneficial for the running of ORC system, in practical application, the lower condensing temperature needs the higher-performance cooling system which can increase the cost of ORC system.

Figure 8 illustrates the variation of exergy efficiency with CNG engine speed and load. As shown in the Figure 8 (a) and (b), the variation tendencies of exergy efficiency using the two different working fluids are consistent. Namely, the exergy efficiency of ORC system with IHE is higher in the engine's low speed with low load regions. Whereas the exergy efficiency is lower in the engine's medium-high speed with medium-high load regions. The reason can be analyzed as follows: according to the equation (12), for the selected working fluid, exergy efficiency only depends on the exhaust temperature on the condition that the temperature of low temperature heat source is constant. Moreover, the variation trend of exhaust temperature is shown in Figure 3.



(a) Exergy efficiency of R416A (b) Exergy efficiency of R245fa

Figure 8: Exergy efficiency of ORC system with IHE

According to all of the results described above, we can conclude that the ORC system with IHE using R416A as the working fluid displays superior thermodynamic properties. Therefore, in order to optimize the power system as a whole and evaluate the improvement in overall power output, a ''combined CNG engine and ORC system with IHE'' is defined. The thermal efficiency of combined system η_{com} is calculated using the following equation:

$$\eta_{\mathrm{com}} = \frac{\dot{W}_{\mathrm{n}} + \dot{W}_{\mathrm{en}}}{\dot{Q}_{\mathrm{f}}} \tag{15}$$

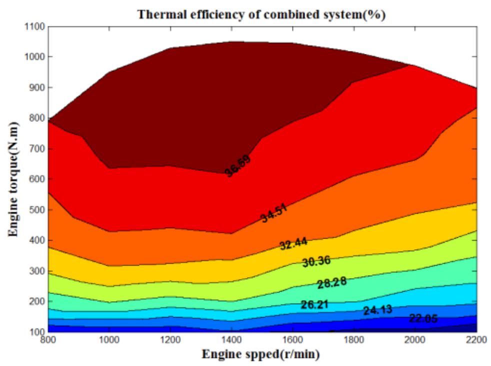


Figure 9: Thermal efficiency of combined system

The thermal efficiency of combined system is shown in Figure 9. When the engine speed is constant, the thermal efficiency of combined system increases with engine torque. On the other hand, when the engine torque is constant, the thermal efficiency of combined system firstly increases and then decreases with engine speed. When the engine speed is 1400 r/min and engine torque is 1050 N.m, the maximum thermal efficiency of combined system is 38.67%, which is higher than that of CNG engine by 7.0%.

5. CONCLUSIONS

- When evaporation pressure is 2.5 MPa and expansion ratio of expander is 4, the output energy density of working fluid, net power output, thermal efficiency, and exergy efficiency of ORC system with IHE using zeotropic mixture R416A are all superior to the same system using R245fa. Therefore, zeotropic mixtures have bigger potential in optimizing the ORC system performances than pure working fluids
- For the ORC system with IHE, condensing temperature for R416A is lower than that of R245fa. Although the lower condensing temperature is beneficial for the running of system, in engineering application, extremely low condensing temperatures cause difficulties in economically providing a low temperature heat source.
- The thermal efficiency of combined system increases with engine load, which is higher than that of CNG engine by 7.0%. From the viewpoint of power performance, the ORC system with IHE is a promising scheme to recover the exhaust waste heat from a CNG engine.

NOMENCLATURE

h	enthalpy	(kJ/kg)
s	entropy	(kJ/kg K)
$\dot{m}$	mass flow rate	(kg/s)
$\dot{W}$	power	(kW)
T	temperature	(K)
$\dot{Q}$	heat transfer rate	(kW)
η	efficiency	(–)
ε	effectiveness of recuperator	(–)

Subscript

1,2,2s,3,4,5,5s,6	state point in cycle
p	pump
e	evaporator
s	expander
r	recuperator
n	net
c	condenser
exh	exhaust gas
in	inlet
out	outlet
L	low temperature heat source
H	high temperature heat source
th	thermal
com	combined system
en	engine
ex	exergy
f	fuel
ava	available

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