PARAMETRIC OPTIMIZATION AND PERFORMANCE ANALYSIS OF ORGANIC RANKINE CYCLE (ORC) FOR ENGINE WASTE HEAT RECOVERY

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ABSTRACT

This study examines the parametric optimization and performance analysis of ORC system using genetic algorithm (GA) for engine waste heat recovery. The effects of three key parameters, including evaporation pressure, superheat degree, and condensation temperature on the net power output per unit heat transfer area and exergy destruction rate under engine various operating conditions are analyzed. Subsequently, the performances of a finned-tube evaporator used in this ORC system are evaluated. The results indicate that the optimal evaporation pressures are mainly influenced by the engine operating conditions. Moreover, superheat degree and condensation temperature presents slight variation over the whole operating range. At rated condition, the ORC system achieves maximum net power output per unit heat transfer area of 0.74kW/m^2. Furthermore, the ratio of maximum effective heat transfer area to the actual area of the evaporator is 69%, which has great influence on the performance of the ORC system.

1. INTRODUCTION

Over the past few years, the energy consumption has kept increasing with the development of industrialization process in China. Thereinto, internal combustion engines (ICEs) have consumed about 60% of overall oil consumption. Due to the low thermal efficiency of the ICEs, only about one-third of total fuel combustion energy becomes power output. The exhaust waste heat recovery of ICEs presents a potential for converting waste heat into electrical energy to improve the engine thermal efficiency and reduce emissions.

As a solution to the low grade waste heat recovery, Organic Rankine cycle (ORC) has been applied widely because of its high efficiency, low cost and simple structure. Zhai *et al*. (2014) showed that the ORC system is one of the most effective methods for recovering energy from low grade heat sources. Walraven *et al*. (2015) observed that the discount rate, electricity price, brine inlet temperature and annual electricity price evolution have a strong influence on the configuration and efficiency of the ORC and on the economics of the project. Heberle *et al*. (2012) showed that the use of mixtures as working fluids leads to an efficiency increase compared to pure fluids. Li *et al*. (2014) showed that there exists a possible relationship between the critical temperature of working fluids and the economical performance of the system.

Currently, much attention has also been paid to waste heat recovery for the ICEs based on ORC system. Wang *et al*. (2014) indicated that the exhaust energy recovery system serve more applicable on the heavy-duty diesel engine. Teng *et al*. (2006) demonstrated that with the hybrid power system of the diesel engine and the Rankine engine operated with waste heat, substantial enhancement in engine power and improvement in fuel economy can be achieved. Vaja *et al*. (2010) showed that a 12% increase in the overall efficiency can be achieved compared with the engine with no bottoming.

Based on the aforementioned analysis, the ORC system has already been extensively applied and investigated in low grade heat sources, especially in ICEs waste heat recovery. In this paper, the thermodynamic model of the ORC system is established based on the first and second laws of thermodynamics. The optimal operating parameters of the ORC system are investigated under diesel engine various operating conditions with evaporation pressure, superheat degree, and condensation temperature of the working fluid as optimization parameters for the maximum net power output per unit heat transfer area (POPA) and the minimum exergy destruction rate (EDR) using genetic algorithm (GA).

2. SYSTEM DESCRIPTION

The schematic and *T-s* diagram of the ORC system for recovering engine exhaust waste heat are shown in Figures 1 and 2. The liquid working fluid is pressurized to the evaporator by the pump. Then the subcooled liquid working fluid absorbs heat from the exhaust gas in the evaporator and turns into saturated or superheated vapor. After that, the vapor flows into the expander to do work and drives the generator to produce electricity. Finally, the superheated vapor exported from the expander condenses into saturated liquid state in the condenser. Subsequently, the liquid working fluid is pressurized by the pump to begin the new cycle again.

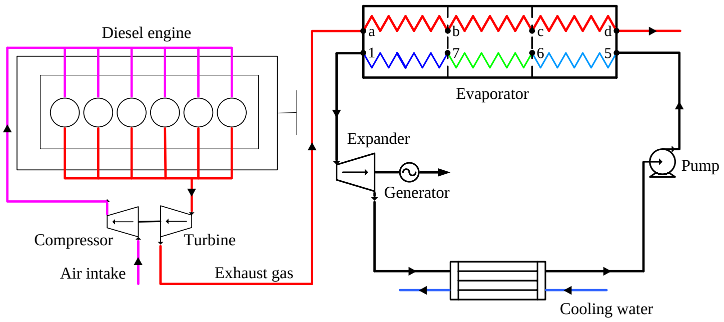

Figure 1: Schematic diagram of the ORC system

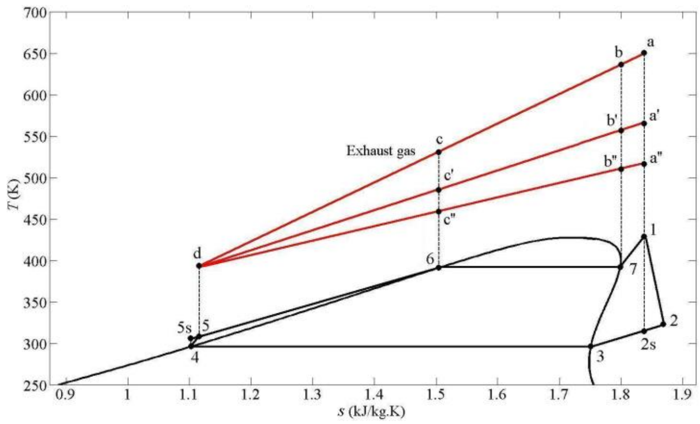

Figure 2: *T-s* diagram of the ORC system

The optimal working fluid should not only ensure higher waste heat recovery efficiency for the ORC system, but also meet the requirement of environmental protection, safety, and economy. Wang *et al*. (2011) compared nine different pure organic working fluids based on their thermodynamic performances. R245fa is the most suitable working fluid for engine waste heat recovery application.

Therefore, R245fa is selected as the working fluid in this study.

3. MATHEMATICAL MODEL

3.1 Thermodynamic model

Process 1-2 (Expander):
The power output and the exergy destruction rate of the expander are given by:

$$\dot{W}_{\mathrm{exp}} = \dot{m}_{\mathrm{wf}}(h_1 - h_2) \tag{1}$$

$$\dot{I}_{\mathrm{exp}} = T_0 \dot{m}_{\mathrm{wf}}(s_2 - s_1) \tag{2}$$

The isentropic efficiency of expander is given by:

$$\eta_{\mathrm{exp}} = \frac{h_1 - h_2}{h_1 - h_{2\mathrm{s}}} \tag{3}$$

Process 2-4 (Condenser):
The heat transfer rate and the exergy destruction rate of the condenser can be expressed as:

$$\dot{Q}_{\mathrm{con}} = \dot{m}_{\mathrm{wf}}(h_2 - h_4) \tag{4}$$

$$\dot{I}_{\mathrm{con}} = \dot{m}_{\mathrm{wf}}[(h_2 - h_4) + T_0(s_2 - s_4)] \tag{5}$$

Process 4-5 (Pump):
The power consumed and the exergy destruction rate of the pump can be determined as:

$$\dot{W}_{\mathrm{p}} = \dot{m}_{\mathrm{wf}}(h_5 - h_4) \tag{6}$$

$$\dot{I}_{\mathrm{p}} = T_0 \dot{m}_{\mathrm{wf}}(s_5 - s_4) \tag{7}$$

The isentropic efficiency of the pump is given by:

$$\eta_{\mathrm{p}} = \frac{h_{5\mathrm{s}} - h_4}{h_5 - h_4} \tag{8}$$

Process 5-1 (Evaporator):
The heat transfer rate and the exergy destruction rate of the evaporator can be expressed as:

$$\dot{Q}_{\mathrm{eva}} = \dot{m}_{\mathrm{exh}}(h_{\mathrm{a}} - h_{\mathrm{d}}) = \dot{m}_{\mathrm{wf}}(h_1 - h_5) \tag{9}$$

$$\dot{I}_{\mathrm{eva}} = \dot{m}_{\mathrm{exh}}[(h_{\mathrm{a}} - h_{\mathrm{d}}) + T_0(s_{\mathrm{a}} - s_{\mathrm{d}})] - \dot{m}_{\mathrm{wf}}[(h_1 - h_5) + T_0(s_1 - s_5)] \tag{10}$$

For each heat transfer area in the evaporator, the heat transfer rate between the exhaust and the working fluid can be calculated using the following equations:

$$\dot{Q}_{71} = \dot{m}_{\mathrm{exh}}(h_{\mathrm{a}} - h_{\mathrm{b}}) = \dot{m}_{\mathrm{wf}}(h_1 - h_7) \tag{11}$$

$$\dot{Q}_{67} = \dot{m}_{\mathrm{exh}}(h_{\mathrm{b}} - h_{\mathrm{c}}) = \dot{m}_{\mathrm{wf}}(h_7 - h_6) \tag{12}$$

$$\dot{Q}_{56} = \dot{m}_{\mathrm{exh}}(h_{\mathrm{c}} - h_{\mathrm{d}}) = \dot{m}_{\mathrm{wf}}(h_6 - h_5) \tag{13}$$

To investigate the heat transfer performance of the evaporator, the ratio of effective heat transfer area to actual area is defined as:

$$\eta_{\mathrm{ht}} = \frac{A_{\mathrm{eff}}}{A_{\mathrm{act}}} \tag{14}$$

The aim of this paper is to conduct the parametric optimization and evaluate the performance of a finned-tube evaporator used in the ORC system. Therefore, the isentropic efficiencies of the pump and the expander are set to constant values. Therein, several assumptions for the thermodynamic model

are listed as follows:
(1) The ORC system operates under steady state conditions.
(2) There are no pressure drops in the pipes and the components.
(3) The heat losses in each component are also neglected.
(4) The isentropic efficiencies of the pump and the expander are set to 0.65 and 0.7 respectively.
(5) The ambient temperature is set to 291.15 K.

3.2 Evaporator model

In this study, we select a fin-and-tube evaporator used in our lab for thermodynamic analysis. The schematic and the geometric dimensions of the fin-and-tube evaporator are shown in Figure 3 and Table 1, respectively. The engine exhaust gas flows outside the tube, and the working fluid flows inside the tube. The evaporator model refers to Zhang *et al*. (2013).

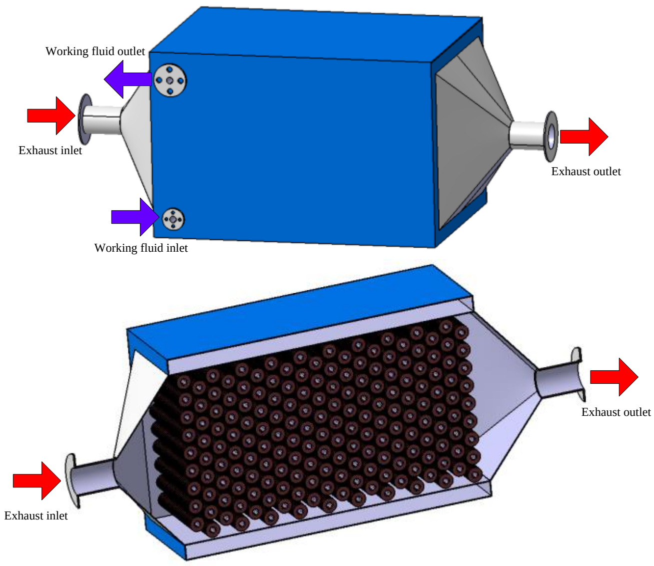

Figure 3: Schematic of the fin-and-tube evaporator

Table 1: Geometric dimensions of the fin-and-tube evaporator

Items	Parameters	Units
Number of tubes in each row	9	-
Number of tube rows	20	-
Tube outer diameter	25	mm
Tube inner diameter	20	mm
Tube pitch	60	mm
Row pitch	100	mm
Fin height	12	mm
Fin width	1	mm
Rib effect coefficient	3	-
Tube row alignment	Staggered type	-
Tube material	Stainless steel316L	-
Fin material	Stainless steel316L	-
Inner heat transfer area	9	m^2
Tube length	8.8	m

4. OPTIMIZATION METHODOLOGY

4.1 Description of genetic algorithm

The diesel engine operates under various operating conditions for the practical applications. Therefore, the coordinated variation of each operating parameters in the ORC system is indispensable to achieve the optimal performance. Genetic algorithm which was invented by professor Holland (1975) is adaptive heuristic search algorithm based on the evolutionary ideas of natural selection and genetics.

In the present study, the POPA and the EDR are evaluated as objective functions with evaporation pressure, superheat degree and condensation temperature of the working fluid as optimization variables. Therefore the optimization of the ORC system is performed by using GA in MATLAB. The multi-objective optimization function can be described as:

$$\max(\mathrm{POPA}) = f(P_7, T_1, T_4) \tag{15}$$

$$\min(\mathrm{EDR}) = f(P_7, T_1, T_4) \tag{16}$$

The lower and upper bounds of decision variables and parameter setting of GA are listed in Tables 2 and 3.

Table 2: Lower and upper bounds of decision variables

Decision variables	Lower bound	Upper bound
Evaporation pressure (MPa)	0.8	3
Superheat degree (K)	0	20
Condensation temperature (K)	298.15	353.15

Table 3: Parameter setting of genetic algorithm

Parameters	Value
Population size	150
Selection process	Tournament
Crossover fraction	0.8
Mutation function	Adaptive feasible
Stop generations	600

5. ENGINE WASTE HEAT EVALUATION

The engine used as the topping cycle in this study is a six-cylinder in-line diesel engine. The main technical performance parameters of the diesel engine are listed in Table 4.

Table 4: The main technical parameters of the diesel engine

Items	Parameters	Units
Rated power	247	kW
Maximum torque	1600	N.m
Displacement	11.596	L
Cylinder number	6	
Air intake type	Turbocharged and Intercooled	
Fuel injection system	High pressure common rail	
Speed at maximum torque	1400	r/min
Stroke and cylinder bore	155×126	mm
Compression ratio	17.1	

Figure 4 shows the performance maps of the diesel engine over the whole operating range. The variation of BSFC (brake specific fuel consumption) is illustrated in Figure 4(a), which indicates that the BSFC is mainly influenced by engine torque. When the diesel engine is operating under the high load conditions, it has the lower BSFC within the range of 196 g/(kW.h) to 205 g/(kW.h). Moreover, at the engine rated condition, the power output of the engine reaches a maximum value of 247 kW. Figure 4(b) shows the variation of the exhaust temperature with the engine speed and load. It can be seen that the exhaust temperature varies slightly with engine speed, but increases rapidly with engine load. When diesel engine is operating under the medium-low speed and medium-high load conditions, the exhaust temperature is relative high and can reach its maximum value of 667 K. The exhaust mass flow rate of the diesel engine, which is the sum of fuel consumption and intake air mass flow rate per unit time, is shown in Figure 4(c). It is obvious that the exhaust mass flow rate increases with engine speed and engine load. At rated condition, the exhaust mass flow rate reaches a maximum of 0.36 kg/s. Figure 4(d) shows the available exhaust energy of the diesel engine over the whole operating range. Note that the available exhaust energy is mainly affected by exhaust mass flow rate and exhaust temperature. Therefore, the available exhaust energy shows the same variation tendency compared with Figures 4(b) and 4(c). Under the medium-high speed and medium-high load conditions, the available exhaust energy ranges from 47 kW to 103 kW. At rated condition, the maximum available exhaust energy of the diesel engine is 103 kW.

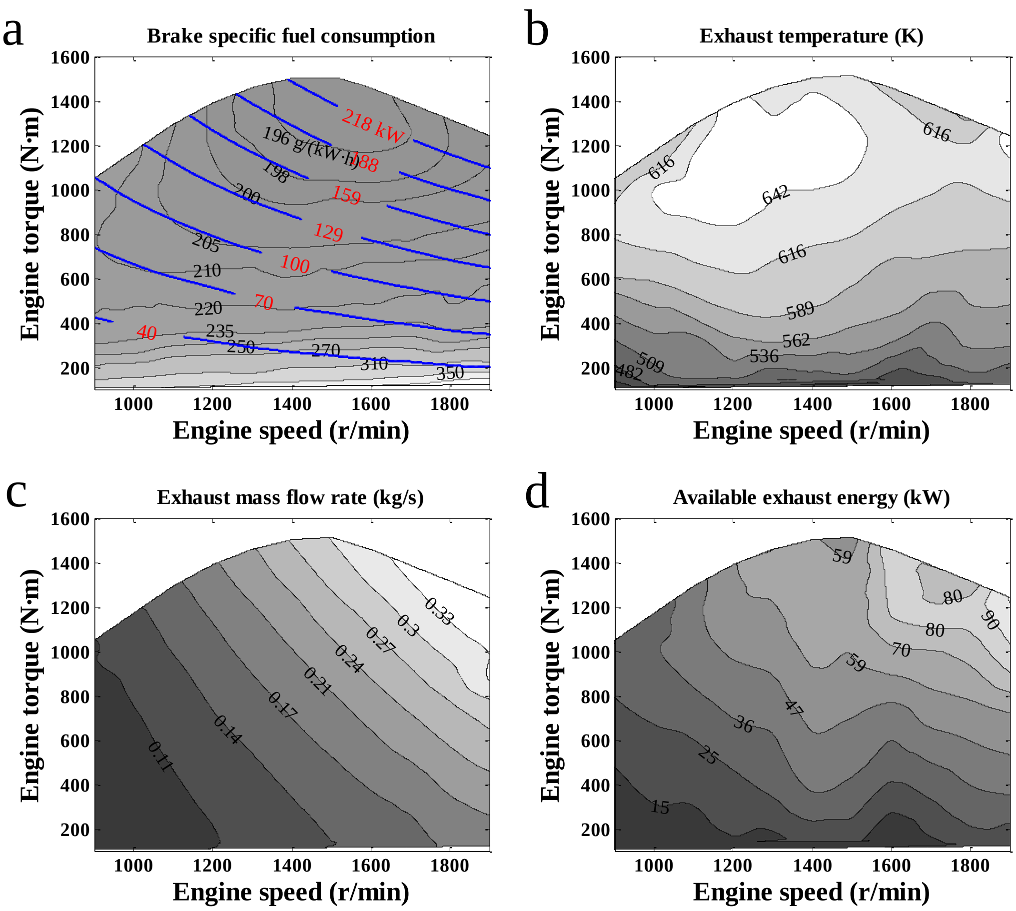

Figure 4: Performance maps of the diesel engine

6. RESULTS AND DISCUSSION

Based on the thermodynamic model established of the ORC system, the optimization using GA for the exhaust waste heat recovery is performed over the whole operating range of the diesel engine. The aim of the optimization process is to achieve the maximum POPA and the minimum EDR.

Figure 5 shows the optimization results of the two objective functions over the whole operating range of the diesel engine. The maximum POPA of the ORC system is presented in Figure 5(a). It can be observed that the maximum POPA is influenced by engine operating conditions significantly, namely available exhaust energy. From Figures 4(d) and 5(a), the maximum POPA increase with the available exhaust energy. This can be explained since the more exhaust energy will lead to an increase in the quantity of evaporated working fluid, which resulted in the increase of the net power output. Over the whole operating range, the maximum POPA of the ORC system ranges from 0.06 kW/m^2 to 0.74 kW/m^2. At rated condition, the maximum POPA of the ORC system is 0.74 kW/m^2. Figure 5(b) shows the variation of the minimum EDR with the engine speed and load. Similarly to the POPA, engine's operating conditions have a great impact on the EDR. When the diesel engine operates under the medium-low speed and medium-low load conditions, the ORC system can provide lower EDR, whose values range from 0.61 kW to 10.75 kW. It also can be seen that when the diesel engine operates under high speed and high load conditions, the ORC system has the higher EDR. This is due to the exhaust temperature increases with the engine speed and load, which resulted in higher temperature difference between the exhaust gas and the working fluid.

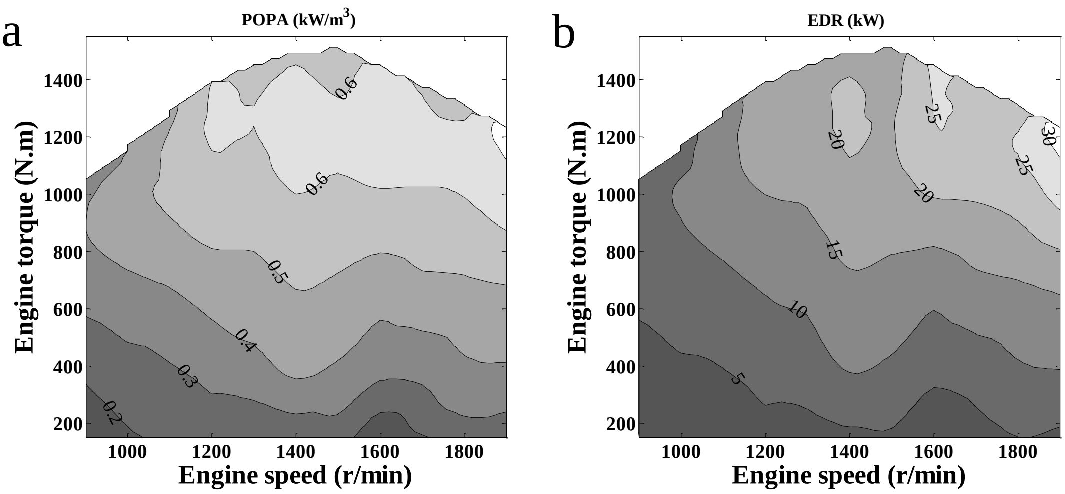

Figure 5: Optimization results of the two objective functions

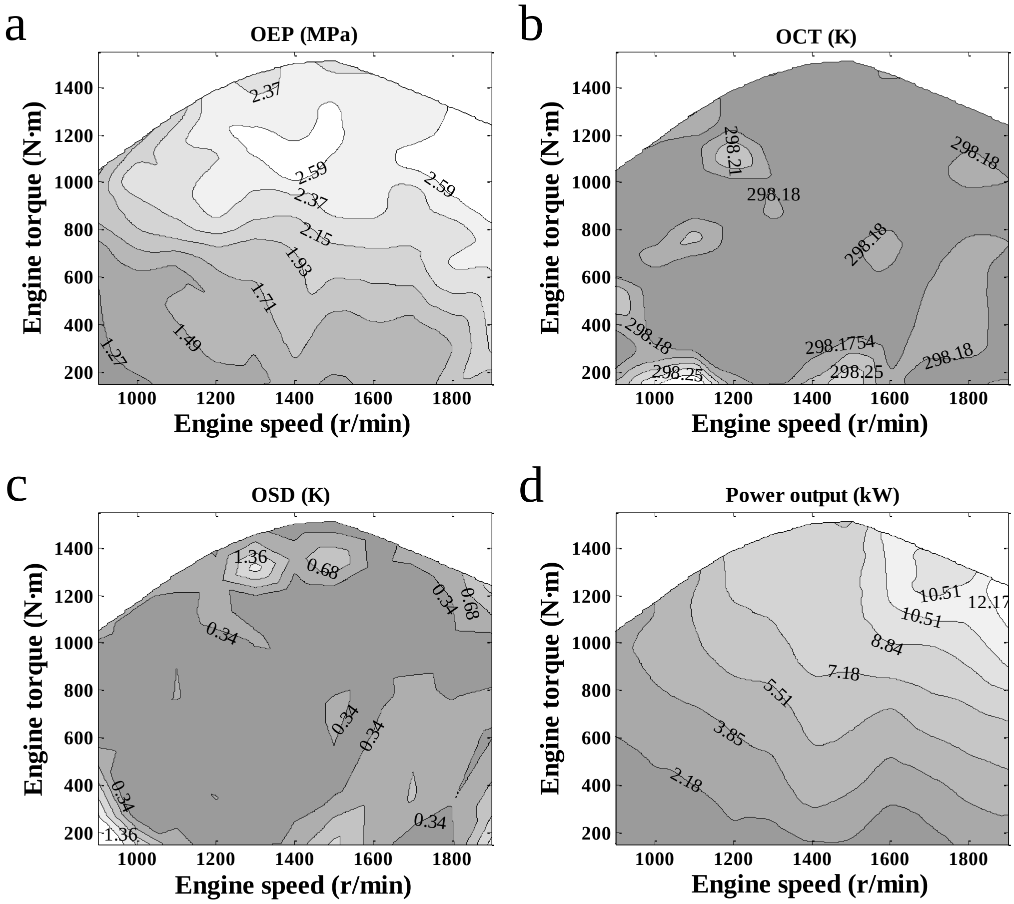

Figure 6: Optimization results of operating parameters and power output for the ORC system

Figure 6 shows the optimization results of operating parameters for the ORC system, including OEP (optimal evaporation pressure), OCT (optimal condensation temperature), and OSD (optimal superheat degree). In addition, the variation of the ORC system power output with the engine's operating conditions is presented in Figure 6(d). Figure 6(a) demonstrates that the overall trend about the OEP increases with the increment of engine speed and engine torque. Over the whole operating range, the OEP for the working fluid is in the range of 1.09 MPa to 2.81 MPa. Therefore, it is necessary to select the OEP for each operating condition of the diesel engine. As can be seen in Figure 6(b), the OCT has a small variation in its range, which is nearly kept constant at 298.15 K. As we all known, higher evaporation pressure and lower condensation temperature will lead to an increase in the thermal efficiency of the ORC system. Therefore, the optimization results of the OEP and the OCT are consistent with the analysis based on the first and second laws of thermodynamics. Figure 6(c) shows the variation of OSD over the engine whole operating range. It can be seen that the OSD is in the range of 0 K to 2.13 K. Under the most operating conditions of the diesel engine, the OSD values are in the range between 0 K to 1 K. That is to say, the working fluid does not need to be superheated, which has the similar results for dry organic fluids with Mago *et al.* (2008). In the practical applications of the ORC system, such small OSD cannot be met perfectly due to the limitation of technology problems. From Figure 6(d), it can be concluded that the net power output exhibits the same variation tendency with available exhaust energy presented in Figure 4(d). The net power output increases with increasing available exhaust energy. Over the whole operating range of the diesel engine, the net power output of the ORC system is in the range of 0.32 kW to 13.84 kW. At the engine rated condition, the net power output is 13.84 kW.

Figure 7(a) shows the thermal efficiency of the ORC system over the whole operating range of the diesel engine. It can be observed that the thermal efficiency of the ORC system is mainly affected by the engine torque. When the diesel engine operates at the high load regions, the ORC system presents higher thermal efficiency. This is due to the increase in the net power output obviously at these regions. At engine rated condition, the thermal efficiency of the ORC system reaches the maximum of 13.33%. The POIR (power output increasing rate) is the ratio of the net power output of the ORC system to the engine power output, which is used to evaluate the performance improvement of the diesel engine. The variation of the POIR over the engine whole operating range is shown in Figure 7 (b). It can be observed that the ORC system generates higher POIR at the low torque regions of the diesel engine. That is to say, when the diesel engine operates at the low torque regions, the ORC system can improve the power performance of the diesel engine notably. At the low torque regions of the diesel engine, the POIR is in the range of 3.23% to 9.41%. When the diesel engine operates at the medium-high torque regions, the POIR changes gently.

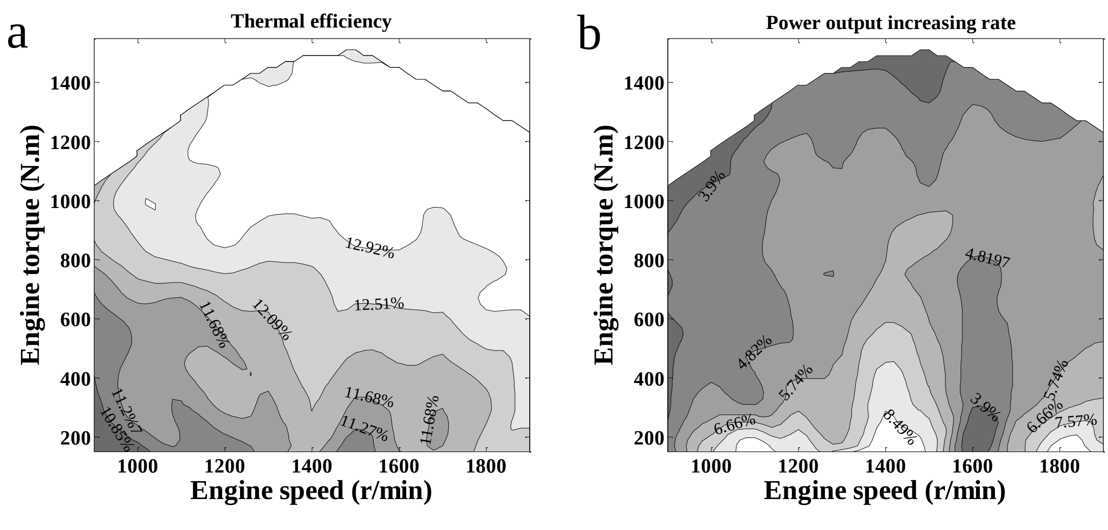

Figure 7: Thermal efficiency of the ORC system and the POIR compared with the diesel engine

Figure 8 shows the ratio of effective heat transfer area to actual area of the evaporator over the whole operating range of the diesel engine. It can be observed that when the diesel engine operates under the medium-low speed or medium-high torque conditions, the ratio of effective heat transfer area to actual area of the evaporator increases with the engine speed and torque. In addition, when the diesel engine operates under the medium-high speed and medium-low torque conditions, the ratio of effective heat transfer area to actual area of the evaporator is mainly affected by the engine torque. That is to say, the operating conditions of the diesel engine have a great effect on the performance of the evaporator. The performances of the evaporator, including heat transfer rate, heat transfer temperature difference, heat transfer coefficient, and heat transfer surface area will vary with the operating conditions of the diesel engine. Over the whole operating range of the diesel engine, the ratio of effective heat transfer area to actual area of the evaporator is in the range of 17.78% to 69.19%.

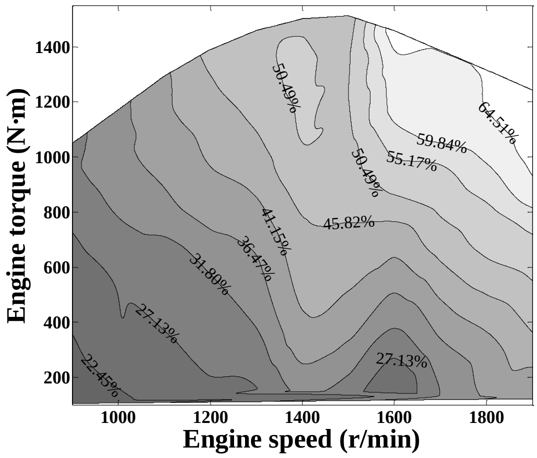

Figure 8: The ratio of effective heat transfer area to actual area of the evaporator

7. CONCLUSIONS

In this study, the parametric optimization and performance analysis of the ORC system using genetic algorithm for recovering exhaust waste heat of a diesel engine are conducted. Based on the first and second laws of thermodynamics, the effects of three key parameters, including evaporation pressure, superheat degree and condensation temperature on the system performances are conducted with net power output per unit heat transfer area and exergy destruction rate as objective functions. The main conclusions can be drawn as follows:

- Over the whole operating range of the diesel engine, the maximum POPA and the net power output are influenced by the engine operating conditions significantly. At rated condition, the maximum POPA and the net power output of the ORC system are 0.74 kW/m^2 and 13.84 kW, respectively.
- The overall trend about the OEP increases with the increment of engine speed and engine torque. Over the whole operating range, the OEP for the working fluid is in the range of 1.09 MPa to 2.81 MPa. The OCT has a small variation in its range, which is nearly kept constant at 298.15 K. The OSD is in the range of 0 K to 2.13 K. Under the most operating conditions of the diesel engine, the OSD values are in the range between 0 K to 1 K.
- For a selected fin-and-tube evaporator, the preheated zone has the maximum heat transfer rate and surface area. Over the whole operating range of the diesel engine, the ratio of effective heat transfer area to actual area of the evaporator is in the range of 17.78% to 69.19%. Considering the OSD almost equals 0, therefore the heat transfer rate and surface area in the superheated zone can almost be neglected.

NOMENCLATURE

$\dot{W}$	power	(kW)
$\dot{Q}$	heat transfer rate	(kW)
$\dot{m}$	mass flow rate	(kg/s)
h	specific enthalpy	(kJ/kg)
s	specific entropy	(kJ/kg.K)
$\dot{I}$	exergy destruction rate	(kW)
T	temperature	(K)
P	pressure	(MPa)
A	area	(m^2)
Greek letters		
η	efficiency	
Subscript		
0	reference state	
1-7,2s,5s	state points in the cycle	
exp	expander	
wf	working fluid	
con	condenser	
p	pump	
eva	evaporator	
exh	exhaust	
eff	effective	
act	actual	

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ACKNOWLEDGEMENT

This work was sponsored by the Scientific Research Key Program of Beijing Municipal Commission of Education (Grant No. KZ201410005003), the Beijing Natural Science Foundation Program (Grant No. 3152005), the National Natural Science Foundation of China (Grant No. 51376011), and the Key Project of Thirteenth Scientific Research Foundation for Graduate Students in Beijing University of Technology (Grant No. ykj–2014–10708).