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Analysis of a 1 kW Organic Rankine cycle using scroll expander for engine coolant and exhaust heat recovery

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Abstract The development of engine waste heat recovery technologies attracts ever increasing interests due to the raising strict policy requirements and environmental concerns. This paper presents the study of engine coolant and exhaust heat recovery using Organic Rankine cycle. Eight working fluids have been selected to evaluate and compare the performance of the integrated waste heat recovery system. Rather than conventional engine ORC system mainly focusing on the utilisation of exhaust energy, this work proposed to fully use the engine coolant energy by changing the designed parameters of the ORC system. The case study selected a small engine as the heat source to drive the ORC system using scroll expander for power production. The evaluation results suggest under the engine rated condition, the solution to fully recover the engine coolant energy can achieve higher power generation performance than that of conventional engine ORC system. Results suggest adding a recuperator into the ORC system can potentially improve the system performance when the working fluids are dry type and the overall dumped heat demand of the system can be reduced by 12 % under the optimal conditions. When the ORC evaporating and condensing temperature are respectively set at 85 °C and 30 °C, the integrated engine waste heat recovery system can improve the overall system efficiency by 9.3 % with R600, R600a or n-Pentane as the working fluid.

Keywords: Organic Rankine cycle, scroll expander, coolant and exhaust recovery, Internal Combustion Engine

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1. Introduction

The increasing of fuel prices and strict requirements on carbon dioxide emissions limits are promoting the development to increase the engine thermal efficiency beyond the limit of in-cylinder techniques [1]. As one of most promising heat driven power generation technologies, Organic Rankine Cycle (ORC) has been extensively studied for engine exhaust heat recovery because the thermodynamic cycle works well for the medium grade energy of the exhaust [2].

The market available ORC system with the power ranges of 0.2 to 2 MWe under the cost around 1 and 4×103€/kWe, and lower powers are in pre-commercial status because of the relatively long payback period using small scale ORC system [3]. The designed evaporation and condensation temperature determines the overall efficiency of a typical ORC system. The higher of temperature difference between evaporation and condensation, the higher overall ORC efficiency can be obtained. Therefore the engine coolant energy is commonly recognized as a heat source that is not worthy recovering because the relatively low temperature ranges of the coolant, which is about 80-100 °C and can only provide 40-60 °C temperature difference for on road vehicle application [4]. However, the engine coolant energy contains about 30 % of the fuel energy and has huge potential to be utilised for engine ORC heat recovery system in order to improve the ORC efficiency and reduce the payback period of the

system [3, 5]. The coolant energy is commonly used to preheat the ORC fluid as reported by Yu et al [6]. Results suggest only 9.5% coolant heat has been recovered under engine conditions from high to low load, which means the majority coolant energy cannot be reused and is dumped to the environment [6]. Adopting dual-loop ORC system could be one of the solutions to achieve high overall system efficiency as reported by Wang et al [7, 8]. Nevertheless, the proposed dual-loop ORC system requires two set of ORC system components and advanced controlling strategies to balance different heat sources. A cost effective using relatively simple ORC system is therefore important to promote the development of engine waste heat recovery system.

On the other hand, the selection of working fluid plays a key role in ORC performance [9-12]. Based on the slope of vapor saturation curves in T - s diagram, ORC fluids can be classified into three group: dry, isentropic and wet types [13]. The dry and isentropic types of working fluids are recommended to be used for ORC system, because the working conditions of the fluid for expansion process can be designed within the vapor phase, which can protect the expansion machine to be damaged by the liquid drop of the fluid using wet type working fluids [10-12]. Saleh et al. [10] conduct performance study of a ORC system using 31 pure working fluids and the results pointed out the n -butane can achieve the maximum thermal efficiency at 0.13 under 120 °C heat source temperature. The investigation on a small-scale engine ORC waste heat recovery system using six different working fluids has been reported by Lu et al. [14]. Results suggest under the engine rated condition, the integrated ORC system can potentially improve the overall energy efficiency by 11.2% and the Brake Specific Fuel Consumption (BSFC) can be improved by 10 % [14]. Wang et al. [9] report a study to compare the performance of 10

kW net power output ORC system using different working fluids for engine exhaust heat recovery. Results indicate R11, R141b, R113 and R123 manifest slightly higher thermodynamic performances than other working fluids [9].

In this study, a small scale ORC system recovering engine coolant and exhaust energy has been reported and studied. The ORC system performance using coolant energy as preheat source and adopting coolant energy during ORC fluid evaporating process have been compared. A scroll type expander has been selected as the expansion machine in the small scale system because of the scroll expander has the advantages of high reliability, relatively high isentropic efficiency and broad availability [15-18]. Eight working fluids have been selected to compare the system performance including net power, thermal efficiency of the ORC, rotational speed of the scroll device and required heat dump loads of the system.

2. Description of engine coolant and exhaust ORC system

The ORC system contains a pump, three heat exchangers, a scroll expander and a condenser as illustrated in Fig. 1. The heat exchanger for the recovery of coolant energy is named as Heater 1 and the engine exhaust energy is recovered by Heater 2, which located at the exhaust of the engine. When the recuperator is used the unused heat at exist of the expander is first recovered by the recuperator to preheat the working fluid pumped from the pump and then dumped to the environment from the condenser. The working conditions of the ORC fluid can be defined as:

- 1 $\rightarrow$ 2 isentropic compression process in the pump;

- $2 \rightarrow 3 \rightarrow 4$ isobaric process during the heating process of the working fluid recovering coolant energy
- $4 \rightarrow 5 \rightarrow 6$ isobaric process for the recovery of exhaust energy
- $6 \rightarrow 7$ isentropic expansion process in the scroll expander
- $7 \rightarrow 8 \rightarrow 1$ isobaric process in the condenser.
- When the recuperator is operated in the system the heat transfer processes of two sides from $2 \rightarrow 2'$ and $7 \rightarrow 7'$ are both recognised as isobaric process. Detailed calculation methods are introduced in the following section.

Fig. 1 Schematic diagram of engine coolant and exhaust recovery system [19]

3. Methodologies

The engine selected in this study is a small scale diesel fuelled Internal Combustion Engine with the model number YTG-6.5S. The specification and evaluated engine operating condition are listed in Table 1. The selected ICE is a one cylinder engine with 0.638 L engine displacement and achieved the rated power 8.8 kW at 2400 RPM [20].

The isentropic efficiency of the pump is defined as Eq. (1), where h_{2_s} is the specific enthalpy at exit of the pump after isentropic expansion and the pump efficiency in this study is set at 0.4 [21].

$$\eta_{pump} = (h_{2_s} - h_1) / (h_2 - h_1) \quad (1)$$

The work provided to the pump can therefore be calculated by the following equation, where $\dot{m}_{ORC}$ is the mass flow rate of the ORC working fluid.

$$\dot{W}_{pump} = \dot{m}_{ORC} \cdot (h_2 - h_1) \quad (2)$$

The heat exchanger efficiency of the Heater 1, Heater 2 and recuperator are all set at 0.8. The heat provided to the Heater 1 and Heater 2 from the coolant and exhaust energy are respectively calculated by Eq. (3) and (4). As stated in the introduction, the engine coolant energy cannot be fully recovered when the coolant heat is used to preheat the working fluid.

Table 1 Engine specification and operating conditions under engine rated power

Characteristic	Value
Engine displacement (L)	0.638
Cylinder bore (mm)	92
Cylinder stroke (mm)	96
Compression ratio	17.7
Max torque (N.m) at 1800 RPM	43.35
Max Power (kW) at 2400 RPM	8.818
Coolant heat (kW) at rated power	4.54
Exhaust heat (kW) at rated power	4.67
Exhaust temperature (°C) at rated power	587
Fuel consumption (kg/h) at rated power	1.975

Therefore the recovered coolant heat for engine ORC

system can be represented as $\dot{Q}_{coolant_rec}$.

$$\dot{Q}_{Heater1} = \dot{m}_{ORC} \cdot (h_4 - h_2) = \dot{Q}_{coolant_rec} \cdot \eta_{Heater1} \quad (3)$$

$$\dot{Q}_{Heater2} = \dot{m}_{ORC} \cdot (h_6 - h_4) = \dot{Q}_{exhaust} \cdot \eta_{Heater2} \quad (4)$$

When the recuperator is in operation, the heat transfer process insider the recuperator can be calculated by Eq. (5), where η_r is the efficiency of the recuperator.

$$\dot{Q}_r = \dot{m}_{ORC} \cdot (h_7 - h_{7'}) \cdot \eta_r = \dot{m}_{ORC} \cdot (h_2 - h_2') \quad (5)$$

As a volumetric expansion machine, the geometry of the equipment mainly determines the suitable operating region of the expander. The selected 1 kW scroll expander has been geometrically studied to obtain the key parameters as shown in Fig. 2. Based on the calculated results from geometric scroll expander model, the exhaust and suction volume of the scroll expander is 41.38 cm³ and 11.82 cm³, respectively. The relationships between ORC fluid mass flowrate and expander rotational speed can therefore be calculated by Eq. (6), where V_{sv} is the swept volume of the scroll expander, v_{su} is the specific volume at the inlet of the expander and the expander rotational speed is represented as N .

$$\dot{m}_{ORC} = \frac{N}{60} \cdot \frac{V_{sv}}{v_{su}} \quad (6)$$

Fig. 2 The geometric and fixed scroll [14]

The power output from the expander can be defined as Eq. (7), where h_{su} is the specific enthalpy at the inlet of the expander, h_{d-s} is the designed exhaust specific enthalpy of the scroll expander after isentropic expansion process P_{d-s} is the designed exhaust pressure, and v_d is the designed specific volume. The expansion efficiency η_{exp} of the expander is assumed at 0.8 and the isentropic expander efficiency is written as η_{iso_exp} .

$$\begin{aligned}\dot{W}_{exp} &= \dot{m}_{ORC} \cdot [(h_{su} - h_{d-s}) + (P_{d-s} - P_{ex}) \cdot v_d] \cdot \eta_{exp} \\ &= \dot{m}_{ORC} \cdot (h_6 - h_7) \cdot \eta_{iso_exp}\end{aligned}\quad (7)$$

The thermal efficiency of the ORC system can therefore be defined as the following equation.

$$\eta_{ORC} = (\dot{W}_{exp} - \dot{W}_{pump}) / (\dot{Q}_{coolant_rec} + \dot{Q}_{exhaust}) \quad (8)$$

The ORC system dumps heat from the condenser in order to maintain the low pressure condition of the expander and provide liquid phase working fluid to the pump. The dumped heat for vehicle application is commonly through the engine radiator system. Therefore the evaluation of required dumped heat using ORC system for engine application is important. The dumped heat using the engine integrated ORC waste heat recovery can be defined and calculated by Eq. (9).

$$\dot{Q}_{dump} = \dot{Q}_{con} + \dot{Q}_{coolant} - \dot{Q}_{coolant_rec} \quad (9)$$

Table3 Properties of the selected ORC working fluids [11, 12]

	M (g/mol)	Tc (K)	Pc (MPa)	Vapor C _p (J/kg K)	Latent heat L (kJ/kg)	Boiling point (K)	Type	GWP (100yr)	ODP	AL (yr)
R245fa	134.05	427.20	3.64	980.90	177.08	288.4	dry	1030	0	7.6
Toluene	92.14	591.75	4.13	1223.90	399.52	383.75	dry	n.a	n.a	n.a
n-Pentane	72.15	469.15	3.36	1824.12	349.00	309.25	dry	0.1	0	n.a
R134a	102.03	374.21	4.06	1211.51	155.42	247.05	isentropic	1430	0	14
R152a	66.05	386.41	4.52	1456.02	249.67	249.15	wet	124	0	1.4
R600	58.12	425.13	3.80	1965.59	336.82	272.65	dry	20	0	0.018
R600a	58.12	409.81	3.63	1981.42	303.44	261.45	dry	20	0	0.019
R124	136.48	395.43	3.62	908.70	132.97	261.1	dry	609	0.022	5.8

p.s. GWP-Global Warming Potential, ODP- Ozone Depletion Potential, AL-Atmospheric Lifetime

The boundary conditions and standard parameters of the simulation model are summarised in Table 2.

Four key performance parameters including net power output, thermal efficiency, expander rotational speed and dumped heat requirement of the engine coolant and exhaust heat recovery ORC system under the engine rated power condition have been studied. Three different ORC operating methods for engine coolant and exhaust recovery are studied, which investigate the system performance using coolant as preheater source, reducing ORC evaporating temperature to fully recover coolant energy and adding recuperator into the full coolant energy recover ORC system. Eight working fluids have been selected and the performance of ORC system using different working fluids are analysed. The properties of selected working fluids can be found in Table 3.

Table 2 Boundary conditions and standard parameters of the ORC model

Parameter	Value
Fluid superheated temperature	5 °C
Efficiency of the heat exchangers and expander ($\eta_{Heater1}, \eta_{heater2}, \eta_r, \eta_{exp}$)	0.8
Pump isentropic efficiency η_{pump}	0.4
Maximum evaporating temperature provided from coolant energy	85 °C
Temperature of condenser T_8	30 °C

4. Results and discussion

4.1 Engine coolant and exhaust heat recovery ORC using coolant heat as preheat source

(a)

(b)

(c)

(d)

(e)

Fig. 3 Relationship between expander inlet temperature and (a) Power from expender, (b) Thermal efficiency, (c) Expander rotational speed, (d) Overall dumped heat (e) Working fluid mass flowrate from the ORC system

The engine coolant can only provide the temperature about 85 °C [4] and the superheated temperature in this study is set at 5 °C. Therefore the performance of the engine coolant and exhaust heat recovery ORC system can be evaluated by identifying the effects of expander inlet temperature. Results suggest the maximum power output from the ORC is around 0.77 kW, when R134a is used as working fluid and the expander inlet temperature is set around 100 °C. However, the thermal efficiency of the system using R134a is the worst among the eight selected working fluids. Under the same expander inlet temperature, the ORC system using working fluids R134a, R152a, R124 and R600a can achieve relatively higher output than that of other working fluids, which are mainly caused by the relative higher mass flow rate of the four working fluids as indicated in Fig. 3 (e). The maximum ORC thermal efficiency among the selected working fluids is about 0.086, when the ORC system adopts n-Pentane as the working fluid under the expander inlet temperature between 120 to 130 °C. The calculation results of scroll expander rotational speed as plotted in Fig. 3 (c) can provide as the reference for the ORC system under no load condition. The maximum range plotted in the Fig. 3(c) is 4500 RPM. Toluene is not suggested to be used as the

working fluid for this small scale ORC system because the scroll expander rotational speed is over 4500 RPM, which is not within the desirable operational condition of this type expansion machine. Moreover, the power output of the ORC system using Toluene is the lowest among other seven working fluids. The dumped thermal load of the system ranges from 8.5 to 8.7 kW. When the R600 is used as the working fluid, the minimum dumped heat of the ORC system is about 8.5 kW under 110 °C expander inlet temperature. The increase of required dumped thermal load with the increase of expander inlet temperature is mainly contributed by the coolant energy, because only limited coolant heat can be recovered with the increase of designed ORC evaporating temperature.

4.2 Engine coolant and exhaust heat recovery ORC to fully recover two heat sources

Fig. 4 Relationship between expander inlet temperature and (a) Power from expander, (b) Thermal efficiency, (c) Expander rotational speed, (d) Overall dumped heat (e) Working fluid mass flowrate from the ORC system

When the ORC system is operated to fully recover both heat sources from coolant and exhaust under engine rated condition, the working fluid mass flow rate is stable with the changes of expander inlet temperature as illustrated in Fig. 4 (e). The power output and thermal efficiency of the ORC system increase with the increase of designed expander inlet temperature as shown in Fig. 4 (a) and (b). Among the selected working fluids, the optimal thermal

efficiency can be achieved under the expander inlet temperature ranges from 85 to 90 °C. Under the designed expander inlet temperature at 90 °C, the power output from the ORC system using R152a can be as high as 0.8 kW, which is higher than the maximum power production from the engine coolant and exhaust ORC system using coolant energy as the preheat source as described in Section 4.1. Moreover, the dumped heat load of the proposed solution can be lower than the previous solution. Under the scroll expander inlet temperature at 85 °C, the ORC system using R600 rejects about 8.3 kW thermal loads to the environment, which is lower than the previous solution.

4.3 Engine coolant and exhaust heat recovery ORCR to fully recover to heat sources

Fig. 5 Relationship between expander inlet temperature and (a) Power from expander, (b) Thermal efficiency, (c) Expander rotational speed, (d) Overall dumped heat (e) Working fluid mass flowrate from the ORC system

The purpose of adding recuperator into ORC system is to recover the unused energy from the exit of the expander in order to improve the ORC overall performance and potentially reduce the dumped heat load from the system. Results suggest except system with R152a as working fluid adding recuperator into the ORC system can potentially improve the system performance. As a wet type working fluid, R152a is not suggested to be used under the designed

conditions. The reason is that after the expansion process, the temperature condition at the exit of the scroll expander is already lower than the condensation temperature. Results also point out when dry type working fluids are used in the ORC system, the dumped heat load can be effectively reduced when a recuperator is introduced. Among the selected eight working fluids, the maximum power output under the designed can be achieved about 0.82 kW, when R600, R600a and n-Pentane are used at 90 °C expander inlet temperature. It is about 9.3 % improvement, when the waste heat recovery system is integrated with the engine under the rated power conditions (8.81 kW). It can be observed that by adding recuperator into the ORC system, when n-Pentane is the working fluid, the system dumped heat load can be reduced from 8.3 kW to 7.3 kW at 85 °C expander inlet temperature, which is about 12 % reduction.

5. Conclusions

This paper reports the study of a small scale engine coolant and exhaust heat recovery system using ORC system. Eight working fluids have been selected in the analysis to investigate the system performance. The purpose of this study is to point out the potential of using simple ORC waste heat recovery to fully recover coolant energy rather than the conventional engine heat recovery solution mainly focusing on the utilisation of exhaust energy. The main conclusions of this study can be summarised as follows.

- When the engine coolant heat is used as preheat source, the maximum thermal efficiency of the ORC system is about 8.6 % using n-Pentane as working fluid at 120 °C expander inlet temperature. However, the maximum power output from the system is not obtained under the maximum ORC

thermal efficiency condition, because the engine coolant energy has not been effectively recovered. Toluene can achieve good overall thermal efficiency but the net power output from the system is the worst among the selected working fluids.

- Because the limitation of supplied temperature from the coolant energy, the thermal efficiency of engine coolant and exhaust heat recovery ORC system is lower than that system using coolant energy as preheater source. However, the power output from the system can be higher than that system using coolant for preheating. Moreover, the overall dumped heat demand of the engine can be slightly reduced when the ORC system is operated as fully recovering coolant energy. The conventional engine exhaust heat recovery ORC system can be easily modified into the proposed solution, which can produce similar power output with reduced dumped heat requirement.
- Adding a recuperator into the ORC system can recover the unused heat at the exit of the expander, when the working fluids are isentropic or dry types. When wet type ORC working fluid is operated in the system, the recuperator is generally not suggested to be used. When R600, R600a and n-Pentane are selected as the working fluids, the maximum power output from the ORC system can be as high as 0.82 kW at 90 °C designed expander inlet temperature. Under the optimal designed conditions, introducing the recuperator into the engine coolant and exhaust heat recovery ORC system can

effectively reduce the dumped heat load of the system by 12 %. Considering the performance of the selected engine under rated power condition, the integrated waste heat recovery system can potentially improve the overall system performance by 9.3%.

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