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Working fluid selection for a small-scale organic Rankine cycle recovering engine waste heat

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Abstract

This paper reports the design and evaluation of a 1 kWe Organic Rankine cycle using different working fluids for engine coolant and exhaust recovery from a 6.5 kW small ICE. Six working fluids have been selected to evaluate and compare the performance of the ORC system. The net power output, thermal efficiency, rotational speed of the scroll expander and condenser load of the ORC system have been studied. Results indicated R134a and R125a have better overall performance than other candidates when the designed inlet temperature of the expander is higher than 150 °C. The highest net power and thermal efficiency are respectively 1.2 kW and 13% when R125a is used as the working fluid. R600 and R245fa are desirable to be used when the optimal rotational speed of the scroll expander is about 3000 RPM. The proposed ORC engine coolant and exhaust waste heat recovery system has the advantages of simple system layout, low dumped heat load of condenser, high power output.

Keywords: Organic Rankine Cycle; scroll expander; coolant and exhaust recovery; Internal Combustion Engine

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1. Introduction

As one of the most promising heat recovery technologies to convert wasted energy from Internal Combustion Engine (ICE) into useful energy, Organic Rankine Cycle (ORC) attracts ever increasing attention from both academic and industry [1-3]. Velez et al. [4] pointed out the market available for ORC systems with the power ranges of 0.2 to 2 MWe under the cost of around 1 and 4×103 € /kWe. Most of the current research interest for engine waste heat recovery is mainly focusing on the recovery of engine exhaust energy because of the relatively high temperature from this energy source, which can potentially form a high thermal efficiency ORC system [2]. However, the engine coolant energy containing about 30% of the overall fuel energy, is difficult to recover and attracts ever increasing attention due to high demand for high energy efficiency and low emissions engines [5-7].

The working fluids in ORC are classified in three groups: dry, isentropic and wet types depending on the slope of the vapor saturation curve on the T-s diagram [8]. The selection of working fluid plays a key role in ORC performance [9-12]. Wang et al. [9] reported a study that compares 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].

A system performance study of a geothermal ORC system using 31 pure working fluids has been conducted by Saleh et al. [10]. The maximum thermal efficiency was found to be 0.13 with n-butane as the working fluid for 120 ^{o}C heat source temperature [10]. Shu et al. [13] proposed a new dual-loop ORC for engine waste heat recovery and investigated the system performance using first and second law analyses. Six working fluids were selected for the evaluation and comparison of system performance. The results pointed to R1235yf as the optimal working fluid under engine high operating load [13]. However, dual-loop ORC requires relatively complex control strategies and more system components, which will increase the capital cost of the system and lead to high payback period.

In this study, we report the design and evaluation of a small scale ORC system using different working fluids for engine coolant and exhaust recovery. A scroll expander was selected as the expansion machine in the small scale ORC system because the scroll expander has the advantage of high reliability, relatively high isentropic efficiency and broad availability [14-16]. Six 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 condenser loads. Moreover, the effects of using different working fluids on the overall engine performance have been conducted and discussed.

2. Description of the designed engine waste heat recovery ORC system

The ORC system designed to recovery the engine coolant and exhaust energy mainly includes a pump, two heat exchangers, a scroll expander, a condenser and a liquid receiver as shown in Fig. 1. The first heat exchanger (Heater 1) is used to recover the coolant energy from the engine. An exhaust heat exchanger has been located at the exhaust of the engine and an oil loop has been designed to transfer the exhaust energy to heat up the working fluid in the second heat exchanger (Heater 2). A liquid receiver has been located at the inlet of the pump in order to provide the pure liquid working fluid to the pump and prevent two phase working fluid entering the pump. A bypass line has been located between the inlet and outlet of the scroll expander, which will be used during the start-up of the system in order to provide full gas phase working fluid to the scroll device for power generation. The unused energy at the exit of the expander is dumped to the environment through the condenser, which cool down be the cooling tower. The working principles of the ORC system are defined as: isentropic compression process in the pump; isobaric process during the heating process of the working fluid recovering coolant energy; isobaric process for the recovery of exhaust energy; isentropic expansion process in the scroll expander; isobaric process in the condenser. Detailed calculation methods are introduced in the following section.

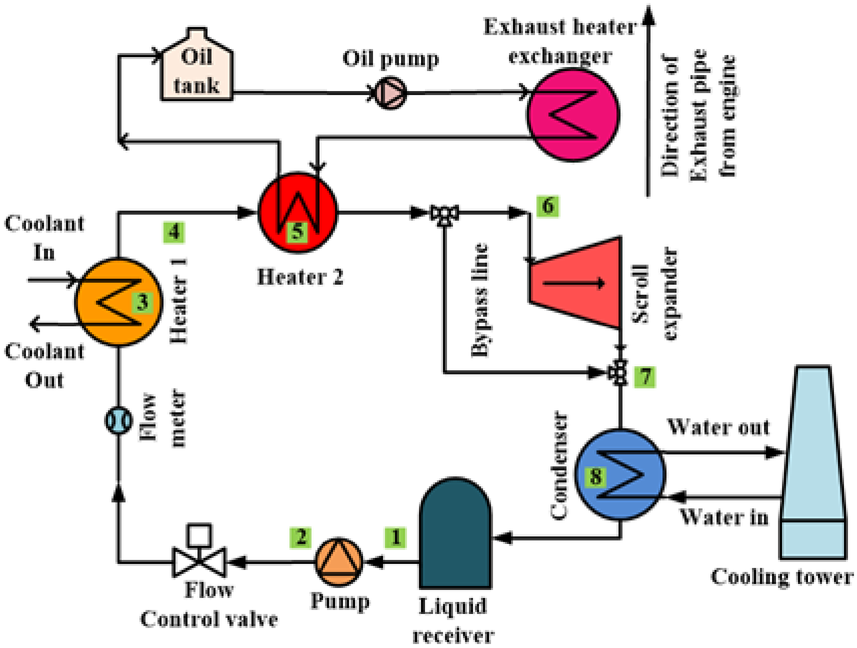


Fig. 1. Schematic diagram of the designed small scale engine coolant and exhaust recovery system using a 1 kW scroll expander.

3. Methodologies

The targeted heat source of the ORC system is a 6.5 kW Yanmar ICE with the model number YTG 6.5S. The selected evaluated point in this study is under the engine full load condition at rated rotational speed of 2400 RPM. Previous experimental study on this small engine under engine rated conditions suggested a recoverable waste heat from the coolant and exhaust of 4.67 kW and 4.54 kW, respectively [17]. The heat provided from the coolant and exhaust energy to the working fluid is calculated by the following equations, where the efficiency of the heater exchanger is set at 80%.

$$\dot{Q}_{Heater1} = \dot{m}\cdot(h_4 - h_2) = \dot{Q}_{coolant}\cdot\eta_{Heater1} \tag{1}$$

$$\dot{Q}_{Heater2} = \dot{m}\cdot(h_6 - h_4) = \dot{Q}_{exhaust}\cdot\eta_{Heater2} \tag{2}$$

The mass flow rate of the working fluid can be used to calculated the rotational speed of the scroll device by using Eq. (3)

$$\dot{m} = \frac{N}{60}\cdot\frac{V_{SV}}{v_{su}} \tag{3}$$

The rotational speed of the expander is represented as N, the swept volume targeted scroll expander is V_{SV} and v_{su} represents the specific volume at the inlet of the expander. We conducted geometric study of the targeted scroll expander for the calculation of physic parameters used in the calculations. The comparison of the geometric model and the measured fixed scroll of the scroll expander is shown in Fig. 2. The geometric calculation results suggest the suction volume and exhaust volume per turn of the scroll expander is 11.82 cm^3 and 41.38 cm^3, respectively.

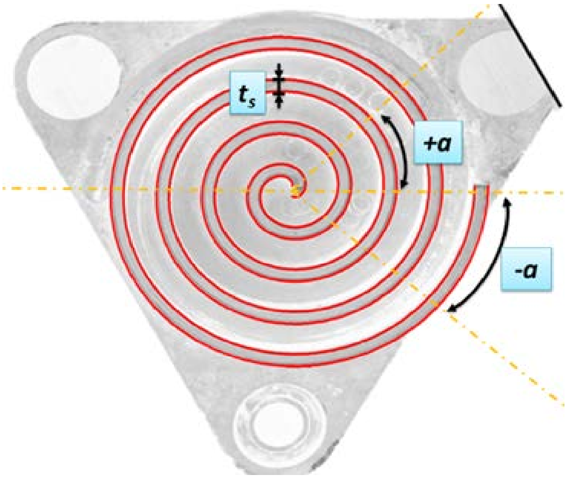


Fig. 2. The geometric and fixed scroll

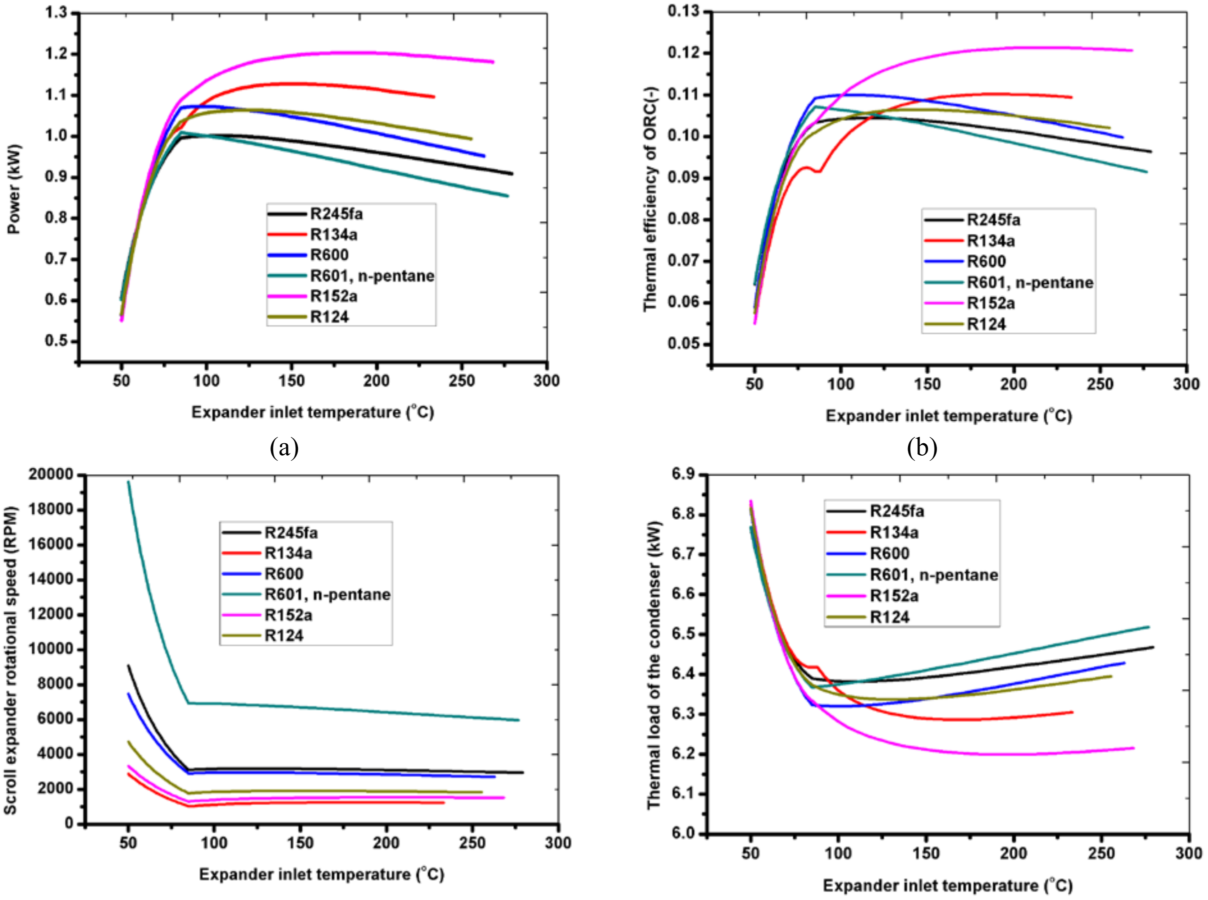
The net power output from the scroll expander is therefore defined as Eq. (4) [15], where h_{su} is the specific enthalpy at the inlet of the expander, h_{d_s} is the designed exhaust specific enthalpy of the scroll expander after isentropic expansion process P_{d_s} is the designed exhaust pressure, and v_d is the designed specific volume.

$$\dot{W} = \dot{m} \cdot \left[(h_{su} - h_{d\_s}) + (P_{d\_s} - P_{ex}) \cdot v_d \right] = \dot{m} \cdot (h_6 - h_7) \cdot \eta_{\mathrm{exp}ander} \tag{4}$$

4. Results and discussion

4.1. Effects of expander inlet temperature on ORC performance at engine rated condition

Six working fluids R245fa, R134a, R600, R601, R152a and R124 have been selected. The effects of designed expander inlet temperature on the ORC performance have been studied to evaluate the net power, thermal efficiency, rotational speed of the scroll expander and condenser load of the ORC system, which are plotted in Fig. 3. The coolant temperature of the engine is about 85 ºC [5] under the real engine operational condition. Therefore, the maximum evaporating temperature of the design is set at 85 ºC in order to fully recover the coolant energy. When the expander inlet temperature is from 50 to 85 ºC, the designed evaporating temperature of the ORC equals to the expander inlet temperature and the inlet condition of the expander is located at the saturated vaporization line of the working fluid. The results under expander inlet temperature from 50 to 85 ºC, can be used to predict the performance under the start-up process of the ICE and ORC system, when the heat source conditions of coolant and exhaust still cannot meet the requirements. When the inlet temperature of the scroll expander is higher than 85 ºC, the heat provided to operate the ORC to the superheated region is from the exhaust energy. Results suggested the net power output from the ORC increased from 0.5 kW to about 1.0 kW with the increase of the designed scroll expander inlet temperature from 50 to 85 ºC. The performance of ORC system using R245fa, R600 and R601 reduces with the increase of superheated temperature as indicated in Fig. 3 (a) and (b). The best performance of the system is 1.2 kW under designed superheated inlet temperature at 200 ºC, when R152a is used as ORC working fluid. The average power generated from the ORC system using R134a is as high as 1.15 kW under designed scroll expander temperature from 140 to 170 ºC.



(a) (b)

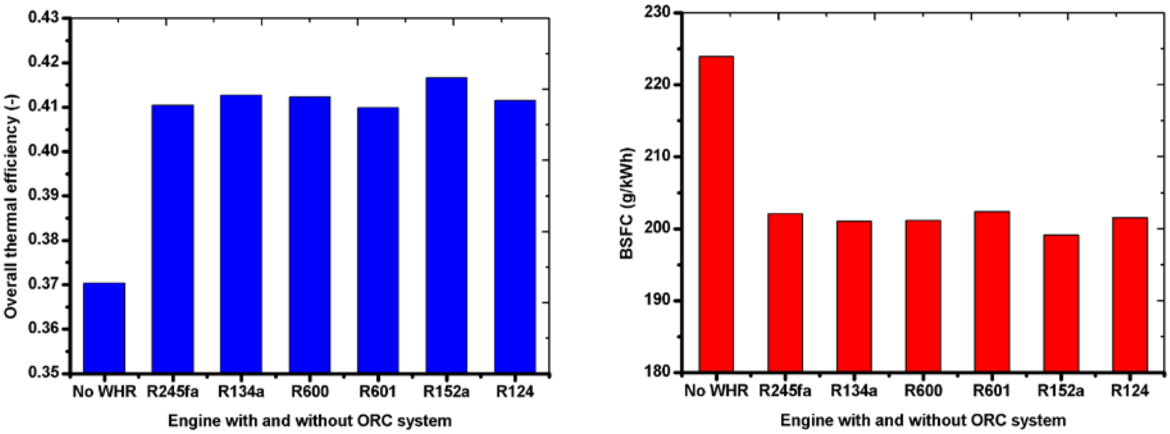
(c) (d)

Fig. 3. Effects of expander inlet temperature on ORC performance with different working fluids (a) Net power performance of ORC; (b) Thermal efficiency of ORC; (c) Rotational speed of the scroll expander; (d) Condenser load

The rotational speed of expansion machine is critical for small scale power generation system in order to convert the mechanical power into electricity. The calculated results can be found in Fig. 3 (c). Results indicate the expander rotational speed sharply reduced under the increase of designed evaporating temperature of the ORC. R601 has the highest rotational speed compared with other working fluids under the same operational conditions. The designed superheating has limited effect on the rotational speed of the scroll device. The operational region of conventional generator is normally from 500 RPM to 3000 RPM, which means the R601 should not be selected in the designed ORC system for electricity generation. When the optimal efficiency of the electric generator is about 3000 RPM, R245fa and R600 are desirable to be selected to meet the optimal performance for electricity generation. R134a and R152a are recommended to be used when the required rotational speed is under 2000 RPM. The evaluation of the thermal load of the condenser has also been conducted. Results indicated the R152a requires the lowest thermal load among the selected working fluids as shown in Fig.3 (d). The condenser load is about 6.2 kW under expander inlet from 150 to 270^{o}C using R152a as ORC working fluid. R134a has the second lowest thermal load requirement on the condenser, which is about 6.3 kW. The results indicated R134a and R152a have the advantages of high net power output, low requirement of condenser load, low requirement of rotational speed of the expander, and relatively high overall thermal efficiency compared with other four candidate working fluid under the same ORC operational conditions.

4.2. Influence on engine performance under rated condition

The effects of using the designed engine coolant and exhaust ORC system with different working fluids under the engine rated condition has been evaluated and the results are shown in Fig. 4. Based on the results obtained in Fig. 3, the inlet scroll expander has been set at 150 ºC for the calculation of the over effects on the system performance. As illustrated in Fig.4 (a), the integrated ORC can potentially improve the overall system efficiency from 37% to averagely 41% with different ORC working fluid. The highest thermal efficiency has been achieved as high as 41.5 % when R152a is used in the ORC system. The fuel economy is the other important parameter to evaluate the effects of using the integrated ORC system for engine waste heat recovery. Under the engine rated power, the Brake Specific Fuel Consumption (BSFC) of the engine is about 223.9 g/kWh [17], which can be potentially be improved by 10 % when the designed ORC system has been used to recovery engine coolant and exhaust energy. The averagely BSFC of the system integrated with ORC system is about 200 g/kWh as shown in Fig. 4 (b).



(a) (b)

Fig. 4. Effects on the engine overall performance: (a) Overall thermal efficiency using designed ORC with different working fluids (b) Effects on Brake Specific Fuel Consumption using designed ORC with different working fluids

5. Conclusions

In this paper, the study of a designed small scale engine coolant and exhaust heat recovery system using different ORC working fluids has been reported. The evaluations of the designed ORC system using a targeted scroll expander to recover the waste heat from a 6.5 kW small ICE have been conducted. Results suggested R134a and R152a have the advantages of relatively high net power output, low requirement of condenser thermal load, low rotational speed of the scroll expander and high overall ORC thermal efficiency compared with other four candidates. The highest power output from the designed under engine rated condition is as high as 1.2 kW, when the designed ORC superheated inlet temperature is set at 200 ºC using R152 as the working fluid. The R134a can achieve the second highest power output performance averagely 1.15 kW when the designed scroll expander inlet temperature from 140 to 170 ºC. The results of the evaluation on the overall energy system of ICE show that the designed ORC can potentially improve the overall efficiency from 37% to averagely 41%, which is about by 11.2% improvement under the engine rated condition. The fuel economy of the ICE can also be potentially be reduced from 223.9 to 200 g/kWh, when the integrated engine coolant and exhaust heat recovery system has been adopted.

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