EXPERIMENTAL STUDY OF AN ORC (ORGANIC RANKINE CYCLE) WITH THERMAL OIL FOR WASTE HEAT RECOVERY OF A DIESEL ENGINE.

Gequn Shu, Mingru Zhao*, Hua Tian, Guangdai Huang, Yongzhan Huo
* State key laboratory of engines, Tianjin University,
92 Weijin Road, Nankai District, Tianjin 300072,
China
zhaomr@tju.edu.cn, Tel:+86-138-2018-1525

ABSTRACT

Waste heat from exhaust gas of diesel engine could be recovered to increase engine's efficiency and decrease exhaust pollution. In this research, an Organic Rankine Cycle (ORC) test bench with thermal oil as heat transfer medium was set up to recovery the waste heat from a 240kW heavy-duty diesel engine. R123 was chosen for working fluid and expansion valve was employed temporarily to investigate the properties of waste heat. Experiments have been conducted to measure the available heat of exhaust gas in the different loads and speeds of engine. The results show the amount and the quality of waste heat that can be potentially recovered by this test bench during different conditions of engine, of which the maximum exergy and the potential power ability are 18.53kW, and 9.67kW separately. The maximum efficiency of exergy and potential power ability are 26.80% and 14.32% separately. Also, with thermal oil cycle integrated, the transient performance of the ORC test bench was investigated.

1. INTRODUCTION

With oil crisis and environmental pollution getting serious, the energy saving and emission reduction technology has become the main research direction in the ICE (internal combustion engine) area. The study (Dieter *et al*., 2012) on diesel engine found that approximately 60% of the energy released by fuel is dissipated to the environment in the form of waste heat, which shows a great potential for diesel engine towards higher efficiency if the waste heat can be recovered. The Rankine cycle system is an efficient way to recover those heat (in comparison with other technologies such as thermo-electricity and absorption cycle air-conditioning). The idea of associating a Rankine cycle to an ICE has existed for a long time and the first technical developments showed up under the 70's energy crisis. For instance, Mack Trucks (1976) designed and built a prototype of ORC system operating on the exhaust gas of a 288 HP truck engine. A 450 km on-road test demonstrated the technical feasibility of the system and its potential market: an improvement of 12.5% of the fuel consumption was achieved. Systems developed today differ from those of the 70's because of the advances in the development of heat exchangers and expansion devices, plus the broader choice of working fluids. However, at the present time, Rankine cycle systems are under development, but no commercial solution on vehicle seems to be available yet.

Rankine Cycle has got a lot of attention for its high recovery efficiency and low impact on the backpressure of engine. However, considering the large variation of engine's working condition, SRC (steam rankine cycle) are unsuitable under low-and-medium duty for its needs of high temperature for superheating, where the organic working fluids show its advantages. Katsanos *et al.* (2012) conducted a theoretical study to investigate the potential improvement of the overall efficiency of a heavy-duty truck diesel engine equipped with a Rankine bottoming cycle for recovering heat from the exhaust gas. Two different working mediums, water and the refrigerant R245ca, are considered. The results under all operating conditions show that the brake specific fuel consumption improvement ranges from 10.2% (at 25% engine load) to 8.5% (at 100% engine load) for R245ca and 6.1% (at 25% engine load) to 7.5% (at 100% engine load) for water.

The ORC technology in cogenerative systems has by now reached a level of full maturity in biomass applications. In Europe, there are over 120 plants in operation with sizes between 0.2 and 2.5 MW electric. In these facilities, hot thermal oil is usually used as heat transfer medium, demonstrating a number of advantages, including low pressure in boiler, large inertia and simple adaptability to load changes, automatic and safe control and operation. Moreover, the transferred temperature for the hot side of ORC is lower, which ensures a very long working life of organic fluid. The utilization of a thermal oil boiler also allows operation without requiring the presence of licensed operators as for steam systems in many European countries. Several companies like Turboden, Pratt&Whitney have already made great achievements at biomass plants, of which the flue gas from boiler has the temperature as high as 1000℃. And the net electric efficiency can be above 18%. Although not as high as that of flue gas from biomass boiler, the temperature of exhaust gas from diesel engine can reach over 500℃, which is beyond the decomposition temperature of most regular organic fluids. So the thermal oil cycle were introduced by some researchers to deal with the high temperature of exhaust gas under heavy duty. Yu *et al.* (2013) did a theoretical study of an ORC system with thermal oil cycle for waste heat recovery of a 243kW diesel engine. R245fa was chosen as working fluid. The system shows that the maximum potential output power and recovery efficiency are as high as 14.5 kW and 9.2% separately. Jacek Kalina *et al.* (2011) compared the ORC containing thermal oil with the single and dual loop ORC when studying the WHR for gas engine theoretically. The results show that the recovery ability of ORC containing thermal oil is between the single and dual loop ORC. What's more, thermal oil cycle can avoid the use of gas-gas heat exchanger which is difficult to manufacture.

During this experiment, the highest temperature of thermal oil under each operating condition is in the range from 90 to 220 ℃, which makes R123 the suitable working fluid. Although with micro toxicity and GWP of 120, the R123 shows the best performance when recovering low-and-mid temperature waste heat. Tzu-Chen (2001) investigated the working fluids Benzene (C6H6), Toluene (C7H8), p-xylene (C8H10), R113 and R123. He reported that R123 had a better performance in recovering a low-and-mid temperature waste heat. Wang *et al.* (2010) studied the relationship between the system performance, the pressure ratio and the mass flow rates of several organic working fluids. The results show that R123 owns the maximal thermal efficiency and net output at the same mass flow rate or heat input among the several working fluids. Also, R123 was selected as the working fluid because of its non-flammable properties, low boiling temperature, chemical stability and low cost. Moreover, it has many characteristics such as high cycle efficiency, high thermal conductivity and moderate working temperature and pressure. Zhou *et al.* (2013) set up an ORC bench using R123 to recover waste heat from low-temperature flue gas. The maximum recovery efficiency reaches 22%. Li *et al.* (2013) constructed a low-temperature regenerative ORC using R123. Finally the system obtained 6kW output power and 7.98% recovery efficiency. So it’s necessary to investigate the maximum potential of the WHR system with R123 in the first time. Considering R123’s drawbacks, R245fa will be the next choice for its lower toxicity and less damage to environment still high efficiency close to R123.

In this case, the experimental study of ORC containing thermal oil seems an interesting topic for WHR of diesel engine on vehicle. Despite of the extra weight and complexity, the thermal oil does help to the frequently changing operating conditions of engine on road. This paper focuses on the characteristics of waste heat from diesel engine and the potential recovery ability of ORC using R123 in all conditions. The whole working condition is conducted also to find out the limit of the test bench and to lay the groundwork for on-road test. What's more, the impacts of thermal oil cycle on the transient performance of the ORC test bench were investigated.

2. EXPERIMENTAL SETUP

In this research, the heat source is the exhaust gas from an 8.4L 6-cylinder heavy-duty diesel engine, of which rated power is 240kW. This kind of engine was turbo-charged and widely used on the linehaul heavy trucks which are very likely to apply WHR techniques. The engine bench was equipped with a whole set of controlling and measurement device, which can keep the engine working steady under any specific condition with all the performance data recorded at the same time.

As shown in the Figure 1, the thermal oil cycle was added as the heat transfer medium, which can serve as a buffer to decrease the impact from the fluctuation of exhaust gas. Besides, since the highest temperature of exhaust gas can be close to 500℃, the thermal oil cycle effectively bring down the working temperature of R123 for its decomposition temperature is only 327℃, however, at the cost of lower quality of heat source, lower efficiency, higher irreversibility and longer response time.

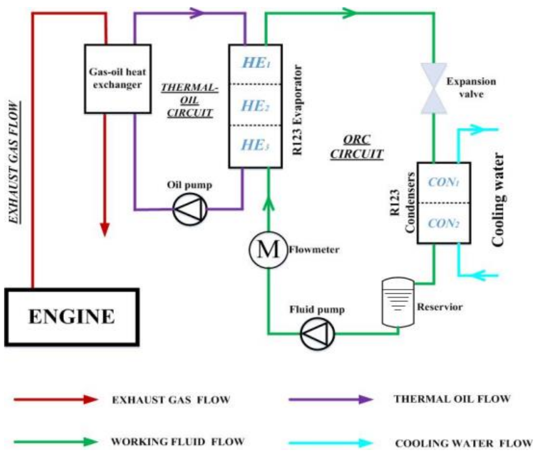


Figure 1: Structure of the ORC system

Dibenzyl-toluene (DBT) is chosen as thermal oil for its boiling point as high as 390℃, as well as its non-corrosiveness to metal which makes heat exchangers easy to manufacture. Also, DBT can resist oxidation well.

Table 1: Important parameters of heat exchangers

Name	Type	Heat transfer area
R123 evaporator	plate heat exchanger	4.61 m^2
R123 condenser	plate heat exchanger	5.18 m^2

Considering back pressure of engine, a shell-and-tube heat exchanger is chosen for thermal oil to absorb heat from exhaust gas. All the other heat exchangers in the thermal oil cycle and ORC system are plate heat exchangers as shown in Table 1. The oil pump is a 0.75kW centrifugal pump. A vortex shedding flowmeter (VSF) is installed to measure the volume rate of oil which is proved to be inaccurate because of its unexpected sensitiveness to the vibration of engine. Besides, the thermal oil can expand by as much as 25% under working condition which also makes it difficult to obtain its mass flow rate for the lack of its accurate density map. Those account for the absence of an energy balance analysis of thermal oil cycle. Then a 40L oil storage tank is installed above the whole system preparing to contain the expanded oil. When thermal oil get cold and shrink, they will flow back to system from tank by gravity.

The ORC system was set up with the thermal oil cycle as a whole test bench as shown in Figure 2. The chosen pump for ORC system is an 4kW plunger-driven diaphragm pump, for the membrane placed between the fluid chamber and the plunger zone ensures a perfect sealing of the circuit toward environment and allows reaching pressure as high as 3MPa, in spite of the low viscosity feature of the pumped liquid and the absence of any lubricant properties. Also, the mass flow rate can be controlled by adjusting the stroke length of plunger. A Coriolis flow meter was set before the pump inlet to measure the mass flow rate, while a damper was set after the pump to smooth the flow.

Figure 2: ORC system

An expansion valve was employed to temporarily take the place of the expander which are under manufacture. By controlling the expansion valve, the evaporating pressure can be adjusted. Since the mass flow rate of R123 is also influenced, the expansion valve was kept widely open during experiment to make sure the mass flow rate can be controlled by ORC pump more accurately and be large enough to absorb more heat from thermal oil.

A refrigerating unit using R22 as refrigerant was set up to produce cooling water whose temperature and flow rate can be adjusted conveniently. Therefore the condensing temperature and pressure of ORC system can be maintained in a certain range.

All the hot parts of the facility have been thermal insulated, also the test bench has been equipped with the necessary auxiliaries such as measurement devices, particular thermocouples and pressure sensors.

3. ANALYSIS OF EXPERIMENT AND RESULTS

In order to comprehensively analyze the characteristics of the waste heat form the exhaust gas, the experiments were conducted as the engine speed varied in the range of 1200 rpm to 2200 rpm with a 200rpm interval, and the engine load varied in the range of 20% to 100% with a 20% interval. The results are drawn as contour map to show the tendencies.

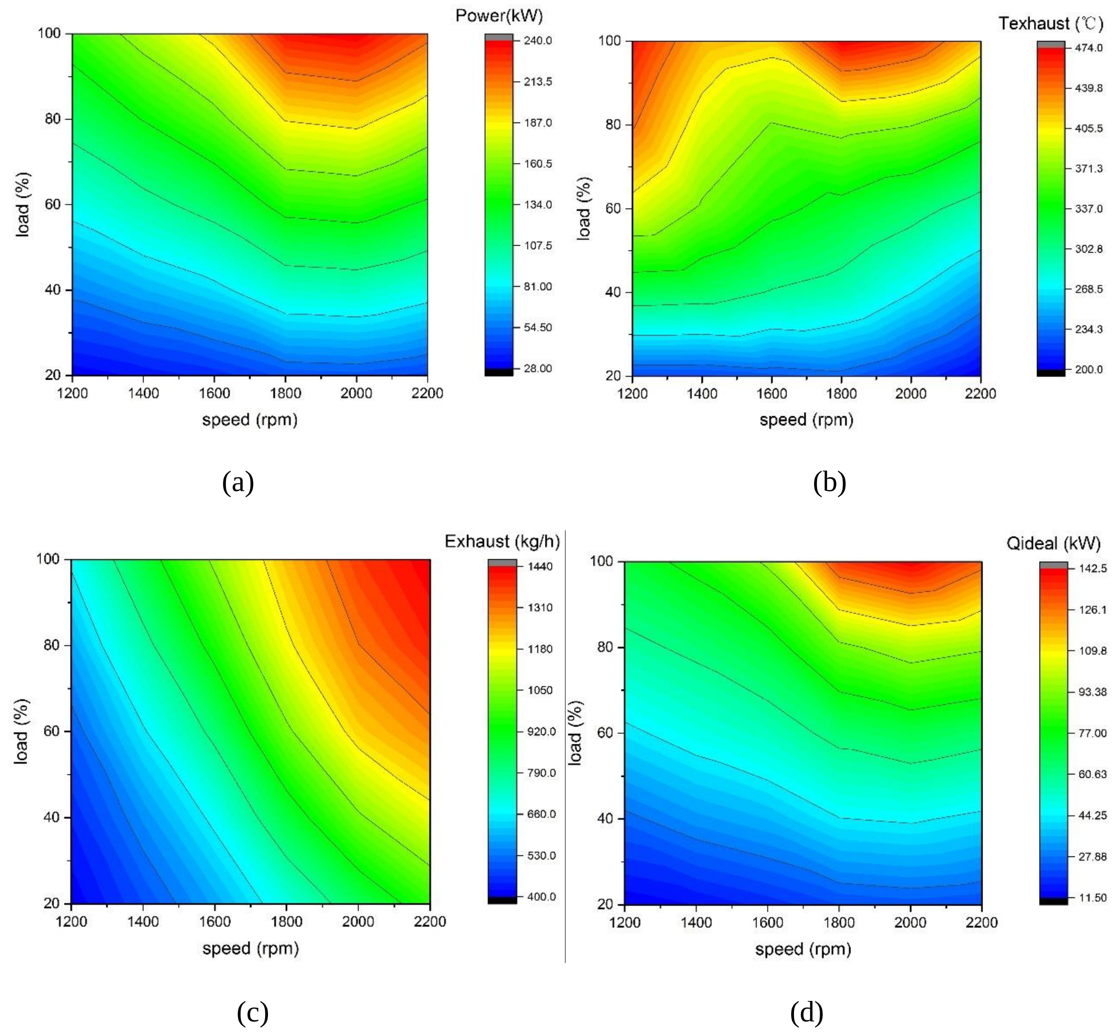


Figure 3: (a) Power output map; (b) Temperature map of exhaust gas; (c) Exhaust mass flow rate; (d) Q_{ideal};

According to the measured test data, the Power output map is shown in the Figure 3(a). At the rated condition as 2000rpm and 100%load, the effective power output of the diesel engine is 240 kW. To better observe the quality of the waste heat from exhaust gas, the temperature map of exhaust gas is shown in the Figure 3(b). When the engine speed goes up from 1200rpm to 1600 rpm, the temperature of exhaust gas decreases for the shorter combustion time of each cycle. When the speed goes above 1600rpm, the temperature slightly increases then decrease for the mutual effect of more air and shorter combustion time. The highest temperature appears as 474℃ at the working condition of 1800rpm and 100% load, instead of the rated condition whose power output is the largest. The exhaust mass flow rate as shown in the Figure 3(c) also helps to study the distribution of the waste heat. In the absence of EGR, the exhaust mass flow rate equals the sum of intake mass flow rate and fuel consumption rate under the ideal condition that fuel leakage is nonexistent. From the figure, the exhaust mass flow rate increases along with the engine speed for more intake air and tops as 1437.32 kg/h at the working condition of 2200rpm and 100% load. The figures above show that the temperature and mass flow rate of exhaust gas doesn't share the similar trend and further investigation is necessary.

Exhaust energy absorbed by thermal oil during experiments is calculated using an approximation method. $C_xH_yO_z$ can denote the average molecular composition for common diesel fuel because the petroleum-derived diesel fuel is a very complicated mixture of alkane, alkene and arene, with minimal amount of sulphur and nitrogen which can be neglected. In this composition, x, y, and z respectively represent the moles of the C, H, O. Their mole ration is shown as the Equation (1) below:

$$\mathrm{x{:}\,y{:}\,z} = 0.87{:}\,0.126{:}\,0.004 \tag{1}$$

The air is presumed as the mixture of nitrogen and oxygen with the mole ration of 3.098:1. Hypothetically, the combustion products only consist of carbon-dioxide, water, nitrogen, and oxygen for the usually oxygen-enriched combustion of diesel engine, of which nitrogen comes from air and oxygen is residual from combustion. Based on the atomicity balance principle, the mass fraction of combustion products is obtained by the Equation (2) below:

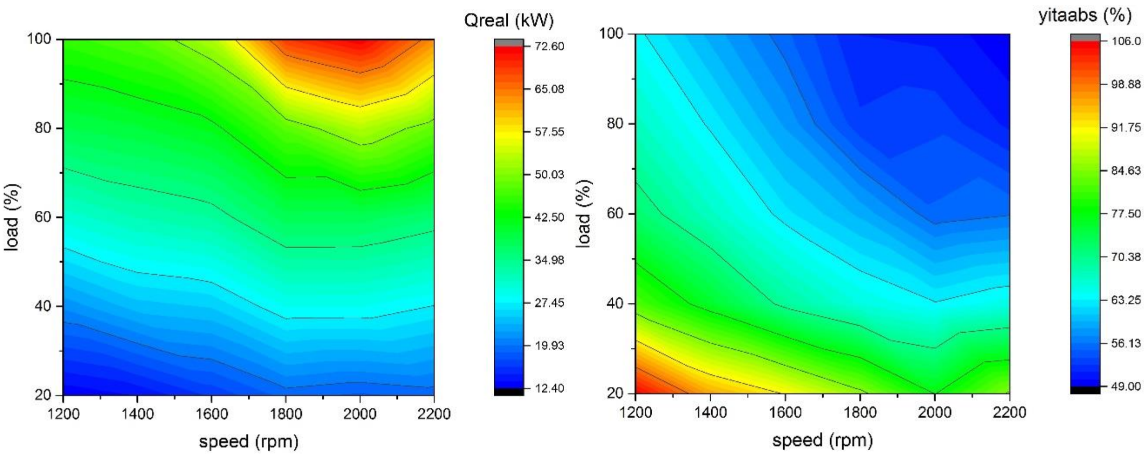
$$C_xH_yO_z + \left(\mathrm{x} + \frac{y}{4} - \frac{z}{2}\right)(O_2 + 3.098N_2) = xCO_2 + \frac{y}{2}H_2O + 3.098\left(\mathrm{x} + \frac{y}{4} - \frac{z}{2}\right)N_2 \tag{2}$$

According to the measured temperature and pressure, the exhaust energy in exhaust gas absorbed by thermal oil can be calculated by the Equations (3, 4) below:

$$h_{exh}(T,P) = \omega_{CO_2}h_{CO_2}(T,P) + \omega_{H_2O}h_{H_2O}(T,P) + \omega_{O_2}h_{O_2}(T,P) + \omega_{N_2}h_{N_2}(T,P) \tag{3}$$

$$Q_{exh} = m_{exh}(h_{exh}(T_{exhin}, P_{exhin}) - h_{exh}(T_{exhout}, P_{exhout})) \tag{4}$$

In practice, the temperature of exhaust is suggested above 120℃, for the possible corrosion to exhaust pipes and heat exchangers by the acid formed in the exhaust gas if the temperature decreases below 120℃. Assuming the temperatures of the exhaust gas after the heat exchanger under every working condition are all 120℃, the exhaust energy in exhaust gas that has been fully absorbed can be calculated by equations (3, 4) above. It is denoted as Q_{ideal} and shown in the Figure 3(d). Surprisingly the variation trend of Q_{ideal} is highly similar to that of power output in Figure 3(a). The maximum exhaust energy is obtained at rated condition as 142.4 kW, because the temperature and mass flow rate are all on a high level at this condition.



(a) (b)

Figure 4: (a) Q_{real}; (b) η_{abs};

But during this research, the exhaust energy cannot be fully recovered under all working condition. Based on the experiment data and equations above, the exhaust energy that is practically transferred to thermal oil can be calculated as shown in the Figure 4(a) and be denoted as Q_{real}. The similar trend with Q_{ideal} shows that the amount and the quality of the waste heat in exhaust gas are all considerable under the high speed and heavy duty conditions, which from the figures are believed to be the suitable conditions to recover exhaust energy. The heat absorption efficiency as η_{abs} is defined as Equation (5) below. The Figure 4(b) shows that the higher speed and heavier load the condition is, the lower absorption efficiency it has. Values greater than 100% mean the temperature of exhaust gas after heat exchanger is below 120℃, which also mean the available heat have been absorbed completely. With the maximum value as 106% and the minimal as 49.18%, the η_{abs} of most working conditions are below 70%, which means the capacity of heat exchanger are not efficient enough. The suitable working conditions for this heat exchanger are to be found under further study.

$$\eta_{abs} = \frac{Q_{real}}{Q_{ideal}} \cdot 100\% \tag{5}$$

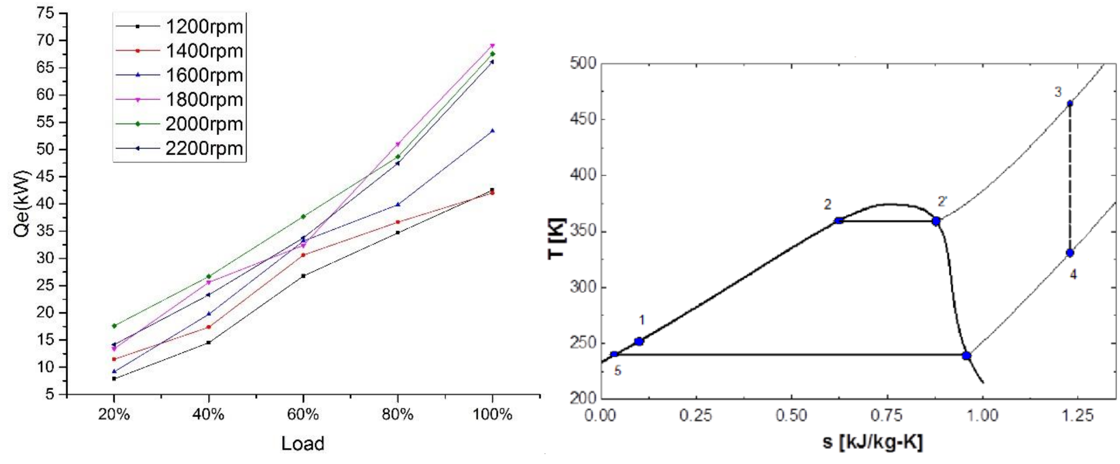
However with the thermal oil onboard and the massive amount of it, two major consequences happened. First, the response of the whole ORC system becomes slow. Since there are more than one parameter needed to be adjusted such as mass flow rate of R123 and cooling water of which the temperature needs adjustment too, the steady state of the ORC system are difficult to maintain for recording. It usually takes more than half hour to be adjusted and stabilized. Second, the massive amount of the thermal oil leads to the massive amount of heat it carries as the heat transfer medium. Under the condition that the engine speed and load vary, especially from the heavy duty to the light duty, the situation may happens that the heat transferred from exhaust gas to thermal oil is less than that from thermal oil to R123, which leads to a misconception of Energy Non-Conservation. To reduce the impacts form above as much as possible, the rules below are followed during the experiments:

1, if the parameters vary within the range of less than 1% in 5 minutes, it can be seen as the quasi-stable state and ready for recording.

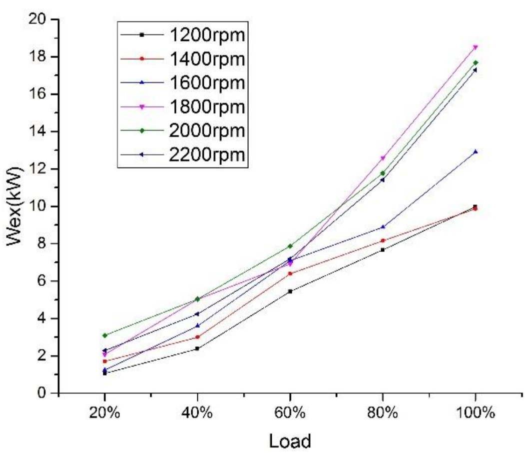
2, the adjustment of the parameters needs to be as minimal as possible to avoid the over-swing.

The waste heat absorbed by R123 as Q_e is calculated by the Equation (6) below and shown in the Figure 5(a). Affected by the thermal oil cycle, the maximum waste heat is absorbed as 69.14 kW at the condition of 1800rpm and 100% load instead of rated condition.

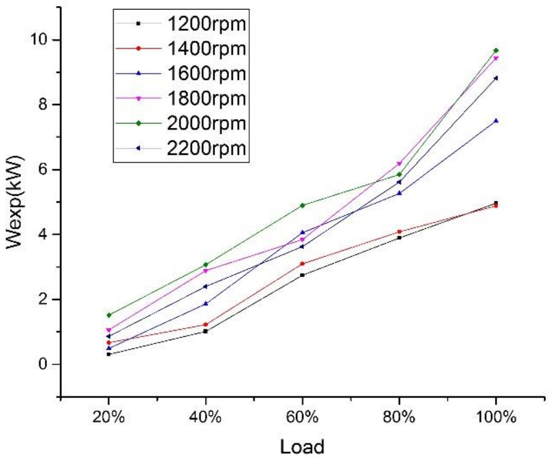
$$Q_e = m_{R123}(h_{R123}(T_{R123in}, P_{R123in}) - h_{R123}(T_{R123out}, P_{R123out})) \tag{6}$$



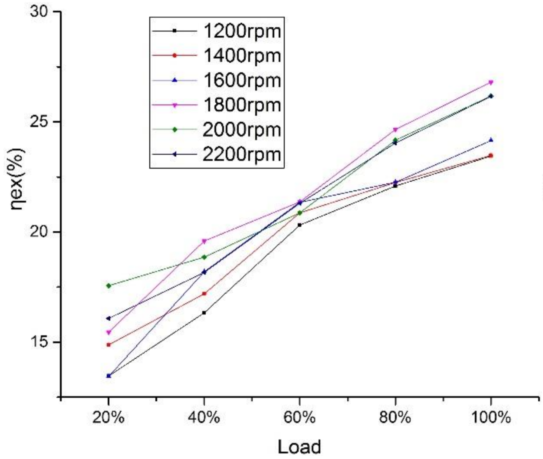
(a) (b)



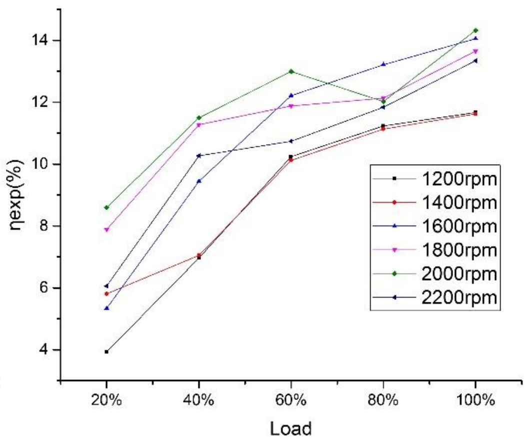
(c)



(d)



(e)



(f)

Figure 5: (a) Q_e; (b) T-s map 0f R123; (c) W_{ex}; (d) W_{exp}; (e) η_{ex}; (f) η_{exp};

Although the expander is unavailable yet, the potential power ability of R123 after evaporator can still be studied with an expansion valve. The Figure 5(b) is the T-s map of R123, point 1 to point 3 is the evaporation process with point 3 as the exit state after evaporation, of which the temperature and pressure are measured. Assuming the expansion process as isentropic, point 4 is the exit state, of which the temperature and pressure are measured. Subscript 0 are for the environmental state. The exergy and potential power ability by expander can be denoted as W_{ex}, W_{exp} separately and calculated by the Equation (7-10) below:

$$e_{ex} = (h_3 - h_0) - T_0(s_3 - s_0) \tag{7}$$

$$e_{exp} = (h_3 - h_4) \tag{8}$$

$$W_{ex} = m_{R123} \cdot e_{ex} \tag{9}$$

$$W_{exp} = m_{R123} \cdot e_{exp} \tag{10}$$

Figure 5(c) shows that the trend of W_{ex} is highly similar with that of Q_e for the obvious reason that the more heat R123 absorbed, the more work it can potentially make. The maximum exergy as 18.53kW is obtained at 1800rpm, 100% load condition. Its corresponding exergy efficiency as η_{ex} in Figure 5 (e) is 26.80%, with the definition of the exergy efficiency as the Equation (11) below:

$$\eta_{ex} = \frac{W_{ex}}{Q_e} \cdot 100\% \tag{11}$$

$$\eta_{exp} = \frac{W_{exp}}{Q_e} \cdot 100\% \tag{12}$$

The trend of W_{exp} in Figure (d) is not like that of Q_e for the condensing pressure at point 4, which determines h_4 , affects W_{exp} too. The maximum potential power ability by expander as 9.67kW is obtained at rated condition. Its corresponding potential power efficiency as η_{exp} in Figure 5 (f) is 14.32%, with the definition of the potential power efficiency as the Equation (12) above. The sudden fall at 2000rpm, 80% load condition results from the failure of maintaining the condensing temperature which leads to an unpleasantly high condensing pressure influencing W_{exp} as mentioned.

Despite of the negative impacts of the thermal oil, the inertia it brings to the ORC system can be helpful against the variation of engine condition. To investigate its positive impacts, the transient performance of ORC system when engine shuts down is studied. The light duty condition of 1200rpm and 20% load is chosen instead of the heavy duty because there could be serious damage to engine when it shuts down in a high temperature level. The Figure 6 shows the performance of ORC system when the engine shut down at 100th second and the data was recorded every 20 seconds. δT indicates the superheat degree of R123 after evaporator, which means vapor state when it's above zero. During the time from 100th second to 680th second, δT goes down obviously from 15.03 ℃ to 4.028 ℃ while Q_e only decreases from 7.452 kW to 7.108 kW and W_{ex} from 0.8818 kW to 0.8022 kW, which means the thermal oil has been transferring heat to R123 even though the engine has shut down. At around 700th second, δT eventually goes down below zero which means R123 is gas-liquid mixed state which has little ability to absorb heat and output power. Figure shows that after 700th seconds, Q_e goes down sharply to 1.423 kW and W_{ex} to 0.1034 kW which means the WHR system can keep outputting effective power for nearly 10 minutes after engine shuts down from a light duty condition. δT is proven to be a reliable indicator for safety that a positive superheat not only protects the expander blade from erosion, but also ensures the effective heat absorption and power output.

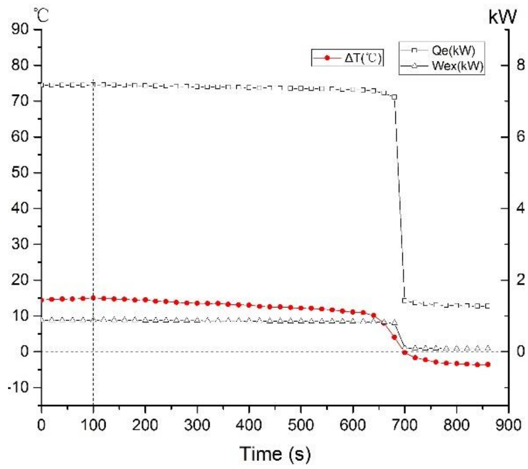


Figure 6: Transient performance of ORC system when engine shuts down

4. CONCLUSIONS

This paper elaborates the preliminary investigation of an Organic Rankine Cycle (ORC) test bench with thermal oil as heat transfer medium to recovery the waste heat from a 243kW heavy-duty diesel engine. The conclusions are below:

1, The exhaust mass flow rate goes up along with engine speed, however the temperature doesn't share the same trend.
2, Although the temperature and the mass flow rate of exhaust gas at rated condition are not the highest, the exhaust energy at this condition are the largest, for the temperature and the mass flow rate are all above a high level.
3, The trends of Q_e and W_{ex} are highly similar, which mean the high speed and heavy duty conditions are the suitable conditions to recover exhaust energy. The maximum values of Q_e, W_{ex} and W_{exp} are 69.14 kW, 18.53kW, and 9.67kW separately. The maximum efficiency of exergy and potential power ability are 26.80% and 14.32% separately.
4, Despite of the difficulties the thermal oil adds to ORC system in experiment, the inertia it brings can be helpful against the variation of engine condition. And the superheat δT proves to be a reliable safety indicator.

NOMENCLATURE

h	specific enthalpy
ω	mass fraction
m_{exh}	mass flow rate of exhaust gas
m_{R123}	mass flow rate of R123
e	specific exergy

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