Performance analysis of exhaust heat recovery using organic Rankine cycle in a passenger car with a compression ignition engine

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Abstract. Compression ignition engines transform approximately 40% of the fuel energy into power available at the crankshaft, while the rest part of the fuel energy is lost as coolant, exhaust gases and other waste heat. An organic Rankine cycle (ORC) can be used to recover this waste heat. In this paper, the characteristics of a system combining a compression ignition engine with an ORC which recover the waste heat from the exhaust gases are analyzed. The performance map of the diesel engine is measured on an engine test bench and the heat quantities wasted by the exhaust gases are calculated over the engine’s entire operating region. Based on this data, the working parameters of ORC are defined, and the performance of a combined engine-ORC system is evaluated across this entire region. The results show that the net power of ORC is 6.304kW at rated power point and a maximum of 10% reduction in brake specific fuel consumption can be achieved.

1. Introduction
The increasing fuel costs and diminishing petroleum supplies are forcing governments and industries to increase the power efficiency of engines. A cursory look at the internal combustion engine heat balance indicates that the input energy is divided into roughly three equal parts: energy converted to useful work, energy transferred to coolant and energy lost with the exhaust gases. To improve the thermal efficiency of an internal combustion engine, an organic Rankine cycle (ORC) can be used to recover the waste heat. Several studies analyzing the ORC performances have been conducted recently [1–6]. One important challenge is determining how to bottom an ORC to an internal combustion engine to maximize energy efficiency during all operating regions of the engine. Results have been reported on applications with a gasoline engine [7, 8] and a diesel engine [9, 10].

In this study, the performance of an ORC designed to recover the exhaust waste heat from a diesel engine is evaluated theoretically. R245fa is selected as the working fluid of the ORC. To evaluate the system performance when combined with a light-duty diesel engine, the waste heat quantities are first calculated using engine test data. Finally, the performance map of the combined system is calculated and compared to a system with a non-bottoming ORC.
2. System description

The schematic of an ORC for exhaust heat recovery of a diesel engine is shown in figure 1. The fresh air is boosted by the compressor. Then, the air enters the engine cylinders, which combusts with the injected fuel. After expanding in the cylinders, the high-temperature gas is exhausted to the turbine to be expanded further. Here, at the turbine outlet, the temperature of the exhaust gas is still high between 200 °C and 600 °C. This high-temperature waste heat can be utilized via bottoming an ORC, where the evaporator is connected to the turbine outlet, which can be considered as a counter flow arrangement in this study. R245fa is used as the working fluid on the tube side of the evaporator because of its good safety and environmental properties [4].

![Figure 1. Schematic of an ORC for engine exhaust heat recovery.](image)

The associated T–s diagram of the ORC is described in figure 2. Taking the practical operating conditions of a vehicle engine into account, the condensation temperature of the R245fa is set to 318 K in this investigation, and the evaporation pressure is set to 2 MPa. In the preheated zone, the sub-cooled liquid is heated to a saturated liquid state 2 sat. Subsequently, the working fluid continues to evaporate to a saturated gas state 3 sat in the two-phase zone. Then, the saturated gas is heated further until the working fluid temperature increases with 25 °C. The ORC working parameters configured above can be used to maximize the thermal efficiency of the ORC according to Ref. [11].

![Figure 2. T–s diagram of the ORC.](image)
The ORC is a vapor power cycle used in numerous applications to generate electrical power. Figure 1 shows a schematic of a simple ORC. It is composed by four main components: a pump, a heat exchanger, a turbine/generator and a condenser. The ideal thermodynamic cycle includes the following processes: an isentropic compression process in a pump (1-2), an isobaric heat transfer process in a heat exchanger (2-3), an isentropic expansion process through a turbine (or other expansion machine) (3-4), and an isobaric heat transfer process in a condenser (4-5-1).

The pump supplies the working fluid to the heat exchanger, where the working fluid is heated and vaporized, removing heat from the exhaust gases. The working fluid leaves the heat exchanger in saturated or superheated state. The high enthalpy vapor is then expanded in the expander (usually a turbine), which is coupled to a generator that delivers the ORC power output. After the expander, the working fluid enters the condenser where it condensates.

### 3. ORC thermodynamic model

This section presents the equations used to estimate the exhaust WHR potential using ORC. Before creating the mathematical model of this system to simplify the analysis, some general assumptions are formulated as follows:

- Steady-state and steady-flow condition;
- No pressure drops in heat exchangers and connecting pipes;
- The condenser temperature is assumed to be 45 °C;
- Exhaust gas temperature at the exit of evaporator is 140 °C due to prevent condensation of components;
- The expander mechanical efficiency, $\eta_e = 70\%$;
- The efficiency of the pump, $\eta_p = 80\%$;
- Ambient temperature is 20 °C;
- Heat exchanger effectiveness $\varepsilon_e = 1$.

The pump power is given by:

$$\dot{W}_p = \dot{m}_{ref} (h_2 - h_1) [kW]$$  \hspace{1cm} (1)

Where: $\dot{m}_{ref} [kg/s]$ is the organic fluid mass flow rate, $h_1 [kJ/kg]$ is the specific enthalpy of the organic fluid leaving the condenser and entering the pump, $h_2 [kJ/kg]$ is the specific enthalpy of the organic fluid leaving the pump and entering the evaporator.

The heat absorbed from the exhaust gases by the working fluid in the heat exchanger is given by:

$$\dot{Q}_{EVP} = \dot{m}_{ref} (h_3 - h_2) = \dot{m}_{gas} c_{Pgas} (t_{gasin} - t_{gasout}) [kW]$$  \hspace{1cm} (2)

Where: $c_{Pgas} [kJ/kgK]$ is the constant specific heat of the exhaust gases, $\dot{m}_{gas} [kg/s]$ is the mass flow rate of the exhaust, $t_{gasin}$ [°C] is the temperature of the exhaust gases leaving the engine and entering the evaporator, $t_{gasout}$ [°C] is the temperature of the exhaust gases leaving the evaporator, and $h_3 [kJ/kg]$ is the specific enthalpies of the organic fluid leaving the evaporator and entering the expander.

The turbine power is calculated by:

$$\dot{W}_E = \dot{m}_{ref} (h_3 - h_4) [kW]$$  \hspace{1cm} (3)

Where: $h_4 [kJ/kgK]$ is the specific enthalpies of the organic fluid leaving the expander and entering the condenser.

And the heat rejected from the condenser is given by:
\[ \dot{Q}_{\text{COND}} = \dot{m}_{\text{ref}} (h_4 - h_1) = \dot{m}_{\text{air}} c_{\text{Pair}} (t_{\text{aout}} - t_{\text{ain}}) \text{[kW]} \]  

(4)

Where: \( c_{\text{Pair}} \text{[kJ/kgK]} \) is the constant specific heat of the cooling air, \( \dot{m}_{\text{air}} \text{[kg/s]} \) is the mass flow rate of the cooling air, \( t_{\text{ain}}, t_{\text{aout}} \text{[°C]} \) is the temperature of the cooling air entering and leaving the condenser.

The ORC thermal efficiency can be defined as the net power produced referred to the heat received at the heat exchanger as follows:

\[ \eta_{\text{thm}} = \frac{\dot{W}_E - \dot{W}_P}{\dot{Q}_{EVP}} \]  

(5)

Exergy is a useful concept for evaluating the performance of various energy systems. Exergy is the maximum amount of work that can be done by a process as it approaches the thermodynamic equilibrium with its surroundings by a sequence of reversible processes [4,22]. Exergy destruction rate labels the loss of exergy during the process [4]. The exergy destruction is due to irreversibility occurring inside the system or the components of the system.

The exergy destruction rate for each process in the cycle (evaporation, expansion, condensation and pumping) can be expressed as follows:

\[ \dot{i}_p = \dot{m}_{\text{ref}} T_{\text{amb}} [(s_2 - s_1)] \text{[kW]} \]  

(6)

\[ \dot{i}_{\text{EVP}} = \dot{m}_{\text{ref}} T_{\text{amb}} \left[ (s_3 - s_2) - \frac{(h_3 - h_2)}{T_3} \right] \text{[kW]} \]  

(7)

\[ \dot{i}_E = \dot{m}_{\text{ref}} T_{\text{amb}} [(s_4 - s_3)] \text{[kW]} \]  

(8)

\[ \dot{i}_{\text{COND}} = \dot{m}_{\text{ref}} T_{\text{amb}} \left[ (s_4 - s_1) - \frac{(h_4 - h_1)}{T_1} \right] \text{[kW]} \]  

(9)

Where: \( T_{\text{amb}} \text{[K]} \) is the ambient temperature, \( T_i \text{[K]} \) is the temperature of the organic fluid before entering in pump, \( T_i \text{[K]} \) is the temperature of the organic fluid after leaving the evaporator, and \( s_1, s_2, s_3, s_4 \text{[kJ/(kgK)]} \) are the actual specific entropies of the working fluid at the inlet and exit of the pump, evaporator, expander and condenser.

Organic Rankine cycle exergy efficiency (the second law efficiency) can be expressed as:

\[ \eta_{\text{exg}} = \frac{\dot{W}_{\text{ORC}}}{\dot{Q}_{EVP} + \dot{i}_p + \dot{i}_{\text{EVP}} + \dot{i}_E + \dot{i}_{\text{COND}}} \]  

(10)

### 4. Engine waste heat evaluation

In this study, a commercial engine applied in vehicles was used for heat recovery. This is a 1.5 liter, compression ignition, four-cylinder engine with linear arrangement. Its technical characteristics and the fuel specifications were presented in table 1.

To design a reasonable system to utilize exhaust waste heat from the diesel engine with high efficiency, studying the energy distribution in the running process of the diesel engine is necessary. When a vehicle is running, the engine speed and load can vary through a wide range. Therefore, the engine performance test was conducted in an engine test cell in order to obtain the thermodynamic parameters of the exhaust and coolant systems overall possible engine operating regions as defined by the engine speed and output torque. The values for the output torque, the output power, the engine speed, the mass flow rate of the intake air, the injected fuel quantity, the exhaust gas temperature, and
the coolant temperatures at the outlet of the engine’s water jacket were all recorded for each load and speed configuration.

![Figure 3. Variations of BSFC (Brake Specific Fuel Consumption) over all engine operating regions [g/kWh]](image)

**Table 1.** The main technical performance parameters of the diesel engine. [13]

| Items                        | Parameters                          | Units            |
|------------------------------|-------------------------------------|------------------|
| Model                        | Diesel                              | [-]              |
| Cylinder number              | 4                                   | [-]              |
| Stroke and cylinder bore     | 88.3x75                             | [mm]             |
| Displacement                 | 1560                                | [cm³]            |
| Compression ratio            | 18:1                                | [-]              |
| Air intake type              | Turbocharged and Intercooled        | [-]              |
| Fuel injection system        | High pressure common rail           | [-]              |
| Rated power                  | 80                                  | [kW]             |
| Rated speed                  | 4000                                | [rpm]            |
| Maximum torque               | 240                                 | [Nm]             |
| Speed at maximum torque      | 1800                                | [rpm]            |
| Fuel lower heating value     | 42                                  | [MJ/kg]          |
Figure 4. Variation of engine power over all engine operating regions [kW]

In figure 3 and figure 4, the performance map of the diesel engine is plotted based on the test results from the diesel engine test bench. According to the measured test data, the engine speed varies in the range of about 1000 rot/min to 4500 rot/min, and the engine torque varies in the range of about 20 Nm to 240 Nm. At the engine rated condition, the effective power output of the diesel engine is 80 kW. When the engine is running in the working region where the engine speed varies in the range of 1500 rot/min to 2500 rot/min and the engine torque varies in the range of 120 Nm to 240 Nm, the diesel engine achieves the optimal fuel economy, and the BSFC (Brake Specific Fuel Consumption) is 225 g/(kWh). The diesel engine generally operates at a high speed and with a high load. Therefore, this diesel engine has good fuel economy.

Figure 5. Variation of exhaust temperature over all engine operating regions [°C]

Figure 6. Variation of exhaust mass flow rate over all engine operating regions [kg/h]

Figure 5 show the exhaust temperature and figure 6 show the exhaust mass flow rate of the diesel engine measured from the test. As we can see when the diesel engine is running with a medium–low load, the diesel engine exhaust temperature is affected less by the engine speed and increases primarily
with the engine load. When the engine is operating with medium-high speed and high load, the diesel engine exhaust temperature is relatively high and can reach 600 °C. As shown in figure 6, the exhaust mass flow rate of the diesel engine increases with the engine speed and engine load. This change primarily depends on the fuel consumption and the intake air mass flow rate. The intake air mass flow rate varies primarily with the engine speed whereas the engine load mainly affects fuel consumption. At the engine rated condition, the exhaust mass flow rate can reach up to 400 kg/h.

**Figure 7.** Variation of fuel combustion energy over all engine operating regions [kW]

The amount of waste heat from the light-duty diesel engine is then evaluated using the measured engine operating parameters as follow:

\[ \dot{Q}_{w} = \dot{Q}_{eg} - \dot{Q}_{air} [kW] \]

(11)

Where, \( \dot{Q}_{eg} [kW] \) is the total heat flux available in exhaust gases and \( \dot{Q}_{air} [kW] \) is the heat flux due to the fresh load.

**Figure 8.** Energy part of exhaust variation over all engine operating regions [kW]
The quantity of waste heat contained in exhaust gas is a function of both the temperature and the mass flow rate of the exhaust gas:

$$\dot{Q}_{eg} = \dot{m}_g c_{pg} T_g [kW] \quad (12)$$

In eq. (12) $\dot{m}_g [kg/s]$ is the exhaust gases mass flow rate; $c_{pg} [kJ/kgK]$ is the heat capacity at constant pressure of exhaust gases and $T_g [K]$ is the temperature of exhaust gases. Heat capacity of exhaust gases is considered from available data in literature according to experimental data [14].

The heat flux due to the fresh load can be determined:

$$\dot{Q}_{air} = \dot{m}_{air} c_{air} T_{air} [kW] \quad (13)$$

Where, $\dot{m}_{air} [kg/s]$ is the air mass flow rate $c_{air} = 1.013 \text{kJ}/(\text{kgK})$ is the heat capacity at constant pressure and its value is considered from data available in literature [15] and $T_{air} [K]$ is the measured ambient air temperature.

The combustion energy increases almost linearly with the engine output power, achieving 220 kW at the rated power point. Note that the indicated power and the waste heat quantity of the exhaust vary in a similar fashion. The reason for this is that the output power of a diesel engine is proportional to the quantity of injected fuel.

It can be concluded that, of the overall energy generated from the fuel combustion, only a relatively small proportion is used to produce useful work output, a large proportion of the energy is wasted, and the fuel energy released by combustion is far more than the effective power output of the diesel engine. Figure 8 shows that, over the whole operating range, the exhaust energy is greater than the engine effective power output.

5. Performance analysis of combined engine – ORC system

After evaluating the waste heat quantities for the exhaust, the performance of the ORC system was analyzed at each measured engine operating point using the established mathematical model. The proposed mathematical model in this report was solved using the Engineering Equation Solver (EES) code which includes thermodynamic properties for a large number of natural and manufactured fluids.

![Figure 9. Net power of ORC system variation over all engine operating regions[kW]](image-url)
Figure 10. Improvement of engine effective power over all engine operating regions[%]

Figure 9 shows the variation of the overall net power output of the ORC system over the whole operating range. The net power outputs of the ORC system increase with the engine speed and engine load. At the engine rated condition, the overall net power output of the ORC system reaches the upper limit and is 6.3 kW.

Figure 11. BSFC of combined system over all engine operating regions[g/kWh]

The improvement in the effective power over the engine’s entire working region is displayed in figure 10. In the high effective thermal efficiency region, the augmentation proportion is lowest (4–5%) because the waste heat quantity ratios are lower. The reason for this is better fuel combustion effects, the engine pumping losses are lower, and the ratio of the output power to the combustion energy is higher than in the other regions.
To assess the fuel economy of the diesel engine-dual loop ORC combined system, the effective thermal efficiency of the combined system is defined as:

\[
\eta_{\text{eng-ORC}} = \frac{P_{\text{eng}} + W_{\text{ORC}}}{\dot{Q}_{cb}}
\] (14)

Figure 12. Improvement of BSFC over all engine operating regions [%]

Where: \(\dot{Q}_{cb} [kW]\) is the heat flux received through fuel combustion; \(P_{\text{eng}} [kW]\) is the amount of mechanical power produced by engine, and \(W_{\text{ORC}} [kW]\) is the power of ORC.

The BSFC of the combined system is defined as follows:

\[
\text{bsfc}_{\text{eng-ORC}} = \frac{\dot{m}_{\text{fuel}}}{P_{\text{eng}} + W_{\text{ORC}}}
\] (15)

Where: \(\dot{m}_{\text{fuel}} [g/h]\) is the injected fuel quantity.

Figure 11 shows the variations of the BSFC of the diesel engine - ORC combined system. Compared with Figure 3, all the BSFCs of the combined system can be concluded to an improvement compared with the diesel engine itself over the whole operating range. Compared with the diesel engine itself, the BSFC can be reduced by 5%. Therefore, the fuel economy of the diesel engine combined with the ORC system is effectively improved.

6. Conclusions
This article focuses on the performance analysis of the diesel engine – ORC combined system. The maximum potential of the combined system performance was evaluated over the engine’s entire operating region. Based on this analysis, the following can be concluded:

- The combustion energy is much greater than the output power through most of the operating region;
- At medium - high speed and medium – high load the net power output of the combined system increased by 6 – 8 %;
- The BSFC is also found to significantly decrease throughout the engine’s operating region;
From the viewpoint of power performance and fuel economy, the ORC system is a promising technology to recover the waste heat from a vehicular light-duty diesel engine. The major downfalls of the technology are the increase of weight, costs and packaging complexity, and finally the difficult transient operation of these systems that should be properly considered before attempting to introduce the technology in mass production.

The results presented in this article only rely on steady state models. Thus, the results presented here are not totally representative of the reality. It is very well known that passenger car engines have a very transient behavior.

7. References

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