Parametric and exergetic analysis of a two-stage transcritical combined organic Rankine cycle used for multiple grades waste heat recovery of diesel engine

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Abstract. Diesel engine has multiple grades of waste heat with different ratios of combustion heat, exhaust is 400°C with the ratio of 21% and coolant is 90°C with 19%. Few previous publications investigate the recovery of multiple grades waste heat together. In this paper, a two-stage transcritical combined organic rankine cycle (CORC) is presented and analyzed. In the combined system, the high and low temperature stages transcritical cycle recover the high grades waste heat, and medium to low grades waste heat respectively, and being combined efficiently. Meanwhile, the suitable working fluids for high stage are chosen and analyzed. The cycle parameters, including thermal efficiency (ηₜₚ), net power output (Pₙₑₜ), energy efficiency (ηₑₑ) and global thermal efficiency of DE-CORC(η₉ₑₑ) have also been analyzed and optimized. The results indicate that this combined system could recover all the waste heat with a high recovery ratio (above 90%) and obtain a maximum power output of 37kW for a DE of 243kW. The global thermal efficiency of DE-CORC can get a max value of 46.2% compared with 40% for single DE. The results also indicate that all the energy conversion process have a high exergy efficiency.

1. Introduction
In China, Diesel Engine (DE) has consumed over 66 percent of overall fuel consumption. Since it is difficult for the maximum efficiency of ICE to be higher than 42% [1], large amount of fuel energy is rejected from the engine to the surroundings as waste heat in several forms. It was predicted by Vazaquez et al. [2] that if only 6% of the heat contained in the exhaust gases were converted to electric power, this would mean reduction of fuel consumption by 10%.

Diesel engine waste energy shows diverse characteristics from low grade to high grade, with different characteristics of energy balance distribution. In order to recover multiple grades waste heat, the traditional Organic Rankine Cycle is mostly studied currently, in which exhaust is used as the main heat source and cooling water is used for preheating. Rody El Chammas et al. [2] used a traditional ORC to recycle waste heat of exhaust and cooling water of a hybrid engine, the results showed that the waste heat of cooling water recovered accounted for 30%-50% of the total amount of the recovered water heat, while this part heat accounted for only 10% of the available heat of cooling water. Diego A. Arias [3], Iacopo Vaja [4], Jacek Kalina [5] also established a similar system models, and the results...
also showed that it was difficult to recover the waste heat of both exhaust and cooling water in the single-stage Organic Rankine Cycle, since these two kinds of the waste heat differs greatly. The second feature of multiple grades waste heat recovery of DE is that the temperature of exhaust gas is much higher than the decomposition temperature of most organic fluids. One solution was to try to adapt the high temperature exhaust gas and it may lead to a low thermal efficiency of the bottoming cycle, that is, using water as the only working medium for the steam Rankine cycle. Another solution is to lower the temperature and energy level of exhaust gas to satisfy the working medium and this would waste part of total exhaust energy to match temperatures of the exhaust gas and working fluid. AVL Company proposed the case that exhaust exchanged heat firstly with thermal oil, and then the heated oil conducted heat to the organic fluids [6,7]. However, the studies showed that these two cases greatly reduced the output power and the efficiency of the cycle. What’s more, the oil circulation needs a secondary heat exchanger, which increased the irreversible loss.

In this paper, a two-stage transcritical combined organic rankine cycle (CORC) is presented. In the combined system, the high and low temperature stages transcritical cycle recover the high grades waste heat, and medium to low grades waste heat respectively, and being combined. The suitable working fluids for high stage, together with the cycle parameters have been analyzed and optimized.

2. System description

2.1. Topping Diesel Engine

An inline 6 cylinder 4 stroke supercharged diesel engine is considered as a topping system. The engine is: the main parameters of the engine are presented in Table 1. The composition of the exhaust gases on the basis of mass has been calculated as: CO$_2$=9.1%, H$_2$O=7.4%, N$_2$=74.2%, O$_2$=9.3%.

| Parameter                          | values | Parameter                          | values |
|-----------------------------------|--------|-----------------------------------|--------|
| Electrical power output (kW)      | 243    | Rotate speed (r/min)              | 2200   |
| EGR temperature (°C)              | 590    | EGR rate                          | 0.2    |
| Exhaust temperature (°C)          | 445    | Exhaust mass flow (kg/h)          | 1020.74|
| Coolant outlet temperature (°C)   | 87     | Coolant inlet temperature (°C)    | 70     |
| Coolant mass flow (kg/min)        | 258.67 | CAC air mass flow (kg/min)        | 16.17  |
| CAC outlet temperature(°C)        | 162    | CAC inlet temperature(°C)         | 54     |

2.2. Bottoming CORC system

The proposed bottoming CORC system layout and its corresponding T-s diagrams (taking Siloxane2 for example of the high-temperature stage cycle and R143a for example of the low-temperature stage cycle) are shown in Figs. 1, 2a and 2b, respectively. The system is a transcritical combined ORC, consisting of combined expander, generator, gas heater, working fluid pump, regenerator, condensate heater (internal heat exchanger), preheater, condenser and other test facilities. The high-temperature stage ORC (HORC) is used to recover the high grades waste heat, and the low-temperature stage ORC (LORC) is used to recover all the medium and low grades waste heat and coupling with HORC. To the HORC (the red heavy line in Fig.1), the transcritical ORC system can be identified as 1-3-3g-4-6-5-2-1 (in Fig.2a) and described as follows. Working fluid, siloxane, flows into the gas heater 1 and is heated by the exhaust from DE(process 1-3). The heated siloxane flows into gas heater 2 where it is heated by the EGR waste heat(process 3-3g). After that, siloxane becomes supercritical fluid and flows into the expander. Its enthalpy is converted into work, and then converted into electricity by generator(process 3g-4). The process 3g to 4ideal means isentropic expansion, which is ideal and impossible. When the turbine efficiency is higher, the point 4 is closer to point 4ideal, which means the real expansion process in expander is closer to the isentropic expansion. The low temperature siloxane vapor exits from expander and flows into the regenerator which is used to preheat the...
siloxane before it flows into the gas heater (process 4-6). The low temperature vapor exiting from the regenerator flows into condensate heater where it is used to heat the working fluid of low-temperature stage and at the same, the vapor itself is liquefied and condensed into saturated liquid (process 6-5). The saturated liquid available at the condensate heater outlet is pumped to be high pressure liquid (process 5-2) by working fluid pump, and then flows into the regenerator where it is preheated before flowing into the gas heater (process 2-1). The preheated siloxane flows into gas heater to finish the high-temperature stage ORC.

![Figure 1](image)

Red heavy line — high-temperature stage; Blue heavy line — low-temperature stage

Figure 1. Scheme of two-stage transcritical combined ORC for DE

To the LORC. (the blue heavy line in Fig. 1), the transcritical ORC system can be identified as e-f-f \text{gm} \text{-} g-j-a-b-d-e (in Fig. 2b) and is as the same as HORC. During the heating process, working fluid, R143a, is heated by the coolant waste heat from DE (process e-f), by the siloxane steam of high-temperature stage (process f-f \text{gm} \text{-} g), by the CAC waste heat (process g-j), by the residual exhaust waste heat (process j-a) which has been used in high-temperature stage before, and finally by the residual low-temperature EGR waste heat (process a-k) which also has been used in high-temperature stage before. The low pressure R143a vapor exits from the expander and flows into the condenser where it is liquefied and condensed into saturated liquid (process b-d) by supplied cooling water.

3. Mathematical modeling
In this paper, the pinch point temperature difference method is used to analyze the supercritical heat transfer process. Before establishing the mathematical model of this system, some general assumptions are formulated as follows: (1) each component is considered as a steady-state steady-flow system; (2) the kinetic and potential energies are neglected; (3) heat and friction losses in each component and system pipe are also neglected; (4) isentropic efficiency of the turbine is 0.7; isentropic efficiency of the pump is 0.8; (5) condensing temperature of LORC is 35 ℃; the temperature of coolant back to engine is 70 ℃; the temperature of charger air flows into engine is 54 ℃; the ambient temperature is 25 ℃. The mathematical model of the system is described as Table 2.

Turbine:
\[ W_t = m_r \left( h_{t, in} - h_{t, out} \right); \eta_t = \frac{\left( h_{t, in} - h_{t, out} \right) - (h_{p, in} - h_{p, out})}{(h_{p, in} - h_{p, out})} \] (1)

Working fluid pump:
\[ W_p = m_r \left( h_{p, out} - h_{p, in} \right); \eta_p = \frac{\left( h_{p, out} - h_{p, in} \right) - (h_{p, out} - h_{p, in})}{(h_{p, out} - h_{p, in})} \] (2)

Condenser:
\[ Q_c = m_r (h_{r, in} - h_{r, out}) \]
\[ Q_c = c_p m_v (T_{cw, out} - h_{cw, in}) \] (3)

Regenerator:
\[ Q_{reg} = m_r (h_{r, abs, out} - h_{r, abs, in}) = m_r (h_{r, rej, in} - h_{r, rej, out}) \] (4)

Gas heaters:
\[ Q_{gh} = m_r (h_{r, out} - h_{r, in}) \]
\[ Q_{gh} = c_p m_v (T_{h, in} - h_{h, out}) \] (5)

Wherein, \( W_t \) means the output power of the turbine (kW); \( W_p \) means the input power of working fluid pump (kW); \( m \) means the mass flow rate of fluid (kg/s); \( h \) means the specific enthalpy (kJ/kg); \( \eta \) means the isentropic efficiency of turbine; \( \eta_p \) means the isentropic efficiency of working fluid pump; \( Q \) means the heat capacity in heat exchanger (kW); Subscripts t, p, r, gh, ch, c, reg, h, cw, in, out, s, high, low, abs and rej mean turbine, working fluid pump, working fluid, gas heater, condensate heater, condenser, regenerator, heat source, cooling water, inlet, outlet, isentropic process, high temperature stage, low temperature stage, absorbing heat process and rejecting heat process respectively.

The exergy analysis is based on the following method in this paper. For each component, all the exergy flow is considered. Take the gas heater 1 in high-temperature stage for example.

In Fig.3, exergy for each inlet and outlet can be calculated by the following formula:
\[ E_i = m_i [(h_i - T_0 s_i) - (h_i - T_0 s_i)] \] (6)

The energy loss for gas heater 1 is equal to the difference between the exergy flowing into the gas heater 1 and the exergy leaving the gas heater 1. Therefore, the exergy loss for gas heater 1 is:
\[ \Delta E_{gh} = E_{gh, in} - E_{gh, out} = m_i [(h_i, in - h_i, out) - (T_i, s_i, in - T_i, s_i, out)] \] (7)

Wherein, \( E \) means the value of the exergy for each point (kJ/s). Subscript i means each point; s means the specific entropy (kJ/kg.k); \( T_0 \) means the ambient temperature which is defined to be 25 ℃; \( h_i \) and \( s_i \) mean the value of specific enthalphy and specific entropy under standard condition which means the temperature and pressure are 25 ℃ and 1atm, respectively.

The exergy analysis of other components in the system can be evaluated according to the same model as above. No more description will be included below. Therefore, the net power output of the CORC system can be obtained as follows:
\[ W_{net} = (W_{t, high} + W_{r, low} - W_{p, high} - W_{p, low}) \] (9)

The thermal efficiency of CORC is obtained by equation.
\[ \eta_{th} = \frac{(W_{t, high} + W_{r, low} - W_{p, high} - W_{p, low})}{(Q_{exhaust} + Q_{EGR} + Q_{coolant} + Q_{CAC})} \] (10)

The exergy efficiency of CORC system can be obtained as follows:
\[ \eta_{ex, ex} = \frac{(W_{t, high} + W_{r, low} - \Delta E_n)}{(E_{exhaust} + E_{EGR} + E_{coolant} + E_{CAC})} \] (11)

The global efficiency of the DE-CORC system can be obtained as follows:
\[ \eta_{glo} = (W_{DE} + W_{net}) / (W_{DE} / \eta_{DE}) \] (12)

\( W_{DE} \) means the shaft power output for the DE and it is 243kW. \( \eta_{DE} \) means the efficiency of the DE, which is proposed to be 40%.
4. Working fluid selection
Siloxanes were considered as the suitable working fluids for high temperature ORC[12]. Instead of the full chemical names of the siloxanes we use the serial numbers as Siloxane1, Siloxane2 and Siloxane3. Therefore, these three siloxanes are taken into consideration as the high-temperature stage working fluids in the analysis. R143a is considered to be the low-temperature stage working fluid in the analysis. Table 2 shows physical property of fluids considered in this study.

| Substance   | M (kg/kmol) | Tcr (℃) | Pcr(MPa) | Fluid type |
|-------------|-------------|---------|----------|------------|
| R143a       | 84.04       | 72.71   | 3.761    | Isentropic |
| Siloxane1   | 162.38      | 245.55  | 1.939    | dry        |
| Siloxane2   | 296.62      | 313.34  | 1.332    | dry        |
| Siloxane3   | 236.53      | 290.94  | 1.415    | dry        |

5. Results and discussion
Fig.4 shows the thermal efficiency ($\eta_{th}$) of the CORC system at different $P_{max}$, when using different working fluids of high-temperature stage. It is obvious that with the increase of $P_{max}$, the $\eta_{th}$ for three working fluids, Siloxane1, Siloxane2 and Siloxane3, increase in the beginning, and then decrease. That means each working fluid has the optimal maximum pressure of high-temperature stage. That’s because when the $P_{max}$ increases, the turbine’s power output and pump’s energy consumption are both increasing, but the former has a faster increase before the optimum $P_{max}$ and slower increase after the optimum $P_{max}$. The optimal pressure for Siloxane1, Siloxane2 and Siloxane3 and their corresponding value of thermal efficiency are 12.0 MPa with 8.35%, 5.7MPa with 7.53%, 7.5 MPa with 7.68%.

Comparing to Siloxane2 and Siloxane3, Siloxane1 can get the higher thermal efficiency. However, higher maximum pressure is required, which means some components should be higher pressure resistance. In Fig. 5, the net power output of the CORC system is plotted for three different working fluids in the range of $P_{max}$. It can be noted that almost the same pattern presents as the Fig. 4. All the three working fluids have their maximum values of net power output corresponding to each optimal maximum pressure. Siloxane1 gets the maximum $P_{net}$ value of 37.71kW at the optimal $P_{max}$ of 12MPa. Siloxane2 gets the maximum $P_{net}$ value of 34.10kW at the optimal $P_{max}$ of 5.5MPa. Siloxane3 gets the maximum $P_{net}$ value of 34.74kW at the optimal $P_{max}$ of 7.5MPa. Also as the thermal efficiency
comparison, Siloxane1 can get a higher net power output. It is probably because its critical pressure is higher than the other two working fluids.

Fig. 6 displays the exergy efficiency (η\text{exg}) of the CORC system at different \( P_{\text{max}} \), referring to the three different working fluids. It shows that the trend of η\text{exg} is almost a straight line up with the increase of \( P_{\text{max}} \). Furthermore, Siloxane1 gets the minimum exergy efficiency comparing to Siloxane2 and Siloxane3 at the same \( P_{\text{max}} \), which is adverse to Siloxane1 in contrast to the previous comparison between Siloxane1 and Siloxane2, Siloxane3.

Fig. 7 reports the global thermal efficiency of DE-CORC combined systems (\( \eta_{\text{glo}} \)) at different \( P_{\text{max}} \). Three curves are not strict parabola shape. \( \eta_{\text{glo}} \) always gets the same value in some small ranges of \( P_{\text{max}} \). There is still maximum \( \eta_{\text{glo}} \) for each working fluid. Siloxane1 gets the maximum \( \eta_{\text{glo}} \) value of 46.21% in the range of 11-13MPa for \( P_{\text{max}} \). Siloxane2 gets the maximum \( \eta_{\text{glo}} \) value of 45.61% in the range of 4-7.5MPa for \( P_{\text{max}} \). Siloxane3 gets the maximum \( \eta_{\text{glo}} \) value of 45.72% in the range of 6.5-8.5MPa for \( P_{\text{max}} \). Therefore, when choosing the optimal \( P_{\text{max}} \), the relatively low \( P_{\text{max}} \) can be chosen to get the maximum \( \eta_{\text{glo}} \) for the DE-CORC combined system. Siloxane1 shows a higher global thermal efficiency, which has the same result as Fig. 5. Siloxane1 gets a better performance in comparison to the other two working fluids.

The combined cycle could recover the multiple grades fully. In this paper, we define the ratio of heat recovery as:

\[
\text{Ratio} = \frac{100 \times Q_{\text{recovered}}}{Q_{\text{max}}} \tag{13}
\]

wherein, \( Q_{\text{max}} \) means the maximum possible recovery heat, which are the heat releasing from inlet temperature to 120°C for of EGR and exhaust, the heat releasing from inlet temperature to 25°C for CAC and the heat releasing from inlet temperature to 70°C for coolant. Taking Siloxane2 system for example, table.4 shows the heat recovery ratio of the four parts waste heat at various \( P_{\text{max}} \). As shown, using the CORC could recover about 95.9% of EGR waste heat, 97.5% of exhaust waste heat, 78.9% of CAC waste heat and 100% of coolant waste heat.

**Figure 5.** Variation of \( P_{\text{net}} \) with \( P_{\text{max}} \)  

**Figure 6.** Variation of exergy efficiency (\( \eta_{\text{exg}} \)) with \( P_{\text{max}} \)

**Table 4.** Heat recovery ratio of four parts waste heat

| \( P_{\text{max}} \)(MPa) | Ratio\(_{\text{EGR}}\) (%) | Ratio\(_{\text{Exhaust}}\) (%) | Ratio\(_{\text{CAC}}\) (%) | Ratio\(_{\text{Coolant}}\) (%) |
|---------------------------|-----------------|-----------------|-----------------|-----------------|
| 4            | 95.9            | 97.6            | 78.96           | 100             |
| 4.857        | 95.91           | 97.55           | 78.96           | 100             |
| 5.714        | 95.91           | 97.52           | 78.96           | 100             |
| 6.571        | 95.92           | 97.5            | 78.96           | 100             |
| 7.429        | 95.92           | 97.48           | 78.96           | 100             |
| 8.286        | 95.91           | 97.47           | 78.96           | 100             |
| 9.143        | 95.91           | 97.46           | 78.96           | 100             |
| 10           | 95.91           | 97.45           | 78.96           | 100             |
6. Conclusions

In this paper, a two-stage transcritical combined organic rankine cycle (CORC) is presented and also be analyzed and optimized. Some conclusions can be obtained.

1) There exists an optimal maximum pressure of high-temperature stage to obtain the highest $\eta_{glo}$. Various working fluids have the variable optimal maximum pressure, such as 12MPa of Siloxane1, 5.7MPa of Siloxane2 and 7.5MPa of Siloxane3.

2) The CORC system could recover the multiple grades waste heat fully, about 95.9% of EGR waste heat, 97.5% of exhaust waste heat, 78.9% of CAC waste heat and 100% of coolant waste heat.

3) The global thermal efficiency of DE-CORC combined systems ($\eta_{glo}$) can get a value of 45.1%-46.2% assuming 40% thermal efficiency of the DE system without CORC bottoming system.

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References

[1] Teng H, Regner G and Cowland C 2007 Waste heat recovery of heavy-duty diesel engines by organic Rankine cycle Part II: working fluids for WHR-ORC SAE paper 01 543
[2] Vazaquez J, Zanz-Bobi M A, Palacios R, Arenas A 2002 State of the art of thermoelectric generators based on heat recovered from the exhaust gases of automobiles Proceedings of 7th European workshop on thermoelectrics(Pamplona, Spain, October 2002)
[3] Rody E C and Denis C 2005 Combined cycle for hybrid vehicles SAE paper 01 1171.
[4] Arias D A, Shedd, T A and Jester R K 2006 Theoretical analysis of waste heat recovery from an internal combustion engine in a hybrid vehicle SAE paper 01 1605
[5] Vaja I and Gambarotta A 2010 Internal combustion engine (ICE) bottoming with organic Rankine cycles (ORCs). Energy 35(2) 1084-93
[6] Kalina J 2011 Applied Thermal Engineering 31 2829-40
[7] Teng H, Regner G and Cowland C 2006 Achieving high engine efficiency for heavy-duty diesel engines by waste heat recovery using supercritical organic-fluid rankine cycle SAE Paper 01 3522
[8] Regner G, Teng H and Cowland C 2006 A quantum leap for heavy-duty truck engine efficiency-hybrid power system of diesel and WHR-ORC engines. The 12th Diesel Engine-Efficiency and Emissions Research Conference(Detroit, Michigan, 20-24 August 2006)
[9] Lai N A, Wendland M and Fischer J 2011 Energy 36(1) 199-211.