

# Thermodynamic Analysis of a Novel Combined Power and Cooling Cycle Driven by the Exhaust Heat Form a Diesel Engine

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## ABSTRACT

A novel combined power and cooling cycle based on the Organic Rankine Cycle (ORC) and the Compression Refrigeration Cycle (CRC) is proposed. The cycle can be driven by the exhaust heat form a diesel engine. In this combined cycle, ORC will translate the exhaust heat into power, and drive the compressor of CRC. The prime advantage of the combined cycle is that both the ORC and CRC are trans-critical cycles, and using CO<sub>2</sub> as working fluid. Natural, cheap, environmentally friendly, nontoxic and good heat transfer properties are some advantages of CO<sub>2</sub> as working fluid. In this paper, besides the basic combined cycle (ORC-CRC), another three novel cycles: ORC-CRC with an expander (ORC-CRCE), ORC with an internal heat exchanger as heat accumulator combined with CRC (ORCI-CRC), ORCI-CRCE, are analyzed and compared. The cycle parameters, including the coefficient of performance (cop), the cooling capacity ( $Q_{ro}$ ) and expansion power of CRC ( $W_e$ ) have been analyzed and optimized as the variation of the high pressure of ORC, the high pressure and the outlet temperature of gas cooler of CRC, and temperature drop of heat source in heat accumulator of ORC. The results indicate that there is an optimal high pressure of CRC (about 8.6MPa to 8.8MPa) for the combined cycles, at which the combined cycles achieve the optimal performance. The results also show that both the expander and heat accumulator could improve the system performance. The higher  $\Delta T_i$  could improve the system performance, but also resulting the more insufficiency of waste heat recovery.

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## INTRODUCTION

Energy is an important entity for the economic development of any country. The rapid industrial and economical growth in China where one fourth population of the world is present has increased the need for energy rapidly in the recent years. Considering the environmental protection and also in the context of great uncertainty over future energy supplies, attention is concentrated on the utilization of sustainable energy sources and the energy conservation methodologies. In China, Diesel Engine (DE) has consumed over 66 percent of overall fuel consumption. Since it is difficult for the maximum efficiency of DE 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, with a significant fraction through the exhaust. The

characteristics of energy balance distribution and waste energy grades of a heavy-duty diesel engine are shown in Table.1[2]. It can be seen from Table 1, the exhaust gets a high temperature up to 400 °C, and up to 21% percentage of fuel combustion energy. On average, about one third of energy generated from the fuel is wasted via exhaust gases.

**Table 1. Energy balance distribution and waste energy grade of a heavy-duty diesel engine**

| Energy form | Percentage of fuel combustion energy % | Temperature °C |
|-------------|----------------------------------------|----------------|
| Shaft work  | 40-42                                  | -              |
| exhaust     | 19-21                                  | 400            |
| coolant     | 17-19                                  | 90             |
| EGR         | 10-12                                  | 540            |
| CAC         | 8-10                                   | 170            |

Organic Rankine Cycle (ORC) could be used to recover waste heat from DE exhaust gas. Compared to steam rankine cycle on a same working condition, there are some advantages of ORC as follows. First, higher thermal efficiency and more net power are achieved by ORC. Second, organic fluids often have lower heat of vaporization and can better follow the heat source to be cooled than water at the same boiling temperature, thus reducing temperature differences and irreversibility at the evaporator, and therefore downsizing system volume and weight, which is significant for vehicle applications. Finally, turbines for organic cycles can provide higher efficiencies at part loads as well and are usually less complex due to the lower enthalpy drop of the fluid. For the ORC system using in DE waste heat recovery, there are several researches. Kane M et al. [3] presented the retrofit with two different ORC systems rated about 10 kW of a small engine with a power of 200 kW, fuelled with biogas. The paper focused on the application of scroll expander for its advantageous when the power of the ORC is limited. The feasibility of the system was supported by field tests conducted on an experimental unit. Danov SN and Gupta AK [4, 5] proposed a setup where a low speed two stroke marine turbocharged Diesel engine acted as the topper of a combined cycle, with exhaust gases used for a bottoming cycle based on a Rankine cycle. Vaja I.[6] considered the assessment of double cascade ORC designs in conjunction to stationary DEs, which would be presented in future works. Teng et al. [1,7] proposed a supercritical, reciprocating Rankine expander for waste heat recovery of diesel engines.

Nowadays, the supercritical Rankine cycle was presented for some advantages like: the heating process of a supercritical Rankine cycle does not pass through a distinct two-phase region like a conventional organic Rankine cycle, resulting in a better thermal match in the boiler with less irreversibility. Chen et al. [8, 9, 10] did a comparative study of the carbon dioxide supercritical power cycle and compared it with an organic Rankine cycle using R123 as the working fluid in a waste heat recovery application. Their work shows that a CO<sub>2</sub> supercritical power cycle has a higher system efficiency than an ORC when taking into account the heat transfer behavior between the heat source and the working fluid. Zhang et al. [11, 12, 13, 14, 15, 16, 17, 18] have also conducted research on the supercritical CO<sub>2</sub> power cycle. Their experiments revealed that the power generation efficiency was 8.78% to 9.45% [11].

Meanwhile, in the refrigerant history, CFCs have been washed out for high ozone depletion potential (ODP). Nowadays, the F Gases Regulations (EC842-2006 and EC40-2006) [19, 20, 21] are presented to limit and reduce emissions of fluorocarbons (HFCs and HCFCs) from refrigeration and heat pump plants for their high global warming potential (GWP). Carbon dioxide is considered as the most potential substitute refrigerant not only for its friendliness to environment, but also for its non-flammable, non-toxic, and good thermodynamic properties, especially after the trans-critical cycle being presented [22]. Theoretical

researches and analysis show that plenty expansion power can be recovered and the system efficiency can be equal to or even higher than that of the common refrigeration cycles by using an expander instead of the expansion valve in carbon dioxide transcritical cycle [23, 24, 25, 26].

In this paper, a novel combined power and cooling cycle based on the Organic Rankine Cycle (ORC) and the Compression Refrigeration Cycle (CRC) is proposed. The cycle can be driven by the exhaust heat from a diesel engine. In this combined cycle, ORC will translate the exhaust heat into power, and drive the compressor of CRC. The prime advantage of the combined cycle is that both the ORC and CRC are trans-critical cycles, and using CO<sub>2</sub> as working fluid. Natural, cheap, environmentally friendly, nontoxic and good heat transfer properties are some advantages of CO<sub>2</sub> as working fluid. In this paper, besides the basic combined cycle (ORC-CRC), another three novel cycles: ORC-CRC with an expander (ORC-CRCE), ORC with an internal heat exchanger as the heat accumulator combined with CRC (ORCI-CRC), ORCI-CRCE, are analyzed and compared. The cycle parameters, including the coefficient of performance (cop), the cooling capacity (Q<sub>ro</sub>) and expansion power of CRC (We) have also been analyzed and optimized as the variation of the high pressure of ORC, the high pressure and outlet temperature of gas cooler of CRC, and temperature drop of heat source in heat accumulator of ORC.

## SYSTEM DESCRIPTION

### Topping DE System

**Table 2. Main parameters of the commercial diesel generator set**

| Parameter                       | values |
|---------------------------------|--------|
| Electrical power output (kW)    | 235.8  |
| Torque (N.m)                    | 1500   |
| Exhaust temperature (°C)        | 519    |
| Smoke intensity (FSN)           | 0.55   |
| Rotate speed (rpm)              | 1501   |
| Fuel consumption (kg/h)         | 47.79  |
| Combustion air mass flow (kg/h) | 943    |
| Exhaust mass flow (kg/h)        | 990.79 |

In this study, a commercial diesel generator set is considered as a topping system. The engine is an inline 6 cylinder 4 stroke supercharged diesel oil fired engine: the main parameters of the engine are presented in Table 2. As the aim of this study analysis is to obtain the parameters optimization for the combined cycles use in exhaust heat recovery of DE, we assume the engine to operate at rated conditions. It has been calculated that the air fuel ratio is 19.75 and the excess air coefficient is 1.38 at the nominal condition, under the hypothesis of perfect combustion of diesel oil. By the way, excess air coefficient is defined as the ratio of real air consumption to theoretical value (14.3 for

diesel fuel). The composition of the exhaust gases on the basis of mass has been calculated as: CO<sub>2</sub>=15.10%, H<sub>2</sub>O=5.37%, N<sub>2</sub>=73.04%, O<sub>2</sub>=6.49%. This composition is used to evaluate the gas properties.

## Bottoming Combined Cycle

The proposed bottoming combined ORC-CRC system layouts and corresponding T-s diagram are shown in Fig.1a-d and Fig.2. The Fig.1a is the basic combined ORC-CRC system, while the others are the improved combined cycles. In Fig.1a, the ORC is a supercritical expansion ORC using carbon dioxide as working fluid, consists of a turbine (C), a condenser (D), a working fluid pump (E), a gas heater (B) and other test facilities. Hot exhaust from DE rejects heat in the gas heater, and then discharges to the atmosphere. The working process of ORC can be described as follows. The generated high pressure supercritical carbon dioxide fluid in the gas heater flows into the turbine and its enthalpy is converted into work (process 1-2), and then drives the compressor of CRC. Process 1-2s means the isentropic expansion, which is ideal and impossible. When the turbine efficiency is higher, the point 2 is closer to point 2s, which means the real expansion process in turbine is closer to the isentropic expansion. The turbine efficiency will be considered to real expansion when compared to isentropic process. The low pressure vapor exited from the turbine flows into the condenser where it is liquefied and condensed into saturated liquid (process 2-3). The saturated liquid available at the condenser outlet is pumped into high pressure liquid by working fluid pump (3-4), and then flows into the gas heater where it is heated into supercritical fluid by the exhaust (4-1). Process 3-4s means the isentropic pumping process.

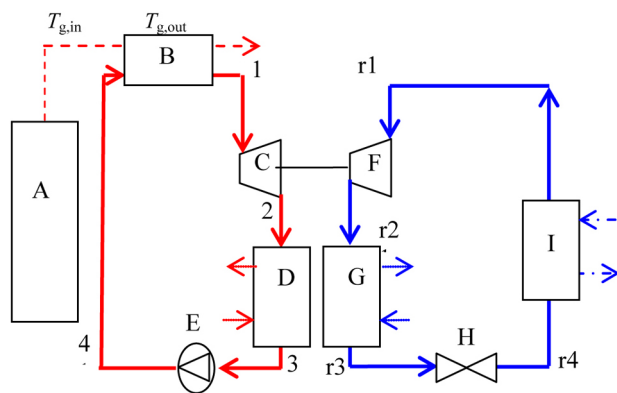


Fig.1a. schematic diagram of ORC-CRC system

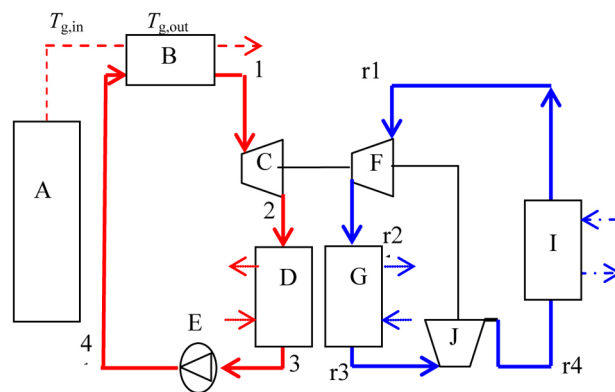


Fig.1b. schematic diagram of ORC-CRCE system

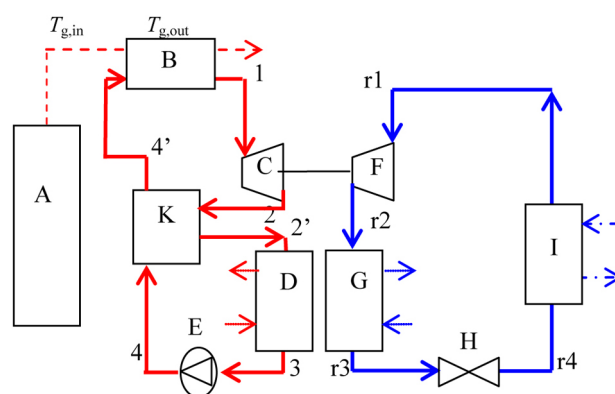


Fig.1c. Schematic diagram of ORCI-CRC system

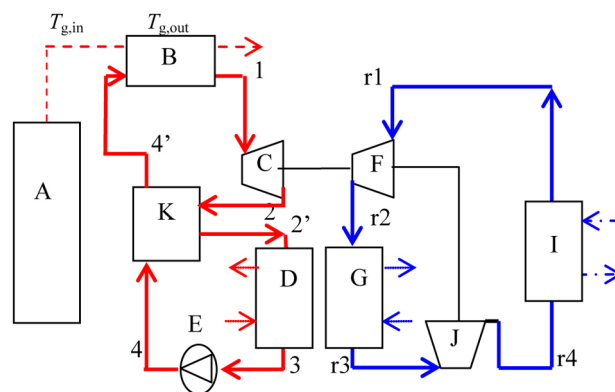


Fig.1d. Schematic diagram of ORCI-CRCE system

Red heavy line-ORC; Blue heavy line-CRC; Red dotted line-exhaust; Party line-ambient sink; Dot dash-cooling capacity

In fig.1a-d: A-DE; B-gas heater; C-turbine; D-condenser; E-working fluid pump; F-compressor; G-gas cooler; H-throttle vavle; I-evaporator; J-expander; K-heat accumulator

Meanwhile, the CRC is a trans-critical compression refrigeration system using carbon dioxide as working fluid, consists of a compressor (F), a gas cooler (G), a throttle valve (H), an evaporator (I) and other test facilities. The working process of CRC can be described as follows. The compressor driven by the ORC, compresses the saturated carbon dioxide vapor into a supercritical fluid (process r1-r2). Process r1-r2s means the isentropic compression, which is ideal and impossible. The supercritical fluid flows into gas cooler to rejects heat to cooling water from ambient sink (r2-r3), and then flows into the throttle valve (process r3-r4 with dotted line). The fluid output from throttle valve is a two-phase fluid, and flows into the evaporator to absorb the heat from the air-conditioned space (r4-r1). Finally, the fluid from evaporator becomes the saturated carbon dioxide vapor, and flows into compressor. In order to improve the performance of combined cycle, another three combined cycles are also proposed and compared. As shown in [fig.1b-d](#), based on the basic ORC-CRC system, the ORC-CRCE system using an expander (J) instead of throttle valve to recover the expansion power of throttle process (process r3-r4 with solid line), the ORCI-CRC system using a heat accumulator (K) to recover vapor heat exhausted from turbine (process 2-2' and process 4-4'), the ORCI-CRCE system using both the expander and the heat accumulator.

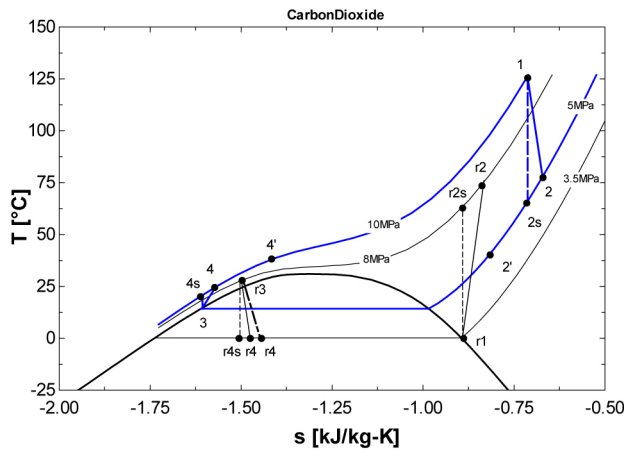


Fig.2. T-s diagram of four combined cycles

## Consideration of Working Fluid Heating Process

In this paper, the pinch point temperature difference method is used to analyze the supercritical heat exchanging process. [Fig.3](#) can give the heat exchange diagram during the working fluid heating process in gas heater. The heating source can be exhaust and cooling water.

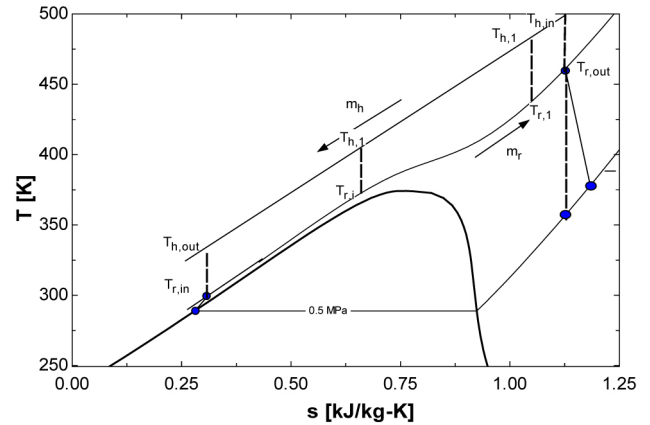


Fig.3. Schematic diagram of supercritical heat exchanging process

The heat source temperature ( $T_{h,in}$ ) and mass flow rate ( $m_h$ ) are tested and constant. The highest temperature of working fluid ( $T_{r,out}$ ) is consumed to be a constant value. A minimum pinch point temperature difference ( $T_{pp}$ ) at Pinch Point (PP) is considered to be 30°C or 5°C to meet the gas-liquid or liquid-liquid heat exchanger performances. The heat exchange between working fluid and heat source can be considered as follows:

Step 1, the PP is consumed at the outlet of heat source, which means  $T_{h,out} = T_{r,in} + T_{pp}$ , and this allows writing a first energy balance referred to complete the whole process:

$$m_{r,1} = \frac{m_h c_{p,h,av} (T_{h,in} - T_{h,out})}{h_{r,out} - h_{r,in}} \quad (1)$$

Wherein, the organic fluid enthalpies are obtained at the operating pressure chosen for the system and  $c_{p,h,av}$  is calculated according to the heat source property at the average temperature between  $T_{h,in}$  and  $T_{h,out}$ .

Step 2, divide the temperature difference between  $T_{h,in}$  and  $T_{h,out}$  into 100 points averagely and name each point. The heat source and corresponding working fluid temperature at each point can be obtained as followed energy balance:

$$T_{r,1} = T_{r,out} - \frac{m_h c_{p,h,av} (T_{h,in} - T_{h,1})}{m_{r,1} c_{p,r,av}} \quad (2)$$

Wherein, the  $c_{p,r,av}$  is calculated according to the working fluid property at the average temperature between  $T_{r,in}$  and  $T_{r,out}$ . Therefore, the temperature differences between heat source and working fluid at each point can be obtained as follows:

$$\Delta T_0 = T_{h,in} - T_{r,out}$$