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Ideal Point Design and Operation of CO_2-Based Transcritical Rankine Cycle (CTRC) System Based on High Utilization of Engine's Waste Heats

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Abstract: This research conducted a study specially to systematically analyze combined recovery of exhaust gas and engine coolant and related influence mechanism, including a detailed theoretical study and an assistant experimental study. In this research, CO_2-based transcritical Rankine cycle (CTRC) was used for fully combining the wastes heats. The main objective of theoretical research was to search an 'ideal point' of the recovery system and related influence mechanism, which was defined as operating condition of complete recovery of two waste heats. The theoretical methodology of this study could also provide a design reference for effective combined recovery of two or multiple waste heats in other fields. Based on a kW-class preheated CTRC prototype that was designed by the 'ideal point' method, an experimental study was conducted to verify combined utilization degree of two engine waste heats by the CTRC system. The operating results showed that the prototype can gain 44.4–49.8 kW and 22.7–26.7 kW heat absorption from exhaust gas and engine coolant, respectively. To direct practical operation, an experimental optimization work on the operating process was conducted for complete recovery of engine coolant exactly, which avoided deficient or excessive recovery.

Keywords: CO_2-based transcritical Rankine cycle (CTRC); ideal point; exhaust gas; engine coolant; waste heat recovery

1. Introduction

Currently, engine waste heat recovery (E-WHR) is attracting increasing interest, which is an effective means to improve fuel utilization efficiency of engines. Among engine waste heats, exhaust gas and engine coolant are the main carriers. Exhaust gas is a high-grade waste heat source, which is generally regarded as the first recovery target. In Wang's research on E-WHR, a diesel engine and a gasoline engine are selected as the recovery objects, whose exhaust temperature approaches over 700 °C and over 400 °C, respectively [1]. While, engine coolant is a relative low grade waste heat, whose temperature is below 100 °C, but still significant due to the comparative amount of waste heat [2]. Using a thermodynamic cycle to generate extra power is a high-efficiency way among the technologies of E-WHR, mainly including single-loop organic Rankine cycle (ORC) [3,4], dual-loop ORC [5,6], steam Rankine cycle [7,8], CO_2-based transcritical Rankine cycle (CTRC). Moreover, thermodynamic cycle is a feasible scheme to make a combined recovery of exhaust gas and engine coolant. Engine coolant can be used as a preheating source based on the basic use of exhaust gas.

CTRC is a suitable technology used for E-WHR due to well environmental performance, small system size and stable chemical property in direct heat transfer with exhaust gas [9,10]. Compared with another general-used power cycle, ORC, the CTRC does not need intermediate systems and consists of only one heat exchanger for the evaporation process, which increases the system's exergy efficiency [11]. Meanwhile, owning to the good flow and heat transfer properties of supercritical CO_2, CTRC system has a superior heat transfer performance compared with ORCs [12,13], which can obtain smaller heat exchangers and make the plant more compact.

Both theoretical and experimental investigations on CTRC for E-WHR were performed in recent years. Theoretical researches mainly focus on the comparison and selection of the system configuration. Chen et al. [14] compared different CO_2 bottoming systems for a vehicle engine, including a CTRC, a CO_2 Brayton cycle and a CO_2 combined power and cooling cycle. Their results indicated that the CTRC was more suitable than the others. Farzaneh-Gord et al. [15] compared four CTRC configurations and presented that the CTRC with a regenerator and preheater obtains the maximum net power output, and the CTRC with a regenerator achieves the highest thermal efficiency. Kim et al. [16] used a cascade cycle and split cycle to optimize CTRC performance, and the results showed that a split cycle was the better choice that can produce the highest power over a wide range of operating conditions. Shu et al. [17] proposed three configuration selection maps based on the best output performance (net power output), the best exergy performance (exergy efficiency) and the best economic performance (electricity production cost). They are used for scheme selection for the CTRC design and application based on different target needs and different operating conditions. Moreover, there are few experimental researches on the CTRC, and most testing research are not based on real engines. A CTRC supplier in American, Echogen Power Systems (EPS) company, provided the preliminary testing results of a 250 kW system, which showed the potential for the exhaust waste heat recovery in a temperature range of 200–540 °C [18,19]. But the heat source in the testing process used steam instead of exhaust gas. Pan et al. [20] conducted a test bench of CTRC with an oil heat source. Due to only 21.4% isentropic efficiency of piston type expander, the thermal efficiency of system is relative low about 5.0% with expansion inlet and outlet pressure of about 11 MPa and 4.6 MPa, respectively. After complete the research of selection maps [17], Shu's group continued to test practical performance of corresponding CTRC configurations based on a small-scale CTRC bench [21], the results indicates that the CTRC with a preheater and regenerator can improved the engine efficiency from 39.4 to 41.4%.

While, the main focus of most researches on the CTRC is the system performance (e.g., power output, thermal efficiency), the utilization rate of waste heat is rarely regarded as a special subject but really significant. Higher utilization rate of exhaust gas and engine coolant mean more energy input for the CTRC, then following produce more power output. Moreover, higher utilization rate of engine coolant will make low coolant temperature flow into cooling system of engine, which can reduce cooling load of fan or cooling tower. For systems only with exhaust gas recovery, the outlet temperature of exhaust gas in gas heater determines utilization rate of exhaust gas, which depends on several factors mainly involving the cycle form (subcritical cycle or transcritical cycle), temperature difference of pinch point and acid dew point temperature of exhaust gas [22]. Due to low inlet temperature of working fluid in gas heater (<50 °C), outlet temperature of exhaust gas has no restriction of pinch point. Hence, complete use of exhaust gas can be obtained only requiring a certain mass flow of working fluid.

Compared with the systems only recovering exhaust gas, systems combining exhaust gas and engine coolant have a more complex influence mechanism of utilization rate for various type waste heats. Except for the respective factors, the combined capability of exhaust gas and engine coolant takes an important role for their recovery rate. Well combined capability makes both complete utilization of exhaust gas and engine coolant. Different fluid has different combined capability of exhaust gas and engine coolant, which only has scattered results in the previous researches. Vaja et al. [23] calculated the preheated ORCs with working fluids of Benzene, R11 and R134a, which achieve only 0.175, 0.242 and 0.579 utilization rate of engine coolant, based on good utilization of exhaust gas waste heat. It seems to be difficult to recovery waste heat effectively available from the engine coolant by ORCs. Kim et al. [24] compared different preheat regions of subcritical ORCs by engine coolant. The results showed that 40% waste heat of the engine coolant is used for preheating the liquid region of evaporation process. The utilization rate would increase to 90% if engine coolant was used to heat the two-phase of evaporation process, but required an extremely low turbine inlet temperature below 100 °C. While, system with or without preheating process by engine coolant had little effect on the utilization rate of exhaust gas, which was always high over 75%. Therefore, for most organic fluids, they cannot combine exhaust gas and engine coolant well, usually weak for engine coolant. Comparatively, the research on the combined capability of preheated CTRC is rarer. Farzaneh-Gord et al. [15] conducted the preheated CTRC but only cared the system performance. In an earlier literature by Kim et al. [25], results indicated that the CTRC has a significant performance improvement by combined low-temperature and high-temperature heat source. The CTRC seemed to own better combined capability of exhaust gas and engine coolant, but this has not been concretely introduced in these researches.

For fully combining recovery of exhaust gas and engine coolant, this research conducted a study on the CTRC system. Instead of traditional thermodynamic performance, the combined capability of these two waste heats and corresponding influence mechanism were the main focus of this research. The main research object was to find an "ideal point" of the recovery system, which was defined as operating condition of complete recovery of two waste heats. The analysis and discussion were conducted from the angle of physical property of fluid, and divided into three parts considering the CTRC with or without regenerator, system performance at 'ideal point' and comparison with other organic fluids. Finally, an experimental study was conducted to verify utilization degree of exhaust gas and engine coolant based on a kW-class preheated CTRC prototype, as well as an operation optimization for exact recovery of engine coolant.

2. System Description and Modeling

Figure 1 shows the system structure and t-s diagram of the CTRC with a preheater (P-CTRC). A gasoline engine is selected as recovery target in this study, whose rated operating parameters are listed in Table 1. Combined with basic use of high-grade exhaust gas (777 °C), the low-grade engine coolant (89.4 °C) is used as a preheat source. The main process of the P-CTRC includes: heating process (1-2-3), expanding process, condensing process (4-5) and pumping process (5-1). Each process is in a stable state and neglects heat loss and pressure drop in pipes. The other assumptions for the components are listed in Table 2 [14,17,26].

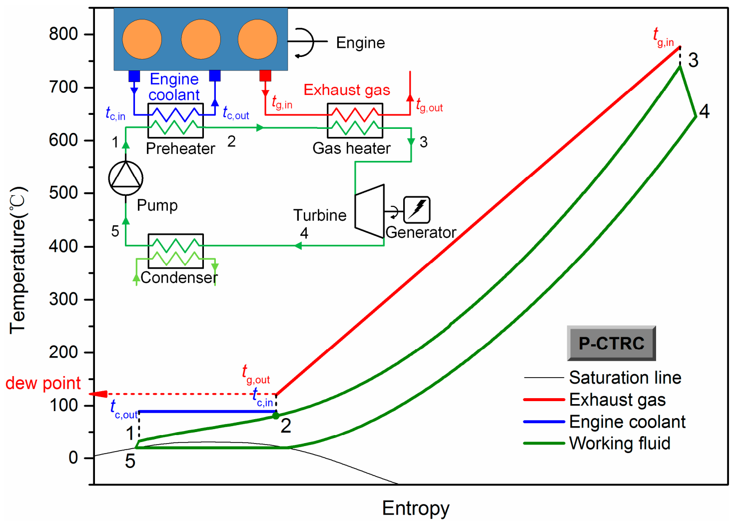


Figure 1. System structure and t-s diagram of the P-CTRC.

The property of exhaust gas is based on the mass fraction of compositions with 19.84% CO_2, 8.26% H_2O, and 71.49% N_2 [1]. The acid dew point temperature of exhaust gas is 120 °C and set as the minimum temperature of exhaust gas during simulating process [23]. The engine coolant is assumed as pure water. To meet the requirement of suitable coolant temperature into engine jacket, the minimum temperature of engine coolant cannot be lower than 85.3 °C.

Table 1. Rated operating parameters of the gasoline engine.

Parameter (Units)	Values
Engine Power (kW)/Speed (r/min)/Torque (N·m)	43.8/5000/81.3
Fuel consumption rate (kg/h)	11.4
Temperature (°C)/Mass flow rate (kg/h) of exhaust gas	777/202.6
Returned/Outlet temperature of engine coolant (°C)	89.4/85.3
Volume flow rate of engine coolant (L/min)	133.8

Table 2. Main parameters of CTRC design [14,17,26].

Parameters (Units)	Values
Δt_{pp} in the gas heater (°C)	30
Δt_{pp} in the preheater (°C)	5
Δt_{pp} in the regenerator (°C)	15
Condensation temperature (°C)	25
Turbine efficiency	0.7
Generator efficiency	0.9
Pump efficiency	0.8

Calculation is based on the MATLAB (2010) software and REFPROP (9.0) software. The calculating equations are illustrated as following:

For the pump:

$$W_{pump} = m_f \cdot (h_1 - h_5) = m_f \cdot (h_{1s} - h_5) / \eta_{pump} \tag{1}$$

where, the subscript "1*s*" means the exit state of pump after isentropic compression, and the subscript "*f*" represents CO_2 fluid.

For preheater:

$$Q_{preh} = m_f \cdot (h_2 - h_1) = m_c \cdot cp_c \cdot (t_{c,in} - t_{c,out}) \tag{2}$$

where, the subscript "*c*" represents engine coolant, the subscript "*c,in*" and "*c,out*" mean the inlet and outlet state of engine coolant in preheater, respectively.

For the gas heater:

$$Q_{gh} = m_g \cdot cp_g \cdot (t_{g,in} - t_{g,out}) = m_f \cdot (h_3 - h_2) \tag{3}$$

where, the subscripts "*g,in*" and "*g,out*" represent the inlet and exit state of exhaust gas in gas heater, respectively.

For the turbine and generator:

$$Power = m_f \cdot (h_3 - h_4) \cdot \eta_{gen} = m_f \cdot (h_3 - h_{4s}) \cdot \eta_{turb} \cdot \eta_{gen} \tag{4}$$

where, the subscript "4*s*" means the exit state of turbine after isentropic expansion.

For the condenser:

$$Q_{cond} = m_f \cdot (h_4 - h_5) \tag{5}$$

Net power output of the system is calculated as:

$$W_{net} = Power - W_{pump} \tag{6}$$

Total heat absorption of the system is calculated as:

$$Q_{tha} = Q_{gh} + Q_{preh} \tag{7}$$

Thermal efficiency of the system is calculated as:

$$\eta_{th} = \frac{W_{net}}{Q_{tha}} \tag{8}$$

The efficiency of turbine, pump and generator are listed in Table 2. During the calculation by the models above, how to calculate mass flow rate of CO_2 (m_f) is the key step for the all calculating process. After getting the m_f, each state of CO_2 are determined. Pinch Point Temperature Difference (PPTD) method is usually used for calculation of m_f [22,23], which was ever used for previous calculation on transcritical cycle in our group [22]. Hence, two mass flow rates are got when the system has two complete recovery processes, represented as following:

$$m_{f,1} = \frac{m_c \cdot cp_c \cdot (t_{c,in} - t_{c,out,lowest})}{(h_2 - h_1)} \tag{9}$$

$$m_{f,2} = \frac{m_g \cdot cp_g \cdot (t_{g,in} - t_{g,out,lowest})}{(h_3 - h_2)} \tag{10}$$

where, the $t_{c,out,lowest}$ is the lowest outlet temperature of engine coolant and refers to the returned temperature of engine coolant, which is much higher than t_1 without the restriction of PPTD; the $t_{g,out,lowest}$ is the lowest outlet temperature of exhaust gas. Therefore, $m_{f,1}$ and $m_{f,2}$ presents the maximum m_f produced by utilization of exhaust gas and engine coolant, respectively. If $m_{f,1}$ is equal to $m_{f,2}$, this mass flow rate is just to satisfied complete utilization of both exhaust gas and engine coolant. Otherwise, choose the minimum one of $m_{f,1}$ and $m_{f,2}$ as final mass flow rate to satisfy complete utilization of one waste heat.

In this paper, utilization rate of waste heat is associated with the ratio between recovery energy from waste heat and the available energy of waste heat. The available energy of waste heat is not the energy versus to an environment state, but maximum energy at a feasible condition. For exhaust gas, complete utilization degree is defined as gas temperature reduces to acid dew point. For engine coolant, complete utilization degree is defined as coolant temperature reduces to returned temperature by the CTRC. Thus, utilization rate of exhaust gas and engine coolant are respectively defined as:

$$U_g = \frac{m_g \cdot cp_g \cdot (t_{g,in} - t_{g,out})}{m_g \cdot cp_g \cdot (t_{g,in} - t_{g,dew})} \tag{11}$$

$$U_c = \frac{m_c \cdot cp_c \cdot (t_{c,in} - t_{c,out})}{m_c \cdot cp_c \cdot (t_{c,in} - t_{c,returned})} \tag{12}$$

where, $t_{g,dew}$, and $t_{c,returned}$ mean the acid dew point temperature of exhaust gas and the engine coolant temperature back to engine.

Detailed calculation strategy is shown in Figure 2, which is illustrated by the block diagram, helping readers easily understand the calculation process.

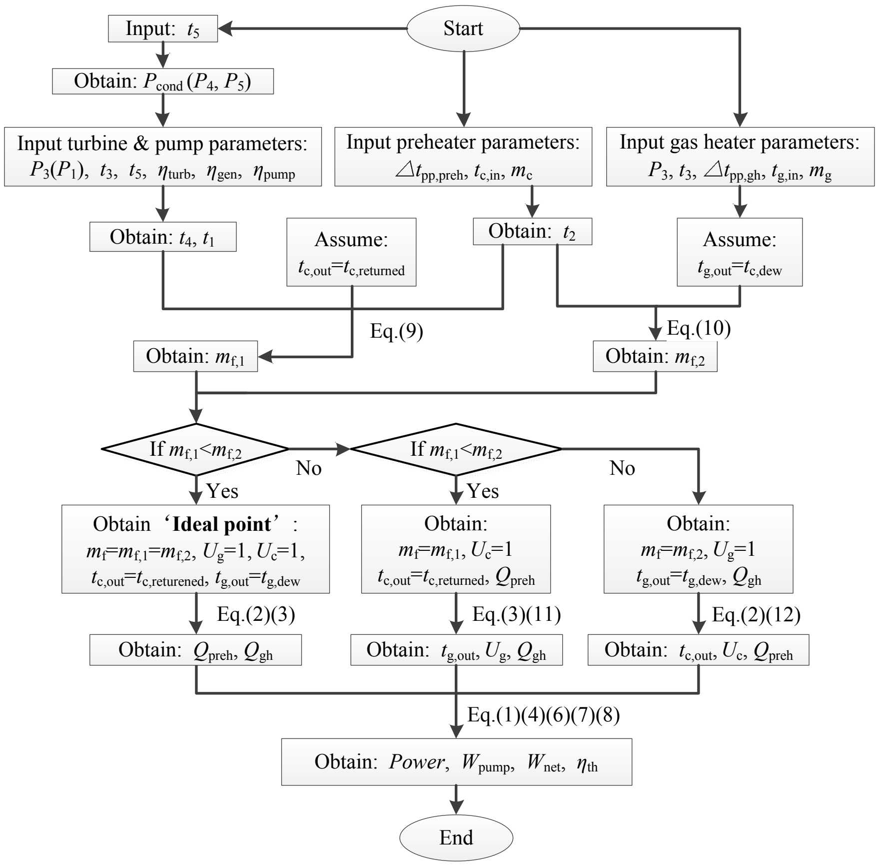


Figure 2. Block diagram of calculation strategy.

Modeling verification of this study has been done in the previous study of the CTRC [17] conducted by our group, which used the same mathematical models and the same system layout.

3. Theoretical Results and Discussion

3.1. "Ideal Point" Analysis of CTRC

Figure 3 shows the variations of utilization rate of waste heats in the P-CTRC at P_3 = 10 MPa and P_3 = 15 MPa, red lines for the exhaust gas (U_g) and blue lines for the engine coolant (U_c). U_g increases then becomes 1 with an increase in t_3, and U_c is 1 at low t_3 then reduces with an increase in t_3. It is because higher t_3 needs more waste heat of exhaust gas, and more waste heat of engine coolant is utilized at the condition of lower t_3.

A desired result appears in Figure 3 that, the change point of U_c from "1" to "not 1", and the change point of U_g from "not 1" to "1", occurs at the same t_3. It means both complete recovery of exhaust gas and engine coolant will achieve when turbine inlet temperature operates at this t_3. This research called it as "ideal point", represented as t_{ip}, which means the ideal condition where both complete utilization of exhaust gas and engine coolant are obtained. The t_{ip} changes with the variation of turbine inlet pressure. The t_{ip} is 264 °C at P_3 = 10 MPa, and 175 °C at P_3 = 15 MPa.

For introducing the producing mechanism of 'ideal point', the discussion about the CO_2 combined capability between exhaust gas and engine coolant is required from the perspective of the CO_2 property. As shown in Figure 4a,b, the bold black line means the *cp* change of CO_2 with the CO_2 temperature at pressure of 10 MPa and 15 MPa, respectively. The *cp* curves have a peak when the CO_2 fluid is at supercritical pressure. The corresponding temperature is called the pseudo-critical temperature. The peak value of *cp* and corresponding pseudo-critical temperature changes with pressure. The *cp* peak value of 10 MPa is larger than that of 15 MPa, and the pseudo-critical temperature is lower. The enclosed area between the *cp* curve and *X* axis represents the *cp* integration value with temperature, so its size represents the amount of heat absorption. In Figure 4a,b, t_1 and t_2

(84.4 °C) respectively represent the inlet and outlet temperature of CO_2 in the preheater So the blue area corresponds to heat recovery per mass from the engine coolant. And the temperature range at the right side of t_2 is for the exhaust gas, the red area means the heat recovery per mass from the exhaust gas at the condition of $t_3 = t_{ip}$. When the turbine inlet temperature is at the 'ideal point', the ratio between the blue and red area is just equal to the ratio between the maximum available energy of engine coolant and exhaust gas, which is called the 'complete recovery ratio'. This means both complete recovery of exhaust gas and engine coolant can be achieved when meeting a suitable mass flow rate. If t_3 is lower than t_{ip}, the engine coolant can get a complete use but the exhaust gas cannot, the blue area is bigger than the red area in this situation. The contrast performance occurs if $t_3 > t_{ip}$. Therefore, it results in the special U_g and U_c performance of "ideal point". When comparing the performance of 10 MPa and 15 MPa, it can be found that the smaller *cp* peak value of 15MPa causes smaller blue area relatively. Hence, the lower t_{ip} is required to meet the ratio between blue and red area equal to the complete recovery ratio. This is the reason why the t_{ip} of 15 MPa is lower than that of 10 MPa.

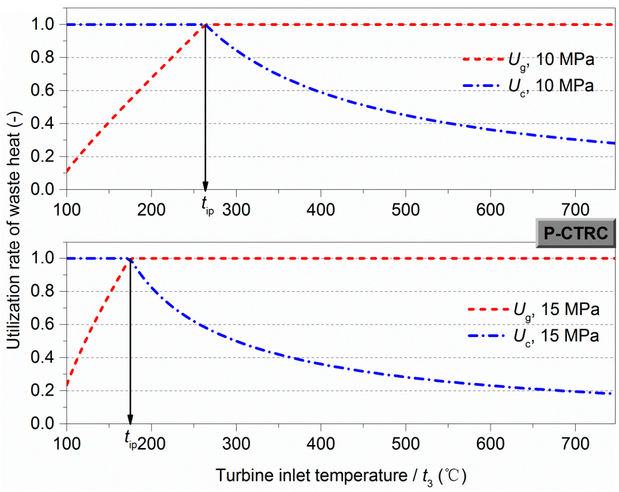
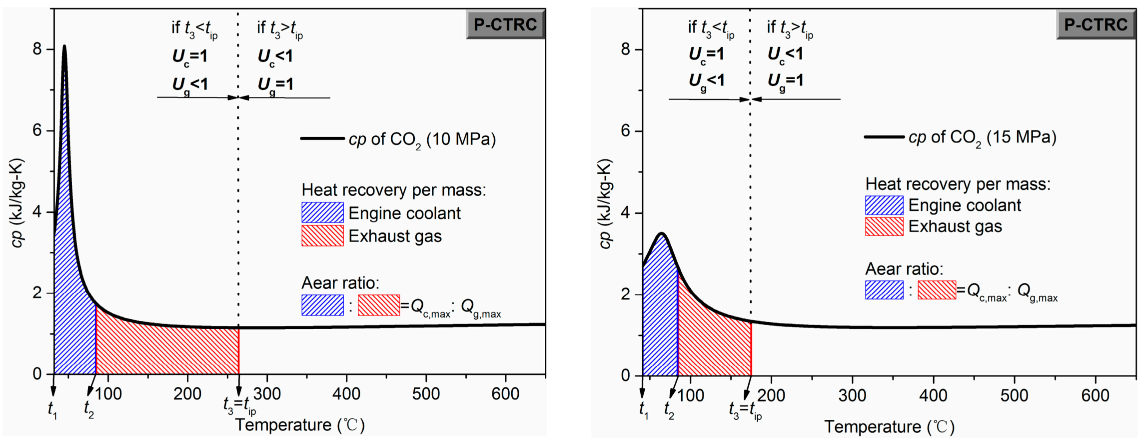


Figure 3. Variation of utilization rate of waste heats with turbine inlet conditions (P_3, t_3) for the P-CTRC.



(**a**) (**b**)

Figure 4. Heat absorption capacity from exhaust gas and engine coolant at the condition of 'ideal point' for the P-CTRC: (**a**) P_3 = 10 MPa; (**b**) P_3 = 15 MPa.

In the previous literatures about the CTRC, adding a regenerator is a usual method to improve the CTRC performance [27,28]. Therefore, it is also significant to analyze the change of "ideal point" when adding a regenerator. The CTRC system with a preheater and a regenerator is called PR-CTRC in this paper, whose system structure and t-s diagram display in Figure 5. The fluid flows into the regenerator (2–7) after preheated in the preheater and is heated by turbine outlet fluid (4–6). Then it flows into the gas heater. It is essential to conduct an analysis considering the effect of regenerator, as the regenerator has impact on the combined recovery of exhaust gas and engine coolant. The number of heating process become three when adding a regenerator, which is two previously. In this paper, the temperature difference of pinch point is set to 15 °C, and occurs between the state 2 and 6. When $t_4 - t_2 \leq 15$ °C, it means no regenerated condition for the system.

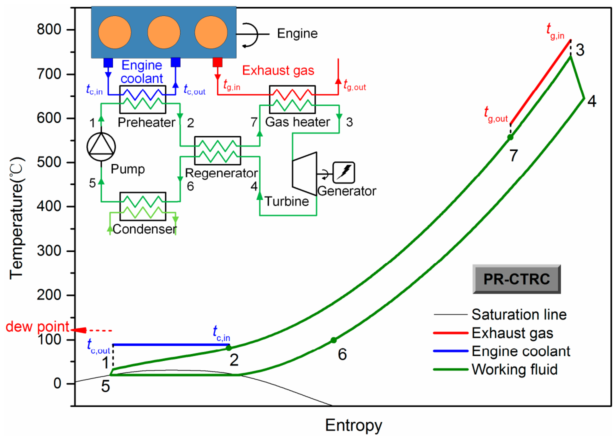


Figure 5. System structure and t-s diagram of the PR-CTRC.

Figure 6 presents the utilization rate of waste heats in the PR-CTRC. The exhaust gas doesn't obtain a complete use after adding the regenerator, in which the regenerated heat recovers a part of application temperature range of exhaust gas. But a peak value of U_g ($U_{g,max}$) occurs simultaneously with U_{ec} changing from "1" to "not 1". Thus, "ideal point" is also used for the PR-CTRC to represent the maximum recovery of exhaust gas and complete recovery of engine coolant. The t_{ip} of the PR-CTRC is 556 °C at P_3 = 10 MPa, and 220 °C at P_3 = 15 MPa.

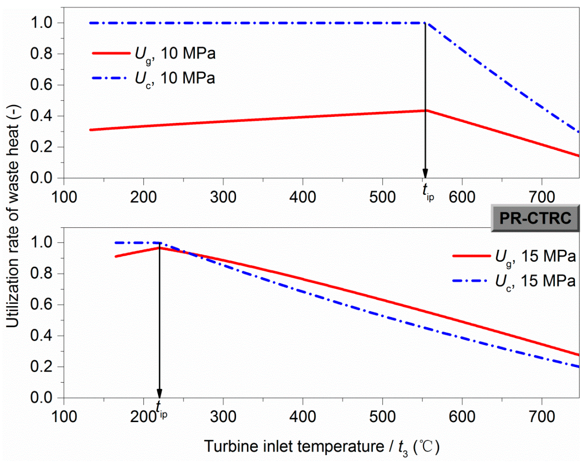
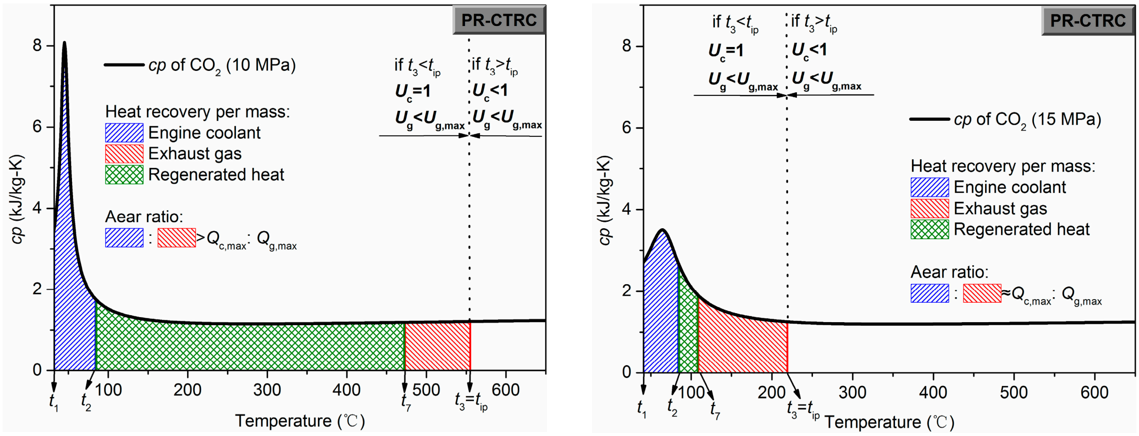


Figure 6. Variation of utilization rate of wastes heats with turbine inlet conditions (P_3, t_3) for the PR-CTRC.

Therefore, the heating process in the regenerator is added to *cp* figure, shown as the green area in Figure 7a,b. Similarly, these two figures show the heat absorption capacity of working fluids at the "ideal point" of the PR-CTRC. As shown in Figure 7a, due to a low pressure ratio when P_3 is 10 MPa, a high turbine outlet temperature is produced, further causing a large temperature range of the regenerated process in the regenerator. This results in that the ratio between the blue and red area is far less than the "complete recovery ratio". Hence, the $U_{g,max}$ at P_3 = 10 MPa is 0.42, much less than 1. Differently, when P_3 is 15 MPa, regenerated heat has little impact on the application temperature range of exhaust gas. The ratio between the blue and red area is still close to the "complete recovery ratio" (Figure 7b), following causes a high $U_{g,max}$ (0.97). Therefore, from the perspective of sufficient use of energy, the PR-CTRC is suitable operating at the "ideal point" of 15 MPa versus to that of 10 MPa.



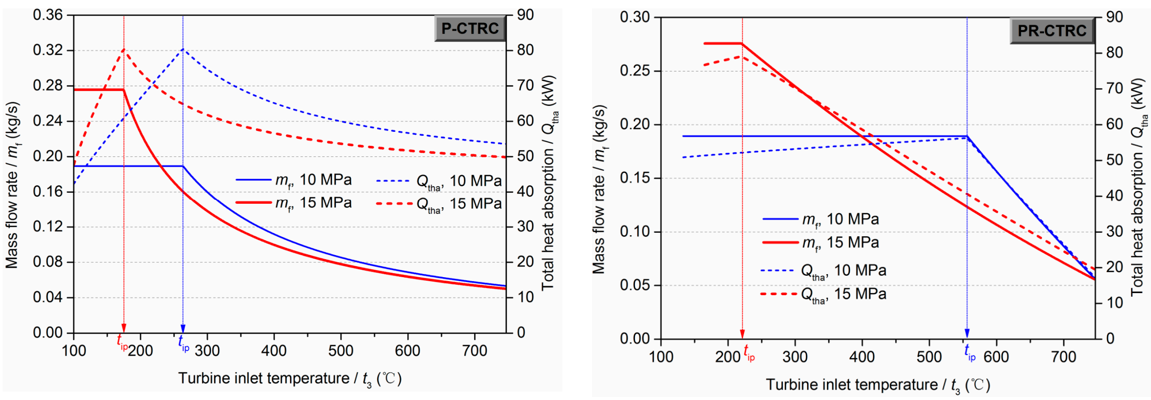
(**a**) (**b**)

Figure 7. Heat absorption capacity from exhaust gas and engine at the condition of "ideal point" for the PR-CTRC: (**a**) P_3 = 10 MPa; (**b**) P_3 = 15 MPa.

3.2. System Performance at "Ideal Point"

Although high utilization rate of exhaust gas and engine coolant are beneficial, output and efficiency performance are still the main pursue in a recovery system. Whether the best performance could be obtained at the "ideal point" is a worth studying. Therefore, performance of net power output (W_{net}) and thermal efficiency (η_{th}) are taken into account in this section, as well as two important parameters, mass flow rate (m_f) and total heat absorption (Q_{tha}).

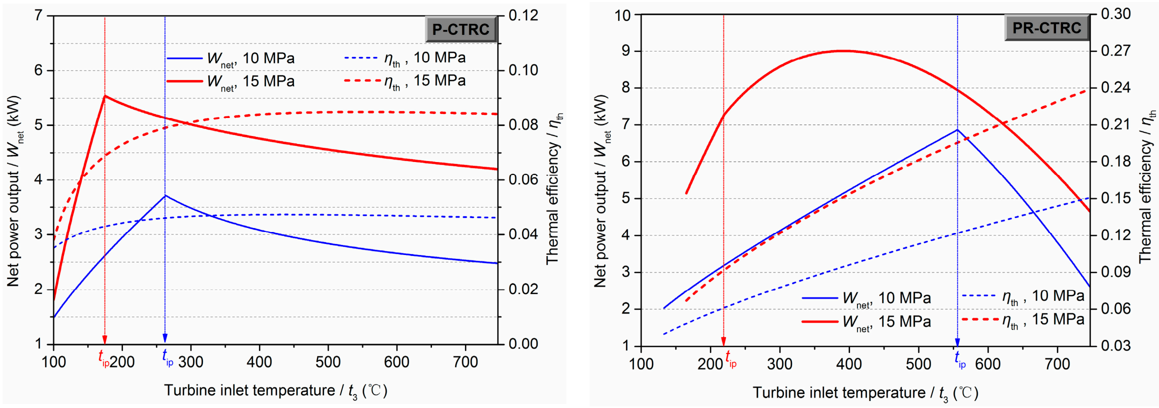
The mass flow rate variation trend with t_3 is shown in Figure 8. As shown in the mathematics models (Equations (9) and (10)), the mass flow rate is obtained by the comparison between the complete utilization of exhaust gas and engine coolant, respectively. Hence, the m_f is determined as a constant value by the constant energy of engine coolant at low t_3 in the P-CTRC. At high t_3 region, the m_f decreases with an increase in outlet temperature of fluid (t_3). Therefore, there is a transition point in the mass flow rate variation line that corresponding to the "ideal point". Figure 8 also shows the variation of total heat absorption. Maximum total heat absorption is achieved at the "ideal point". This is because "ideal point" means the highest U_g and U_c, following producing maximum energy input.



(**a**) (**b**)

Figure 8. Variation of mass flow rate of working fluid and total heat absorption with turbine inlet conditions (P_3, t_3): (**a**) P-CTRC; (**b**) PR-CTRC.

Although the "ideal point" has maximum mass flow rate and total heat absorption, it does not mean the maximum net power output as shown in Figure 9. There is a peak value existing in the variation line of net power output for the P-CTRC and the PR-CTRC. The corresponding turbine inlet temperature at the peak is identical with t_{ip} for the P-CTRC at 10, 15 MPa (Figure 9a) and the PR-CTRC at 10MPa (Figure 9b). But for the PR-CTRC at 15MPa (Figure 9b), the net power output variation trend only has change of increase rate and does not represent the maximum net power output. With a tighter connection with turbine inlet temperature, the highest thermal efficiency appears at high turbine inlet temperature range as expected, which has no link to the "ideal point".



(**a**) (**b**)

Figure 9. Variation of net power output and thermal efficiency with turbine inlet conditions (P_3, t_3): (**a**) P-CTRC; (**b**) PR-CTRC.

The analysis discussed above focuses on the performance of 'ideal point' at a certain turbine inlet pressure. A discussion on various performance of 'ideal point' at various turbine inlet pressures is conducted following.

After calculating under the P_3 from 10 MPa to 20 MPa, the t_{ip} of the P-CTRC and the PR-CTRC at each pressure is presented in Figure 10. The rule of searching for t_{ip} is based on the definition in Section 3.1. With an increase in P_3, the t_{ip} of the P-CTRC decreases, as well as t_{ip} of the PR-CTRC. The reason from the CO_2 property is that the peak value of cp decreases with an increase in P_3, and the pseudo-critical temperature increases and drifts away from the recovery region of engine coolant, as shown in Figure 11. With an additional heating process, the t_{ip} of PR-CTRC is obviously higher than that of the P-CTRC at the same P_3.

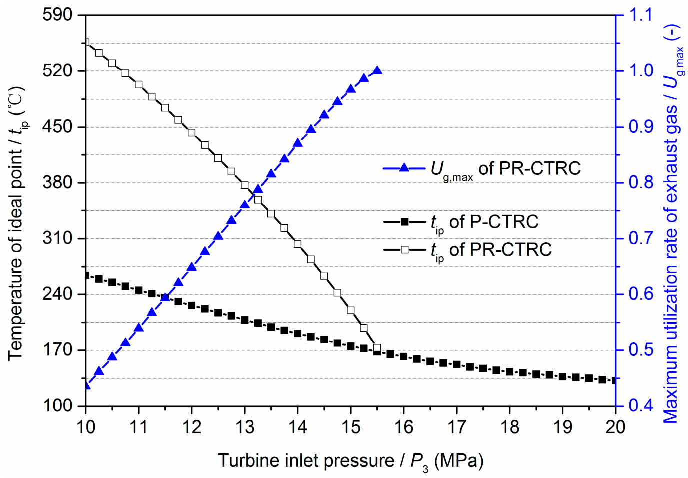


Figure 10. Variation of t_{ip} for the P-CTRC and PR-CTRC, and maximum utilization rate of exhaust gas for the PR-CTRC with turbine inlet pressure.

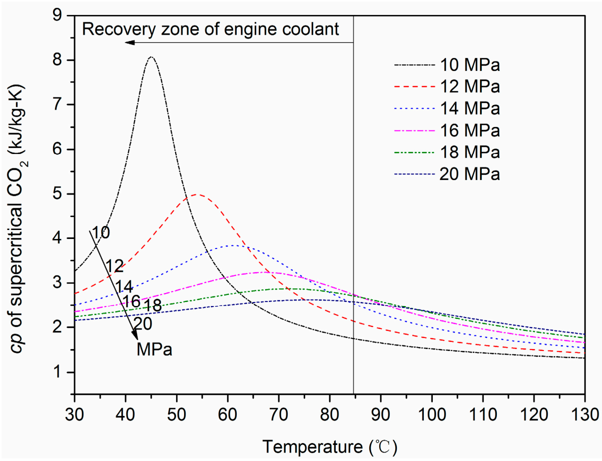


Figure 11. Variation of *cp* of supercritical CO_2 with temperature at different pressures.

For the PR-CTRC, there exists a maximum turbine inlet pressure, no t_{ip} occurs above it. The maximum turbine inlet pressure is 15.5 MPa in this study. With an increase in P_3, the temperature difference between the turbine inlet and outlet becomes larger, which reduces the regenerated degree of turbine outlet fluid. When the P_3 is higher than the maximum value, there is no regenerated condition for the PR-CTRC. At this situation, the PR-CTRC can be considered as the P-CTRC actually. Hence, the end of t_{ip} for the PR-CTRC in Figure 10 approaches to the variation line of the P-CTRC. Different with the P-CTRC obtaining complete recovery of exhaust gas at t_{ip}, the PR-CTRC has a maximum utilization rate of exhaust gas ($U_{g,max}$), as shown in Figure 10. The $U_{g,max}$ increases with an increase in P_3, then approaches to 1 at the maximum P_3 of each system, whose principle is similar to the discussion of t_{ip}. It indicates that both complete recovery of exhaust gas and engine coolant cannot achieve in the PR-CTRC, which only occurs in the P-CTRC.

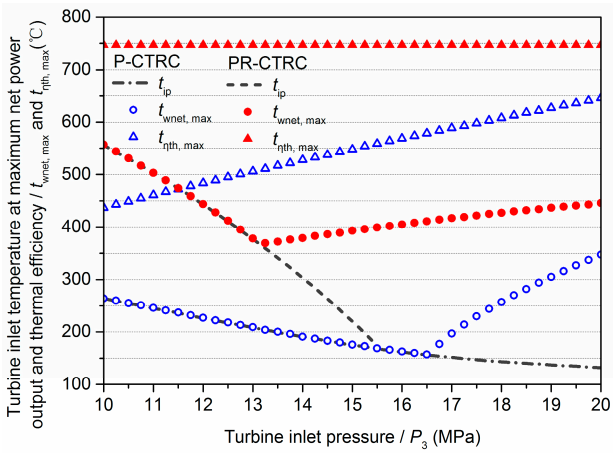


Figure 12. Comparison between "ideal point" and conditions where obtain the maximum net power output and the highest thermal efficiency.

As a comparison with t_{ip}, the turbine inlet temperature at each P_3 where can get maximum net power output ($t_{wnet,max}$) and thermal efficiency ($t_{\eta,max}$) is shown in Figure 12. The $t_{wnet,max}$ is equal to the t_{ip} firstly with an increase in turbine inlet pressure, then goes through a transition point and be different with t_{ip}. The transition pressure for the P-CTRC and the PR-CTRC is the 16.5 and 13.0 MPa, respectively. Different with the $t_{wnet,max}$ and t_{ip}, the $t_{\eta,max}$ remains high value, especially for the PR-CTRC always being the highest turbine inlet temperature (747 °C). Therefore, a consequence we can get from these performance laws is that maximum net power output can be also obtained when system design at the 'ideal point' of low pressure range (<13 MPa for the P-CTRC, <16.5 for the PR-CTRC). While, parameters selection would be considerable at high pressure range. The t_{ip}, $t_{wnet,max}$ and $t_{\eta,max}$ are at low, mid and high turbine inlet temperature range. As an example, when the P-CTRC at 20 MPa, t_{ip}, $t_{wnet,max}$ and $t_{\eta,max}$ is 132, 347 646 °C. Decision to select turbine inlet temperature depends on the system tending to the "ideal point" or the maximum net power output or the highest thermal efficiency.

3.3. "Ideal Point" Comparison between CO_2 and Organic Fluids

For clearer explanation of CO_2 owning well combined capability of exhaust gas and engine coolant, comparison study with organic fluids is conducted.

The application temperature range of engine coolant is narrower than that of exhaust gas, but the energy of engine coolant is approximate equivalent to that of exhaust gas ($Q_{c,max}/Q_{g,max}$ = 0.870). Therefore, higher *cp* is required in the application temperature range of engine coolant than that of exhaust gas if systems aim to realize complete utilization of exhaust gas and engine coolant meanwhile. It is obvious to find that the *cp* peak of supercritical CO_2 is just located in the application temperature range of engine coolant. The *cp* property of the supercritical CO_2 exactly owns better combined condition for exhaust gas and engine coolant. Just as shown in Figure 13, the values of cp/cp_{peak} of CO_2 and some frequently-used organic fluids (R125, R143a, R290, R600 and R123) at 1.5 times critical pressure. Only the *cp* peak of CO_2 is located in the application temperature range of engine coolant. On the contrary, most organic fluids has a slight change of *cp* under normal pressure, or owns a peak value of *cp* under supercritical pressure but located at higher application temperature range than that of engine coolant. This is the reason why the organic fluids is inefficient to utilize the engine coolant but enough for exhaust gas.

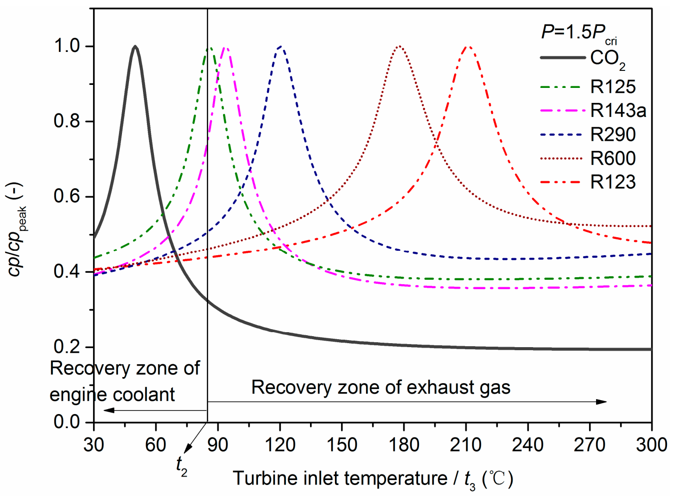


Figure 13. Variation of cp/cp_{peak} with temperature at 1.5 times critical pressure for different fluids.

Based on the transcritical Rankine cycle with a preheater, Table 3 lists the investigated fluids with critical parameters, utilization rate of waste heats, and temperature of idea point at 1.5 times critical pressure. It is easy to find that the pseudo-critical temperature has significant effect on the t_{ip}. With a higher critical temperature, the t_{ip} is lower or inexistent. With a low critical temperature of 31.0 °C, the t_{ip} is 244 °C for the CO_2 system. And for the R600 and R123 with a high critical temperature of 152.0 and 183.7 °C, respectively, the t_{ip} of them are inexistent. The t_{ip} of R143a and R290 are close to the critical temperature, which is difficult to operate at the t_{ip} condition as expansion process is easy turn into two phase region for these two wet working fluids. Moreover, higher t_{ip} entering into the turbine makes more power output at the same condition. Therefore, CO_2 with low critical temperature and higher t_{ip} is more suitable for the combined recovery of exhaust gas and engine coolant. As shown the results of t_3 = 300 °C, system with organic fluids can obtain a complete recovery of exhaust gas (U_g = 1), but a lower utilization rate of engine coolant (U_c = 0.227–0.325) than CO_2-based system (U_c = 0.763). Therefore, as a comparison with references [23,24] using organic fluids as working fluids, using CO_2 has large advantage of utilization rate of engine coolant.

Table 3. Investigated fluids with critical parameters, utilization rate of waste heats, and t_{ip}.

Fluids	P_{cri} (MPa)	t_{cri} (°C)	U_g (-)	U_c (-)	U_g (-)	U_c (-)	t_{ip} (°C)
			t_3 = 200 °C	t_3 = 200 °C	t_3 = 300 °C	t_3 = 300 °C	
CO_2	7.38	31.0	0.758	1	1	0.763	244
R125	3.62	66.0	1	0.651	*	*	150
R143a	3.76	72.7	1	0.528	1	0.325	129
R290 (propane)	4.25	96.7	1	0.390	1	0.240	124
R600 (butane)	3.79	152.0	1	0.368	1	0.213	-
R123	3.66	183.7	1	0.459	1	0.227	-

* t_3 = 300 °C exceeds the thermal decomposition temperature of R125; **-** inexistent.

From the comparison study, it should be interesting to conduct a fast way to obtain the "ideal point" of any kind of fluids. Such a method should be more convenient to engineering design. Therefore, this study proposes a prediction formula of turbine inlet temperature at the condition of "ideal point". From the analysis and calculation models above, the turbine inlet temperature at the condition of "ideal point" (t_{ip}) mainly depends on quantity and quality of exhaust gas and engine coolant, turbine inlet pressure, inlet and outlet parameter of fluid in preheater, and inlet parameter of fluid in gas heater, which is described as following:

$$t_{ip} = t(P_{ip}, h_{ip}),\quad h_{ip} = \frac{m_g \cdot Cp_g \cdot (t_{g,in} - t_{g,out})}{m_c \cdot Cp_c \cdot (t_{c,in} - t_{c,returned})} \cdot (h_{preh,out} - h_{preh,in}) + h_{gh,in} \tag{13}$$

Wherein,

$$h_{preh,out} = h(P_{ip}, t_{preh,out}),\ t_{preh,out} = t_{c,in} + \Delta t_{pp,gh};$$

$$h_{preh,in} = h(P_{ip}, t_{preh,in}),\ t_{preh,in} = t_{pump,out};$$

$$h_{gh,in} = h(P_{ip}, t_{gh,in}),\ \ t_{gh,in} = t_{preh,out}\ \text{(P-CTRC)},\ \ t_{gh,in} = t_{reg,out}\ \text{(PR-CTRC)};$$

$$t_{g,out} = t_{g,dew}\ \text{ when }\ t_{g,dew} > t_{gh,in} + \Delta t_{pp,gh};$$

$$t_{g,out} = t_{gh,in} + \Delta t_{pp}\ \text{ when }\ t_{g,dew} \le t_{gh,in} + \Delta t_{pp,gh};$$

Among them, $t_i = t(P_i, h_i)$ and $h_i = h(P_i, t_i)$ are the processes of gaining property of working fluid, which can be realized by REFPROP 9.0 software.

The t_{ip} can be rapidly and simply obtained by this prediction formula, which is used as a direction for engineering design when utilizing exhaust gas and engine coolant simultaneously, and also as a methodology used for combined recovery of two or multiple waste heats in other fields.

4. Performance of P-CTRC Prototype

Based on the "ideal point" design method, a kW-class P-CTRC prototype was conducted to investigate the practical utilization degree of exhaust gas and engine coolant. An optimization work on the practical operating of the P-CTRC loop was conducted for exact recovery of engine coolant, which avoided deficient or excessive recovery.

4.1. Description of the P-CTRC Prototype

The P-CTRC prototype was constructed specially for E-WHR, whose practical layout is shown in Figure 14 and diagram is shown in Figure 15. The main components and measuring devices are listed in Table 4. As the turbine was not available and under design, an expansion valve was adopted temporarily instead of it to provide high and low pressures of the prototype.

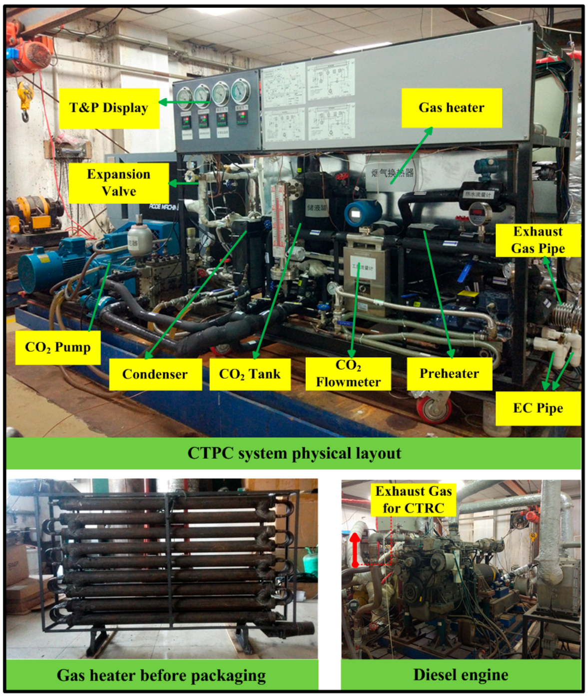


Figure 14. Photo of the P-CTRC prototype.

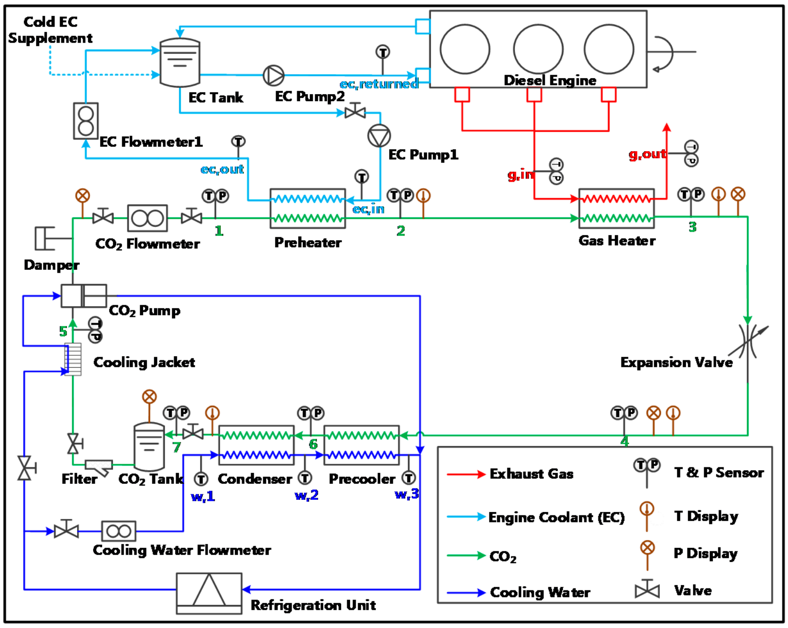


Figure 15. Diagram of the P-CTRC prototype integrated with the diesel engine.

A diesel engine was temporarily used to provide waste heat condition of exhaust gas and engine coolant. Main parameters of the diesel engine are listed in Table 5, which is the bigger scale than the gasoline engine. Hence, the diesel engine operated at part load to match the small-scale P-CTRC system. The engine coolant used water as working fluid, and was driven by the engine coolant pump2 (EC pump2). An open EC tank was set for heat rejection of hot water with a large evaporation area when the engine didn't link to the P-CTRC. But the heat rejection capacity of EC tank was not generally enough, then a cold EC supplement system was additionally used to mix some cold water to maintain the thermal balance. In this system, the coolant flow did not go through the preheater straightly. Instead of it, a part of flow driven by the EC pump1 was introduced to preheat CO_2 from the EC tank. The scheme of two EC pump is convenient and suitable for controlling the engine-CTRC combined system at laboratory conditions. Hence, the recovery for the engine coolant by the preheater can replace additional cold EC supplement system.

Table 4. Main components and measuring devices.

Components	Types	Specifications/(Precision)
Gas heater	Double-pipe	3.09 m^2
Preheater, Precooler, Condenser	Brazed plate	1.56 m^2
Expansion valve	Needle valve	0–100% opening/(±1%)
CO_2 tank	Vertical	10 L
CO_2 pump	Reciprocating plunger	1.7 m^3/h
Refrigerating unit	R22	-
Measuring Devices	**Types**	**Specifications/(Precision)**
CO_2 flowmeter	Coriolis	0–0.3 kg/s (±0.2%)
Air flowmeter	Laminar flow	0–1350 kg/h (±0.5%)
Fuel consumption meter	-	5–2000 kg/h (±0.8%)
EC flowmeter1	Turbine	0–10 m^3/h (±0.5%)
Cooling water flowmeter	Turbine	0–12 m^3/h (±1%)
Liquidometer for the CO_2 tank	Magnetic flap type	0–30 cm (±3.3%)
Thermocouple (exhaust gas)	Armoured	−60–650 °C/(±1%)
Resistance temperature detector (CO_2 and water)	Armoured P100	−200–500 °C/(±0.15%)
Pressure transmitter (gas and water)	Low pressure type	0–0.5 MPa (±0.065%)
Pressure transmitter (CO_2 at low pressure side)	High pressure type	0–12 MPa (±0.065%)
Pressure transmitter (CO_2 at high pressure side)	High pressure type	0–14 MPa (±0.065%)

Table 5. Main parameters of the diesel engine.

Parameters [Unit]	Values
Engine type [-]	In-line, six cylinders
Air intake mode [-]	Supercharged and intercooled
Cylinder diameter [mm]	113
Stroke length [mm]	140
Total displacement [L]	8.424
Rated power [kW]	243
Rated speed [rpm]	2200
Maximum torque [N·m]	1280 (1800 rpm)

4.2. Verification for Sufficient Recovery of Exhaust Gas and Engine Coolant

After operating the engine at a fixed condition, the CO_2 pump at a constant speed, the EC pump1 at a constant speed, and by adjusting various opening of the expansion valve, a group results with various high pressures (P_3) ranging from 7.57 MPa to 10.35 MPa were tested and shown in Table 6.

The exhaust gas with inlet temperature of 480.1–482.9 °C is utilized in gas heater and reaches to outlet temperature of 71.1–83.1 °C, which means the full recovery of it and even excessive recovery due to the limitation of 120 °C. In addition to part of heat dissipation to the environment on the large surface of the gas heater, the CO_2 gains 44.4–49.8 kW heat absorption in the gas heater. For the engine coolant, the flow with inlet temperature of 75.1–75.7 °C is utilized in preheater and cooled down to outlet temperature of 28.5–33.8 °C, which nearly approaches to the inlet temperature of CO_2. Hence, the engine coolant was also utilized fully by the CO_2 that gets 22.7–26.7 kW heat absorption in preheater. Therefore, the P-CTRC proves to own the capability for sufficient recovery of exhaust gas and engine coolant. As the gas heater has large surface contacting environment, and larger mass flow rate of exhaust gas (diesel engine) through the exchanger than that of design (gasoline engine), heat transfer effectiveness of the gas heater is 73.7–80.2%, which needs to be improved in the future. But for the preheater, well thermal match and compact exchanger type produces higher effectiveness, almost above 90%.

Similar with the analysis in Section 3.1, this process in *cp* perspective using condition 5 as an example is shown in Figure 16. The CO_2 absorbs 22.7 kW heat from the engine coolant in the preheater with a temperature increase from 33.1 °C to 52.2 °C, and absorbs 45.0 kW heat from the exhaust gas in the gas heater with a temperature increase from 52.2 °C to 165.0 °C. It is obvious to find that heat recovery zone of engine coolant locates around the critical area, and benefits the peak of *cp* peak of supercritical CO_2. In the preheater, CO_2 temperature only gets 29.1 °C increase, but large amount of heat absorption is obtained. Therefore, it proves that the waste heat recovery of engine coolant with characteristic of low-grade but large-amount can be accomplished by the P-CTRC in a narrow temperature range. On this basis, it still does not affect the full recovery of exhaust gas.

$$\eta_i = \frac{Q_i}{Q_{i,max}},\quad Q_{gh,max} = m_g \cdot cp_g \cdot (t_{g,in} - t_2),\quad Q_{preh,max} = m_c \cdot cp_c \cdot (t_{c,in} - t_1) \tag{14}$$

Table 6. Main parameters and performance of the CTRC test system.

Parameters (Unit)	Value				
	Condition 1	Condition 2	Condition 3	Condition 4	Condition 5
P_3 (kPa)	7570	8202	8834	9632	10,349
P_4 (kPa)	6834	6847	6834	6846	6846
m_f (kg/s)	0.238	0.233	0.228	0.220	0.219
t_1 (°C)	27.8	29.1	30.4	31.9	33.1
t_2 (°C)	35.1	39.5	43.7	48.4	52.2
t_3 (°C)	160.8	158.2	158.8	159.9	165.0
$t_{g,in}$ (°C)	480.1	480.8	481.6	482.1	482.9
$t_{g,out}$ (°C)	71.1	71.6	75.6	79.4	83.1
$t_{ec,in}$ (°C)	75.2	75.1	75.3	75.4	75.7
$t_{ec,out}$ (°C)	28.5	29.4	30.7	32.2	33.8
$t_{ec,returned}$ (°C)	83.9	84.1	84.1	84.4	84.7
Q_{preh} (kW) *	26.7	25.4	24.5	23.1	22.7
Q_{gh} (kW) *	49.8	47.8	46.3	44.4	45.0
η_{preh} (%) #	96.1	93.3	97.1	89.8	90.8
η_{gh} (%) #	80.2	77.6	75.7	73.7	75.2

* By a detailed error analysis that used the method in previous study by our group [29], the maximum uncertainty of heat transfer quantity in preheater and gas heater is 5.8% and 3.6% respectively; # As *m·cp* of both exhaust gas and engine coolant are smaller than that of CO_2, the efficiencies of heat exchangers are calculated by following equation according to Ref. [30].

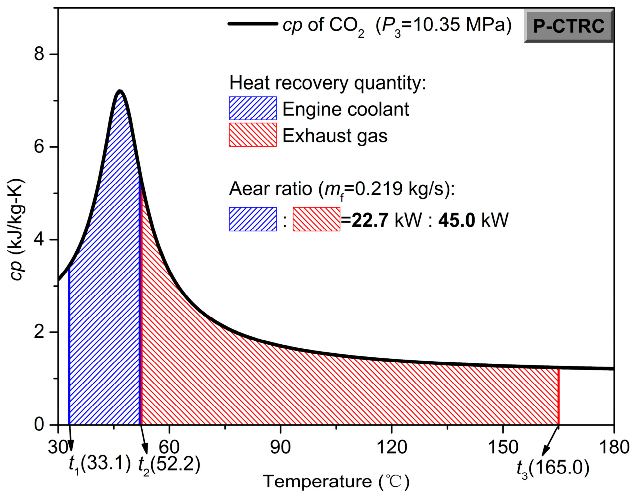


Figure 16. Heat recovery capacity from exhaust gas and engine coolant in the P-CTRC prototype.

4.3. Operation Optimization

In practical operation, a problem should be considered that exhaust gas and engine coolant may not achieve complete recovery exactly, and deficient and excessive recovery may happen. As shown in Table 6, system operates from condition 1 to condition 5 sequentially, then the $t_{ec,returned}$ increased from 83.9 to 84.7 °C. If operation still continued afterwards, the $t_{ec,returned}$ should reaches up to the boiling temperature of engine coolant. This conditions are considered as the deficient recovery of engine coolant, which means inadequate heat dissipation of engine coolant and makes $t_{ec,returned}$ increase continuously. The $t_{ec,returned}$ is a significant parameter for the engine operation, too high causes boiling problem of engine coolant, too low leads to low-efficiency operating of engine. In this study, an optimization was conducted to find the optimal recovery condition of engine coolant.

The diesel engine also operated at constant condition: 50% of rated speed (1100 rpm) and 50% load (601 N·m). The theoretical work in Section 3 was to find ideal turbine inlet temperature and corresponding mass flow rate of CO_2 to match complete recovery of the waste heats. But in this study, adjusting CO_2 flow would obviously influent the high pressure of system, thus another strategy was adopted from the perspective of engine coolant side. The flow rate of engine coolant into the preheater could be adjusted by frequency changer of EC pump1. The final objective was to find an optimal frequency, namely an optimal mass flow rate of engine coolant that made the $t_{ec,returned}$ stable. But change of $t_{ec,returned}$ is not obvious in a short time, so the $t_{ec,in}$ is selected as the observation object to replace the $t_{ec,returned}$. Because the $t_{ec,in}$ will increase and decrease when the $t_{ec,returned}$ increases and decreases correspondingly. Therefore, if $t_{ec,in}$ keeps stable at the experiment time, it means thermal balance of engine coolant system is constructed and achieve a complete recovery of engine coolant. If $t_{ec,in}$ keeps increasing, it means the preheater is unable to undertake enough heat rejection and make temperature increasing, which calls deficient recovery of engine coolant in this study. On the contrary, if $t_{ec,in}$ keeps decreasing it means excessive recovery of engine coolant. It should be note that, there is difference between the $t_{ec,back}$ and the $t_{ec,in}$ relating to the heat loss on the long pipe between the EC tank and the P-CTRC system.

By a previous work, a rough frequency range of EC pump1 was selected and finally operated at 8.00, 8.33, 8.75, 9.17, 9.50 Hz. High frequency means more flow of engine coolant into the preheater. The CO_2 pump operated at a constant speed (80 rpm) and expansion valve was at a fixed opening for the five groups. For convenient observation, temperature change of $t_{ec,in}$ versus starting time was selected as the target, which means 0 °C temperature change is the optimal. The starting and ending temperatures at the five groups are listed at Table 7.

Table 7. Starting inlet temperature of engine coolant ($t_{ec,in}$).

Frequency (Hz)	$t_{ec,in}$ at 0 s (°C)	$t_{ec,in}$ at 120 s (°C)	$t_{ec,back}$ at 0 s (°C)	$t_{ec,back}$ at 120 s (°C)
8	75.2	76.1	85.6	85.9
8.33	75.2	75.4	84.1	84.4
8.75	75.4	75.5	83.8	83.8
9.17	76.1	75.9	84.1	83.9
9.5	76.8	76.1	84.2	83.6

As shown in Figure 17, the results of temperature change at 120 s were obtained for five frequencies. The region of above and under 0 °C means the deficient and excessive recovery, respectively. From 8.00 Hz to 9.50 Hz, the flow rate of engine coolant increases, thus engine coolant crosses from deficient recovery to excessive recovery. When EC pump1 operates at 8.00 Hz, lacking engine coolant flows into the preheater and leads to deficient recovery, then makes $t_{ec,in}$ has an increasing change of about 0.8 °C in 120 s. And for the 9.50 Hz, overmuch engine coolant flows into the preheater and leads to excessive recovery, then makes $t_{ec,in}$ has about an decreasing change of 0.7 °C in 120 s. 8.75 Hz is optimal frequency among the five, and temperature change controls less than 0.1 °C. With necessary heat rejection of engine coolant on the pipe and EC tank under laboratory conditions, this optimal condition can be considered as the complete recovery condition of engine coolant. As shown in Table 7, the $t_{ec,back}$ also indicates the same change performance but a smaller change than the $t_{ec,in}$, so the discussion of $t_{ec,in}$ is effective.

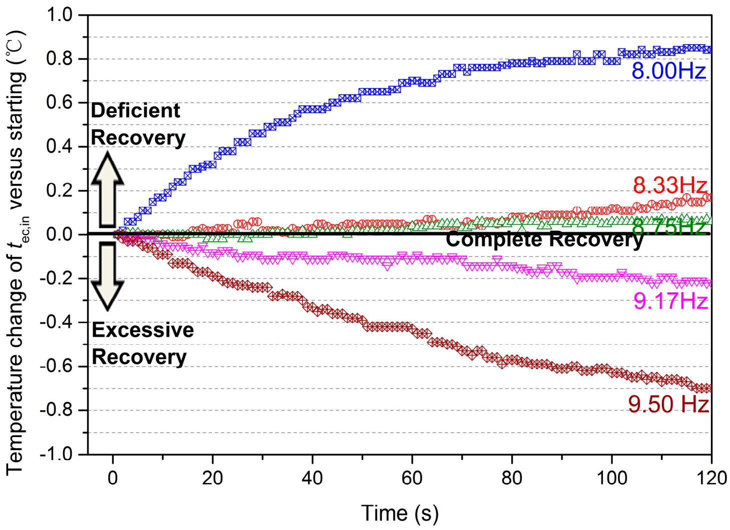


Figure 17. Temperature change versus starting of engine coolant inlet temperature of preheater ($t_{ec,in}$) at various frequency of EC pump1.

5. Conclusions

A detailed theoretical study of the CTRC system was conducted for analyzing complete recovery of exhaust gas and engine coolant, as well as a simple test study. Main conclusions can be listed as follows:

1. For the P-CTRC, the 'ideal point' is defined as an important reference to the combined capability of exhaust gas and engine coolant, which represents the condition of both complete recovery of two waste heats and is expressed by t_{ip}. The t_{ip} changes with the variation of turbine inlet pressure. The t_{ip} is 264 °C at P_3 = 10 MPa. The 'ideal point' also exists in the PR-CTRC, but has some difference with that of the P-CTRC. The exhaust gas doesn't obtain a complete use after adding the regenerator, in which the regenerated heat recovers a part of application temperature range of exhaust gas and increases the t_{ip}. The t_{ip} is 556 °C at P_3 = 10 MPa. There exists a maximum turbine inlet pressure (15.5 MPa) for the PR-CTRC, no t_{ip} occurs above it.
2. Maximum net power output can be also obtained when system design at the 'ideal point' of low pressure range (<13 MPa for the P-CTRC, <16.5 MPa for the PR-CTRC). While, there is no link between the 'ideal point' and the highest thermal efficiency. Parameters selection will be considerable at high pressure range, Decision to select turbine inlet temperature depends on the system tending to the "ideal point" or the maximum net power output or the highest thermal efficiency.
3. This study proposes a prediction formula of t_{ip}, which is used as a direction for engineering design for utilizing exhaust gas and engine coolant meanwhile, and also as a methodology used for combined recovery of two or multiple waste heats in other fields.
4. The operating results with P_3 range of 7.57–10.35 MPa show that sufficient recovery of the two waste heats can be achieve by the P-CTRC prototype. The exhaust gas with initial temperature of 480.1–482.9 °C is utilized to 71.1–83.1 °C, and the engine coolant with initial temperature of 75.1–75.7 °C is utilized to 28.5–33.8 °C. The CO_2 gains 44.4–49.8 kW and 22.7–26.7 kW heat absorption from the exhaust gas engine coolant respectively. Heat transfer effectiveness of the gas heater and the preheater are 73.7–80.2% and 89.8–96.1%, respectively.
5. An experimental optimization is conducted to search the optimal operating condition of achieving complete recovery of engine coolant exactly, and avoid deficient and excessive recovery. After operating five different frequencies of EC pump1, an optimal amount of engine

coolant flowing into the preheater was found, and the corresponding optimal frequency of EC pump1 is 8.75 Hz.

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Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations:

cond	condenser
c	engine coolant
dew	acid dew point
f	working fluid
g	exhaust gas
gen	generator
gh	gas heater
i	each component
ip	ideal point
in	inlet
max	maximum
out	outlet
pp	pinch point
preh	preheater
reg	regenerator
th	thermal
tha	total heat absorption
turb	turbine
w	cooling water
EC/ec	engine coolant
1-7	state point
1*s*, 4*s*	state points for the ideal case
η	efficiency
cp	specific heat capacity (kJ/(kg·K))
h	specific enthalpy (kJ/kg)
m	mass flow rate (kg/s)
P	pressure (kPa)
Power	electricity power (kW)
Q	heat flow rate (kW)
t	temperature (°C)
$\Delta t_{pp,gh}$	temperature difference (°C)
U	utilization rate of waste heat
W	work (kW)
CTRC	CO_2-based transcritical Rankine cycle
E-WHR	engine waste heat recovery
ORC	organic Rankine cycle
P-CTRC	CO_2-based transcritical Rankine cycle with a preheater
PR-CTRC	CO_2-based transcritical Rankine cycle with a preheater and a regenerator
PPTD	Pinch Point Temperature Difference

References

1. Wang, T.; Zhang, Y.; Zhang, J.; Peng, Z.; Shu, G. Comparisons of system benefits and thermo-economics for exhaust energy recovery applied on a heavy-duty diesel engine and a light-duty vehicle gasoline engine. *Energy Convers. Manag.* **2014**, *84*, 97–107.

2. Fu, J.; Liu, J.; Feng, R.; Yang, Y.; Wang, L.; Wang, Y. Energy and exergy analysis on gasoline engine based on mapping characteristics experiment. *Appl. Energy* **2013**, *102*, 622–630.
3. Song, S.; Zhang, H.; Zhao, R.; Meng, F.; Liu, H.; Wang, J.; Yang, B. Simulation and Performance Analysis of Organic Rankine Systems for Stationary Compressed Natural Gas Engine. *Energies* **2017**, *10*, 544.
4. Benato, A.; Macor, A. Biogas Engine Waste Heat Recovery Using Organic Rankine Cycle. *Energies* **2017**, *10*, 327.
5. Yao, B.; Yang, F.; Zhang, H.; Wang, E.; Yang, K. Analyzing the performance of a dual loop organic rankine cycle system for waste heat recovery of a heavy-duty compressed natural gas engine. *Energies* **2014**, *7*, 7794–7815.
6. Wang, X.; Tian, H.; Shu, G. Part-Load Performance Prediction and Operation Strategy Design of Organic Rankine Cycles with a Medium Cycle Used for Recovering Waste Heat from Gaseous Fuel Engines. *Energies* **2016**, *9*, 527.
7. Altosole, M.; Benvenuto, G.; Campora, U.; Laviola, M.; Trucco, A. Waste heat recovery from marine gas turbines and diesel engines. *Energies* **2017**, *10*, 718.
8. Andreasen, J.; Meroni, A.; Haglind, F. A Comparison of Organic and Steam Rankine Cycle Power Systems for Waste Heat Recovery on Large Ships. *Energies* **2017**, *10*, 547.
9. Cardemil, J.M.; Silva, A.K.D. Parametrized overview of CO_2, power cycles for different operation conditions and configurations—An absolute and relative performance analysis. *Appl. Therm. Eng.* **2016**, *100*, 146–154.
10. Sarkar, J. Review and future trends of supercritical CO_2, Rankine cycle for low-grade heat conversion. *Renew. Sustain. Energy Rev.* **2015**, *48*, 434–451.
11. Wang, X.; Dai, Y. Exergoeconomic analysis of utilizing the transcritical CO_2 cycle and the ORC for a recompression supercritical CO_2 cycle waste heat recovery: A comparative study. *Appl. Energy* **2016**, *170*, 193–207.
12. Guo, T.; Wang, H.; Zhang, S. Comparative analysis of CO_2-based transcritical Rankine cycle and HFC245fa-based subcritical organic Rankine cycle using low-temperature geothermal source. *Sci. China Technol. Sci.* **2010**, *53*, 1638–1646.
13. Cayer, E.; Galanis, N.; Nesreddine, H. Parametric study and optimization of a transcritical power cycle using a low temperature source. *Appl. Energy* **2010**, *87*, 1349–1357.
14. Chen, Y.; Lundqvist, P.; Platell, P. Theoretical research of carbon dioxide power cycle application in automobile industry to reduce vehicle's fuel consumption. *Appl. Therm. Eng.* **2005**, *25*, 2041–2053.
15. Farzaneh-Gord, M.; Mirmohammadi, A.S.; Behi, M.; Yahyaie, A. Heat recovery from a natural gas powered internal combustion engine by CO_2 transcritical power cycle. *Therm. Sci.* **2010**, *14*, 897–911.
16. Kim, Y.; Sohn, J.; Yoon, E. Supercritical CO_2, Rankine cycles for waste heat recovery from gas turbine. *Energy* **2017**, *118*, 893–905.
17. Shu, G.; Shi, L.; Tian, H.; Deng, S.; Li, X.; Chang, L. Configurations selection maps of CO_2-based transcritical Rankine cycle (CTRC) for thermal energy management of engine waste heat. *Appl. Energy* **2017**, *186*, 423–435.
18. Persichilli, M.; Held, T.; Hostler, S.; Zdankiewicz, E.; Klapp, D. Transforming Waste Heat to Power through Development of a CO_2-Based-Power Cycle. In Proceedings of the Electric Power Expo**,** Rosemount, IL, USA, 10–12 May 2011.
19. Held, T.J. Initial Test Results of a Megawatt-Class Supercritical CO_2 Heat Engine. In Proceedings of the 4th International Symposium on Supercritical CO_2 Power Cycles, Pittsburgh, PA, USA, 9–10 September 2014.
20. Pan, L.; Li, B.; Wei, X.; Li, T. Experimental investigation on the CO_2 transcritical power cycle. *Energy* **2016**, *95*, 247–254.
21. Shi, L.; Shu, G.; Tian, H.; Huang, G.; Chen, T.; Li, X.; Li, D. Experimental comparison between four CO_2-based transcritical Rankine cycle (CTRC) systems for engine waste heat recovery. *Energy Convers. Manag.* **2017**, *150*, 159–171.
22. Tian, H.; Shu, G.; Wei, H.; Liang, X.; Liu, L. Fluids and parameters optimization for the organic Rankine cycles (ORCs) used in exhaust heat recovery of Internal Combustion Engine (ICE). *Energy* **2012**, *47*, 125–136.
23. Vaja, I.; Gambarotta, A. Internal combustion engine (ICE) bottoming with organic Rankine cycles (ORC). *Energy* **2010**, *35*, 1084–1093.

24. Kim, Y.M.; Dong, G.S.; Chang, G.K.; Cho, G.B. Single-loop organic Rankine cycles for engine waste heat recovery using both low- and high-temperature heat sources. *Energy* **2016**, *96*, 482–494.
25. Kim, Y.M.; Kim, C.G.; Favrat, D. Transcritical or supercritical CO_2 cycles using both low- and high-temperature heat sources. *Energy* **2012**, *43*, 402–415.
26. Amini, A.; Mirkhani, N.; Pourfard, P.P.; Ashjaee, M.; Khodkar, M.A. Thermo-economic optimization of low-grade waste heat recovery inYazd combined-cycle power plant (Iran) by a CO_2 transcritical Rankine cycle. *Energy* **2015**, *86*, 74–84.
27. Cayer, E.; Galanis, N.; Desilets, M.; Nesreddine, H.; Roy, P. Analysis of a carbon dioxide transcritical power cycle using a low temperature source. *Appl. Energy* **2009**, *86*, 1055–1063.
28. Vélez, F.; Segovia, J.; Chejne, F.; Antolín, G.; Quijano, A.; Martín, C.M. Low temperature heat source for power generation: Exhaustive analysis of a carbon dioxide transcritical power cycle. *Energy* **2011**, *36*, 5497–5507.
29. Shu, G.; Zhao, M.; Tian, H.; Huo, Y.; Zhu, W. Experimental comparison of R123 and R245fa as working fluids for waste heat recovery from heavy-duty diesel engine. *Energy* **2016**, *115*, 756–769.
30. Nellis, G.; Klein, S. Heat exchangers. In *Heat Transfer*, 1st ed.; Cambridge University Press: New York, NY, USA, 2012; Volume 8.3.2, Chapter 8, pp. 852–853, ISBN 978-1-107-67137-9.