THE DYNAMIC PERFORMANCE OF A CO2 MIXTURE TRANSCRITICAL POWER CYCLE FOR WASTE HEAT RECOVERY OF INTERNAL COMBUSTION ENGINE

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ABSTRACT

Waste heat recovery technologies are regarded as one of the most promising method to improve efficiency of internal combustion engines of trucks. Therein, CO_2 mixture transcritical power cycle (CMTPC) can output large power and have smaller volume of heat exchanger at the same time under the design working condition. However, since the working conditions of trucks are complex, the waste heat recovery system also needs to work under different conditions. When the engine working condition changes, the properties of CO_2 vary obviously and the cycle may not be able to meet the design point anymore. Therefore, a dynamic model of CMTPC is established in this work by Simulink. The model is used to research the dynamic and off-design performance of the CMTPC system.
Keywords: CO_2 mixture transcritical power cycle; waste heat recovery; dynamic simulation; off-design condition

1. INTRODUCTION

In a common heavy-duty diesel engine (HDDE) only about 40-45% of the fuel energy is converted into useful work, while most of the energy is lost to ambient (Lion et al., 2017). Converting waste heat of exhaust gas and jacket water into usable power by implementing engine waste heat recovery (WHR) on heavy-duty diesel engine is taken as a potential way to improve the overall system efficiency(HORST et al., 2013). The CO_2 transcritical power cycle (CTPC) has been considered as one of the most capable system to recovery both the energy of exhaust and jacket water due to the unique physical properties of CO_2(Shi et al., 2018) . Besides, the addition of refrigerant can improve system performance of CTPC and reduce the operating pressure(Wu et al., 2017).

Existing studies about CO_2 mixtures transcritical power cycle (CMTPC) mainly concentrated on system design and parameter optimization under a constant heat source(Shengjun et al., 2011; Shu et al., 2018). Nevertheless, the heat source from engines always transient and volatile(Aghaali et al., 2015; Wang et al., 2011), thus most of the time the CMTPC system operates under the off-design conditions. The heavy-duty diesel engine doesn't work under rate condition all the time, and under off-design conditions the thermal efficiency of CMTPC may decrease obviously, which results bad energy-saving effect(Shu et al., 2016). So, it's necessary to study the dynamic performance of CMTPC system which is still lacking of relevant researches.

The dynamic system model is established in this paper by combining the main component models based on their interrelationship, which can reflect the dynamic process and finally reach steady state. Therein, the preheater and gas heater are established by finite volume (FV) method and the condenser is built by moving boundary (MB) method. Then the design point is optimized in order to obtain the maximum net power and smaller volume of heat exchanger at the same time. After that, the dynamic characteristics of the CMTPC system under different HDDE working conditions are discussed.

2. SYSTEM DESCRIPTION

A heavy-duty direct injection six cylinder in-line diesel engine with a rate power output of 243 kW is taken as the reference engine in this work. Detailed parameters of the target engine can be found in the relevant studies of our group(Li et al., 2018). The configuration of the CMTPC system is showed in Fig. 1, Superheated working fluid is expanded in the expander to the condensation pressure. During the condensation process, heat is rejected into the cooling water. Thereafter, the working fluid is pumped above supcritical pressure and absorbed heat in preheater and gas heater. Finally, the working fluid enters the expander to finish a cycle.

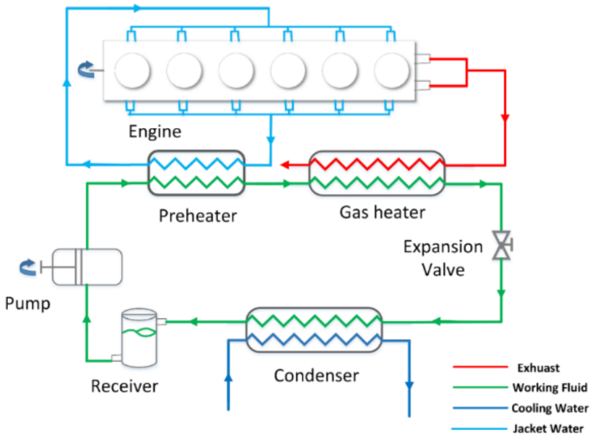

Figure 1: Schematic layout of the system

3. THE ESTABLISH OF THE MODEL

This work is performed with s-function code in Matlab and Refprop 9.0 is used to calculate the thermophysical properties of working fluid. The system dynamic model as shown in Fig. 2 is established by combining the main component models based on their interrelationship.

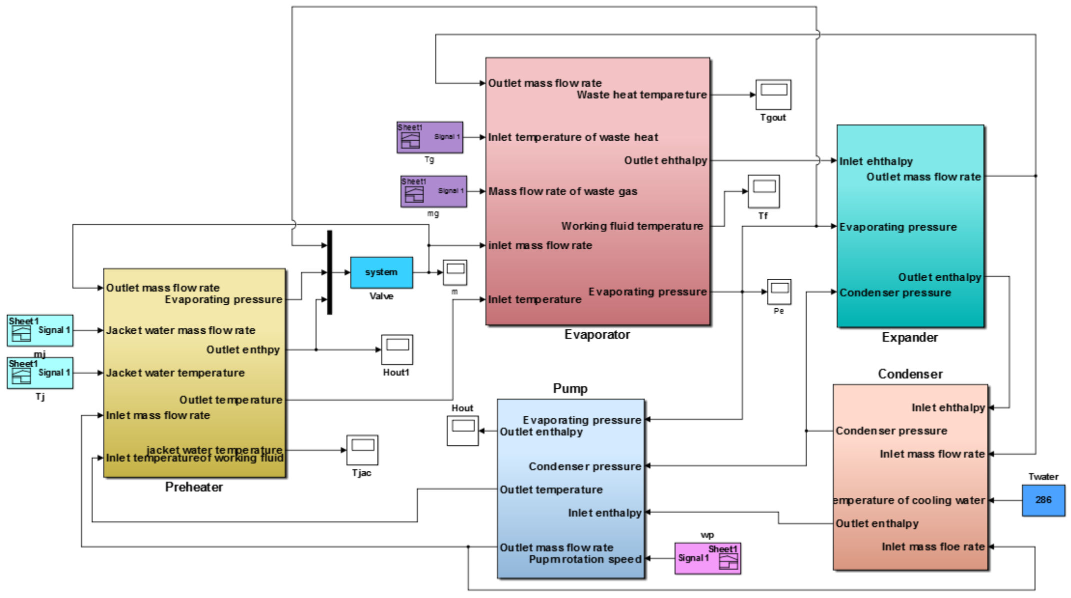

Figure 2: The CMTPC system model built in Simulink

3.1 Reference engine and optimization of design point

The selection of the most appropriate design point greatly affects the performance of the CMTPC system, so it is necessary to investigate the actual operation condition. Data obtained and elaborated from EPA (United States Environmental Protection Agency) showed that the heavy-duty diesel engine spends most of the time at full load and medium to high speed conditions at the typical operating cycle. Therefore, the heat source come from the engine which working at 80 % load and 80 % torque. The parameters of heat source and other assumptions for the components at design point are listed in Table 1.

Table 1: Main parameters of CMTPC design

Parameter	Unit	Values
Outlet temperature of exhaust/jacket water	°C	389.7/85
Mass flow rate of exhaust/jacket water	kg/s	0.32/3.23
Condensation temperature	°C	22
Pinch point in the gas heater	°C	30
Pinch point in the preheater	°C	10
Pinch point in the condenser	°C	5
Turbine/pump efficiency	——	0.7/0.6

3.2 Mathematic model

The gas heater and preheater are modeled by FV method and the condenser coupled with a receiver is built with MB method. The differential equations of mass and energy conservation for the working fluid and the differential equation of energy conservation for the exhaust, jacket water and gas heater, preheater wall are considered. Fig. 3 exhibit the finite volume (FV) model of the gas heater and the preheater.

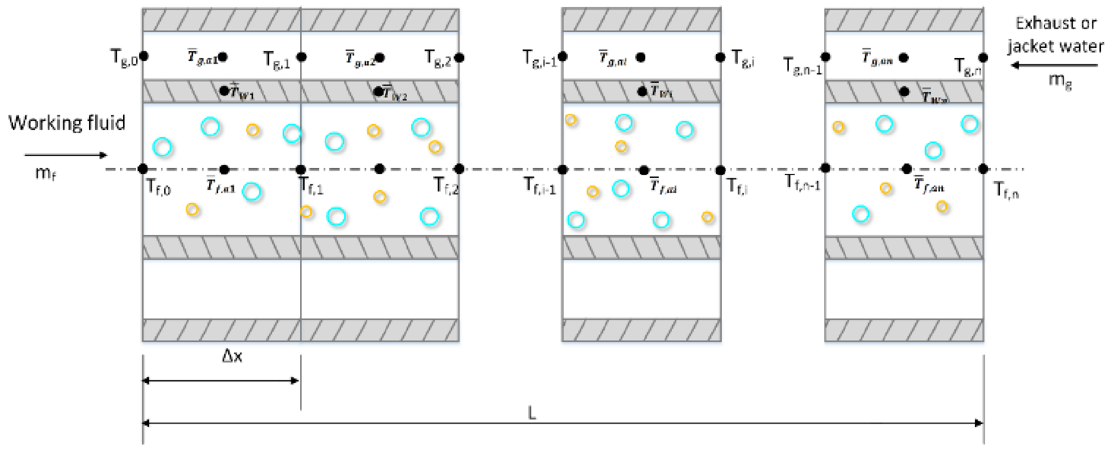

Figure 3: The finite volume model of gas heater or preheater

The move boundary (MB) model of condenser which is coupled with a receiver(Shu et al., 2017) is showed in Fig. 4. The general mass balance for working fluid, the energy balance differential equation for cooling water and working fluid are considered. The general energy balance differential equation for condenser wall is also take into consideration. This modeling method has been widely used(Wang et al., 2017).The Leibniz integration rule is used to integrate the governing partial differential equations (PDEs).

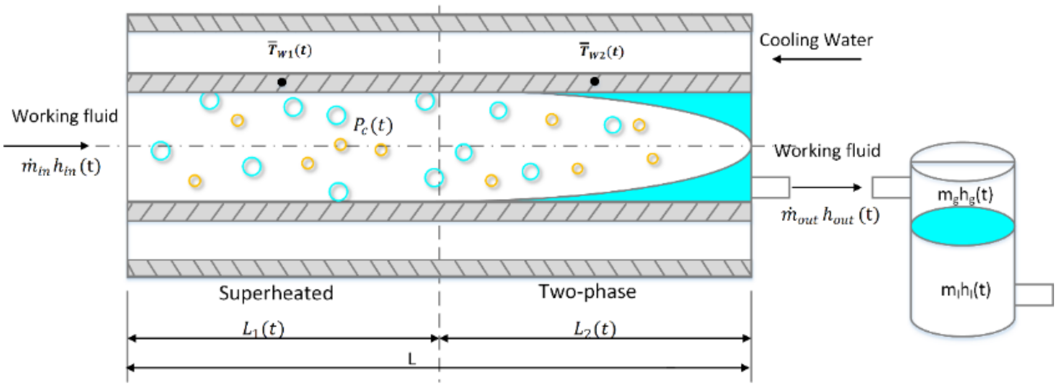

Figure 4: The move boundary model of condenser

The pump and expander always response faster compared with the heat exchangers, so static models are used for plunger pump and expander. The model of expander is replaced by a nozzle(Jensen and Tummescheit, 2002) .

4. RESULT AND DISCUSSION

4.1 Steady-state point optimization

In this paper, R134a is chosen as the organic addition mixed with CO_2 due to the non-toxicity, non-corrosiveness, low ODP and GWP(Shengjun et al., 2011) and the reliability in calculating the mixture properties has been verified by our team(Shu et al., 2018). Based on the heat sources of HDDE, the operating pressure, the turbine outlet pressure and the proportion of CO_2 and R134a greatly affect the net power output and the total length of heat exchangers (including the preheater, the gas heater and the condenser). As shown in Fig. 5 and 6, the relationship between net power output and the total length of heat exchanger is trade-off among the operating pressure, the turbine outlet temperature and the proportion of CO_2. In order to obtain the maximum net power and smaller volume of heat exchanger at the same time, the operating pressure should be above 11.5 MPa and the turbine inlet temperature should lower than 220 ℃ , and the proportion of CO_2 should be more than 0.8 (The additive is added to improve the performance of CO_2, so the proportion of R134a is thought no more than 20% in this paper). It can be observed that, the CMTPC can obtain better performance under the design points of 13MPa (operating pressure), 200*℃* (turbine inlet temperature), 0.85 (the proportion of CO2). Based on the optimal parameters of the design point, the CMTPC system is designed according to Fig. 7. It is worth noting that the acid dew point of exhaust is set as 120 *℃* so as to avoiding corrosion of the gas heater.

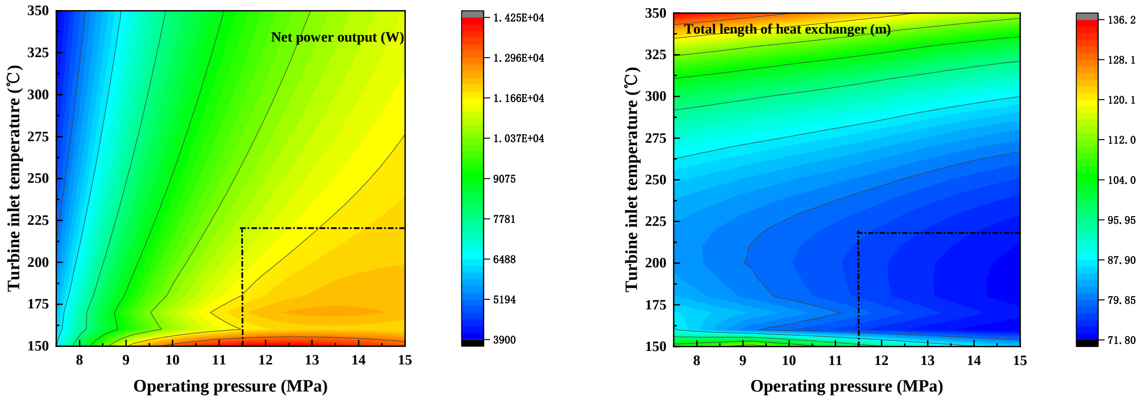

Figure 5: A two-dimensional figure of net power output and total length of heat exchanger

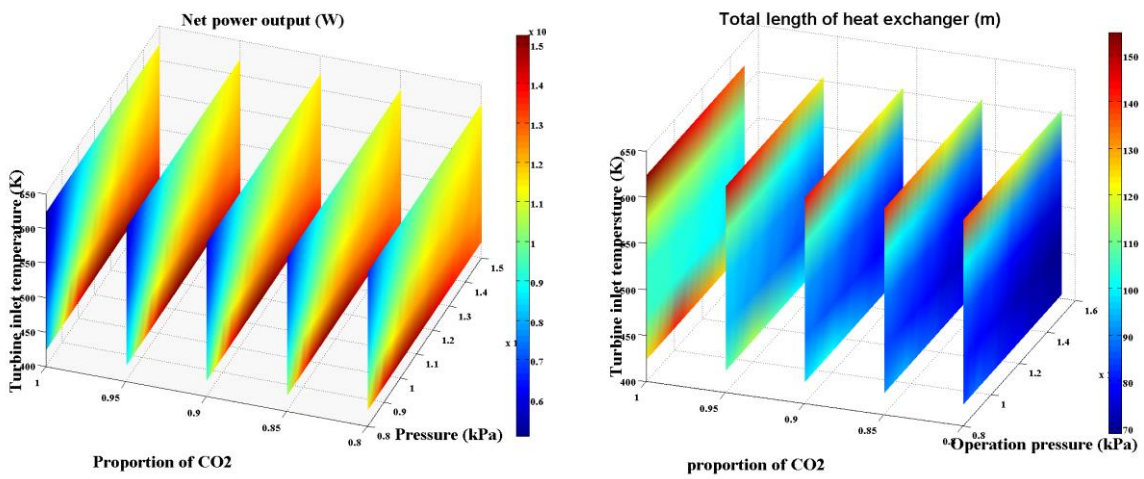

Figure 6: A three-dimensional figure of net power output and total length of heat exchanger

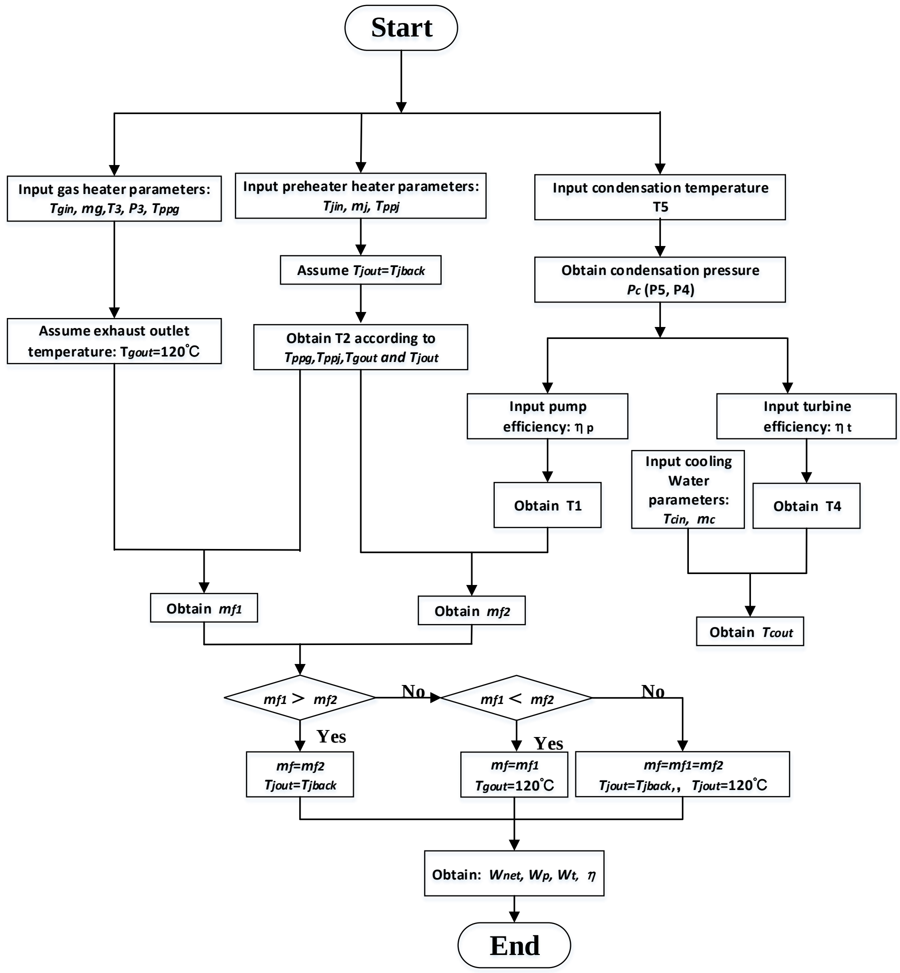

Figure 7: Block diagram of calculation strategy

4.2 Dynamic characteristics of the CMTPC system

The engine always works under different driving states according to real operation condition. In this paper, four different operation conditions represented by a serious of stable parameters as shown in Table 2 are used to observed the dynamic character of the CMTPC.

Condition 1 is the design condition, which the engine working at 80 % load and 80 % torque. The CMTPC operating condition changes from condition 1 to another condition at 300s after the system work keeps stable in condition 1 for a period of time. The change in temperature and mass flow rate of exhaust and jacket water are shown in Table 2. Fig. 8 shows the variation of net power output and operating pressure with the change of engine operating condition. It can be found that the change of the system net power output is consistent with the trend of operating pressure. The reason can be explained as follow: evaporating pressure affects the thermal efficiency, which directly decide the net power output of the CMTPC system, and the trend of operating pressure change in accordance with the available energy provided by the engine. As showed in Fig. 9, the mass flow rate of working fluid and condensation temperature basically returned to the initial value at design condition under different off-design conditions. That’s because the cooling condition has not changed nearly and the pump rotation speed state constant, so the mass flow rate almost have no change under different conditions. The temperature difference between hot working fluid and cooling water is small, so the change of condensation temperature of working fluid is slight, which causing the condensation pressure change rang is very small. The change amplitude of system’s parameter under the condition 3 is the largest because the condition 3 deviates the farthest from the design condition 1.

The total energy of exhaust and jacket water absorbed by the CMTPC system is shown is Fig. 10, and the utilization of jacket water and the net power output are also revealed. The dynamic net power output in condition 1 is 11.91 kW, when the operating condition changes at 300 s from condition 1 to 2, 3, 4, the net power output changed to 12.33 kW, 9.55 kW, 10.10 kW respectively. What's more, the energy of jacket water absorbed by the system in four conditions are 67.65 kJ,69.64 kJ,72.89 kJ and 69.34 kJ respectively, which has very small gap. That's because the jacket water temperature and mass flow rate have little difference under the four operation conditions. However, the utilization of jacket water changes obviously due to the large difference of available jacket water energy under different operating condition. That's means the thermal management system should distribute the energy reasonably according to real operating condition. This will be studied in future work.

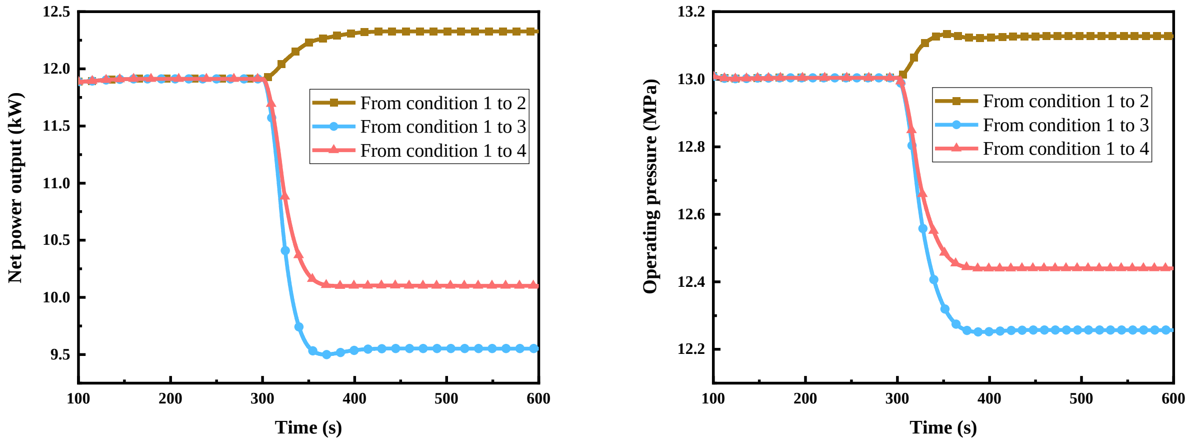

Figure 8: Variations of net power output and operating pressure

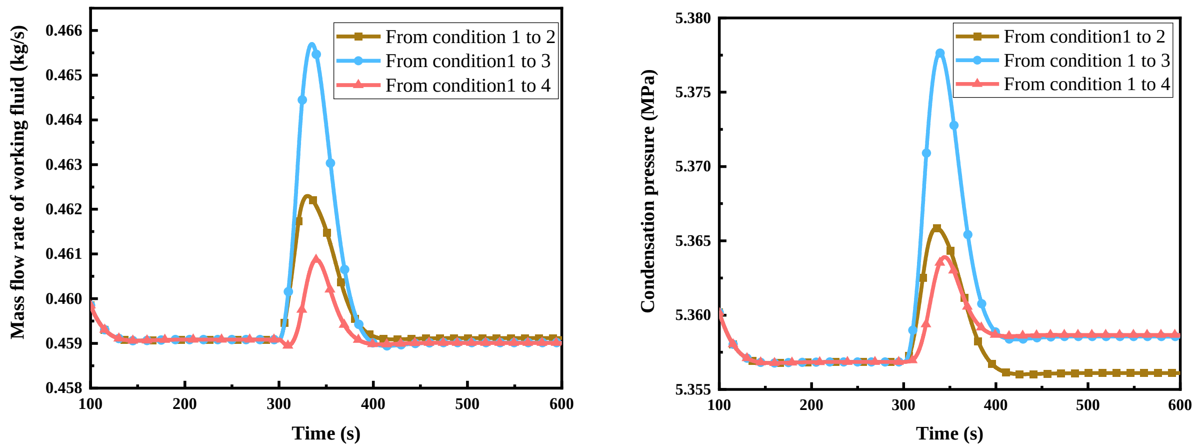

Figure 9: Variations of working fluid mass flow rate and condensation pressure

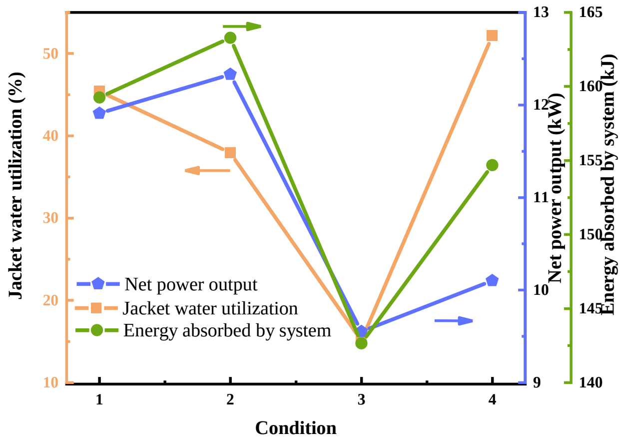

Figure 10: The analyze of four conditions

Table 2: Four typical conditions of the HDDE

Parameter	Condition 1	Condition 2	Condition 3	Condition 4
Power output [kW]	192.96	194.8	144.5	159.98
Exhaust temperature [°*C*]	389.7	381.7	339.4	367.32
Mass flow rate of exhaust [kg/s]	0.320	0.339	0.284	0.280
Jacket water temperature [°*C*]	85	87	86	84.4
Mass flow rate of jacket water [kg/s]	3.23	3.36	3.21	3.05
Energy of exhaust [kJ]	91.6	94.021	69.75	86.35
Energy of jacket water [kJ]	149.0	183.54	161.71	132.91

5. CONCLUSION

In this study, the dynamic model of the CO_2 mixture transcritical power cycle for a heavy-duty diesel engine is built by Simulink to research the dynamic performance at off-design conditions. Finite volume and moving boundary methods are adopted for heat exchangers. The design point parameters are optimized so as to obtain large power output and smaller heat exchangers at the same time. Four different engine operating conditions are chosen to observe the performance of CMTPC system.

Results show that the relationship between net power output and the total length of heat exchanger is trade-off among the operating pressure, the turbine outlet temperature and the proportion of CO_2. The system can get nice comprehensive performance when the operating pressure is above 11.5 MPa and the turbine outlet temperature is lower than 220 °*C*. When the system changes from design condition to different off-design conditions, the change the system net power output is consistent with the trend of operating pressure. What's more, the operation condition deviate from the design point farthest has the biggest change in the amplitude.

This work has certain guiding significance to the design of CMTPC system and future control work.

NOMENCLATURE

C_p	Specific heat	(kJ/kg·K)
D	Density	(kg/m^3)
A	Area	(m^2)
ρ	Density	(kg/m^3)

h	Specific enthalpy	(J/kg)
α	Heat transfer coefficient	($W/m^2 \cdot K$)
Subscripts		
c	Cooling water	
j	Jacket water	
g	Exhaust	
p	Pump	
t	Turbine	
pp	Pinch point	
Abbreviations		
CMTPC	CO_2 Mixture Transcritical Power Cycle	
WHR	Waste Heat Recovery	
FV	Finite Volume	
MB	Moving Boundary	

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