TEST ON A CO_2-BASED TRANSCRITICAL POWER CYCLE (CTPC) UNDER VARIOUS ENGINE CONDITIONS

Ligeng Li[1], Hua Tian[1], Gequn Shu[1,2] and Lingfeng Shi[2*]

[1]State Key Laboratory of Engines, Tianjin University
Tianjin 300072, China

[2]University of Science and Technology of China,
Hefei 230027, China

* Corresponding Author
Lingfeng Shi (slf@ustc.edu.cn)

ABSTRACT

CO_2-based transcritical Power Cycle (CTPC) could be used for engine waste heat recovery as the safety and environment-friendly characteristic of fluid, which also satisfies miniaturization demand of recovery systems. Through previous investigation, the results showed that waste heat of exhaust gas and engine coolant can be combined and highly recovered by the CTPC, while other ORCs show lower utilization capacity of engine coolant. In this study, a kW-level CTPC system was constructed as the bottoming system and experimentally investigated to recover waste heat from exhaust gas and engine coolant of a heavy-duty diesel engine (DE). Test was based on constant operating condition of CTPC, while operating condition of the DE considered the various engine speed and torque. The CTPC system performance changing with different waste heat conditions are main focus points of this study. Observations of key states as well as estimations and comparisons of potential output power were carried out stepwise. Results indicated that performance of CTPC showed following trend of increase with an increase in waste heat conditions, which were caused by increasing engine speed and engine torque. Specifically, a more stable heat source of engine coolant with relative temperature difference of less than 20% is obtained and an increase net power estimation of 1.9kW to 4.0kW and 2.3 kW to 4.9 kW in regard to fixed engine speed and torque can be found out. Different impact trend of engine speed and load is discovered.

1. INTRODUCTION

It is well known that more than half of the fuel energy combusted by engines is wasted through exhaust gas and coolant (Chu *et al*., 2012). Energy conservation and emission reduction has led in the past years to an increasing interest and research in the field of engine waste heat recovery (WHR). Among the WHR techniques, Organic Rankine Cycle (ORC) has been considered as a promising way due to its relatively high efficiency and low cost (Wang *et al*., 2012).

Nevertheless, three main limitations of ORCs are described as follows based on previous research. Environmental problems could be caused as ORCs normally adopt organic working fluids such as R134a (Zanelli *et al*., 1994), R123 (Mathias *et al*., 2009), R245fa (Declaye *et al*., 2013), etc. High temperature decomposition may bring about low recovery efficiency of exhaust gas, which is considered to be recovered first. To solve this, oil circuit has been added between exhaust gas and working fluid going against compactness requirement of WHR (Gewald *et al*., 2012). Additionally, a constant temperature in evaporation process may cause irreversibility loss.

In recent years, CO_2 transcritical power cycle (CTPC) has been studied worldwide due to following advantages. On one hand, CO_2 is an environment-friendly working fluid with zero ODP (=0) and low GWP (=1). On the other hand, it behaves high thermal stability when directly recovering high-temperature exhaust gas. Besides, the direct heat transfer process with continuously increased temperature of CO_2 reduces energy and exergy loss due to a better temperature match.

Thus, many researches paid attention to the performance comparison between CTPC systems and ORC systems. Due to a good heat transfer characteristics and flowing characteristics of supercritical CO_2, the CTPC shows better exchanger performance and economic performance than the ORCs (Guo *et al*., 2010). Great improvement in utilization of engine coolant can be obtained in CTPC compared with ORCs (Shu *et al*., 2016). Designed appropriately, the CTPC system could simultaneously recover engine coolant energy and exhaust energy even at both utilization rates of 100% (Shi *et al*., 2017). Meanwhile, extremely compact turbine with high efficiency and microchannel heat exchangers with 87.4% weight reduction could be achieved in CTPCs (Persichilli *et al*., 2012).
At present, the investigations on the CTPC mainly focus on the theoretical aspect, while there are not many experimental investigations on the CTPC due to its high pressure (over 10 MPa). Test benches of the CTPC using oil (Pan *et al*., 2016) and steam (Persichilli *et al*., 2011) as the simulative heat source have been carried out. Korea Institute of Energy Research developed three supercritical carbon dioxide power cycle experimental loops, namely the 1 kW-class, 10 kW-class and 80 kW-class experimental loop. Among them, the 1 kW-class experimental loop is a transcritical cycle at a maximum temperature of 200 ℃ (Cho *et al*., 2016). Alternatively, the CTPC systems are generally designed to meet certain performance requirements such as maximizing net power output or lowering costs under steady operating conditions. When applying to a real engine scenario, they need to face unsteady heat sources caused by highly transient and variable engine operating conditions. Since July 2011, transient simulation of experiment was carried out by Argonne National Laboratory (ANL).
The investigations on the CTPC test introduced above mainly used simulative heat sources instead of practical waste heat source. Using a practical waste heat source is necessary and meaningful for the study of waste heat recovery, especially for the engine’s field. When adopting the diesel engine (DE) as the bottom cycle, experimental study of four configurations of CTPC has been conducted in (Shi *et al*., 2017) to verify the functions of regenerator and the utilization of both exhaust gas and coolant. According to the experimental results, a 100.6% net power output increase can be obtained when adopting both preheater and regenerator at the same time. A higher engine speed can provide more heat leading to a higher net power output which is analyzed in (Shi *et al*., 2017). Besides, regression and predicting modeling has been used to make up the limitations of test bench based on experimental results which represents considerable value for future test direction (Shi *et al*., 2018).
The current work, based on the analysis above, mainly focuses on the dynamic performance of CTPC including power output and thermal efficiency responses under different operation conditions of DE considered the variation of both engine speed and engine torque. Based on dynamic results, the benefits of increasing engine speed and torque are obtained and responses of key parameters of CTPC are analyzed.

2. SYSTEM LAYOUT AND DESCRIPTIONS OF TEST BENCH

In this study, an 8.4 L inline 4-stroke 6 cylinder water-cooled heavy-duty DE whose rated power is 243 kW has been selected as the topping system and its main technical parameters are listed in Table 1. Full complement of controlling and measurement devices are equipped to keep engine working steadily on prospective conditions (speed and torque) as well as a perfect record and store of all the requisite parameters. The exhaust gas flows into the gas heater of the CTPC system while part of the cooling load of DE is designed to warm the preheater in CTPC system.

Table 1: Main parameters of selected DE

Parameters	Unit	Value
Engine type	-	Inline, 6 cylinder
Intake system type	-	Supercharged and intercooled
Displacement	L	8.424
Compression ratio	-	17.5
Rated power	kW	243
Rated speed	rpm	2200
Maximum rotque	N·m	1280

A kW-scale CTPC test bench was constructed as the bottoming system at the aim of recovering exhaust gas and coolant energy dissipated from the topping system. Test bench diagrams of the CTRC system integrated with the DE and photos of particle layout are shown in Figure 1. The CTPC system mainly consists of a pump, a receiver, several heat exchangers and an expansion valve, whose specifications are listed in Table 2. The working fluid is pressurized by a reciprocating plunger pump to achieve conversion from liquid state to supercritical state (namely transcritical). The pump is controlled by a frequency converter, thereby its speed could be easily changed by altering the frequency. A damper is utilized to weaken the effect of interval operating of pump and to make a more accurate measurement of supercritical flowmeter later. After pressurization, the CO_2 is heated continuously by engine coolant and exhaust gas in preheater and gas heater, respectively. The preheater is a brazed-plate exchanger while the gas heater is serpentine pipe sleeve type exchanger with exhaust gas flowing in the sleeve side and CO_2 in the pipe side. It should be noted that an expansion valve is used as a device of pressure control to temporarily replace the turbine when conducting the tests due to difficult manufacture of a small-scale CO_2 expander or turbine. Precooler, condenser and refrigeration unit are designed to make an enough cooling load before the liquid working fluid flows back to the tank. For dynamic performance analysis, a highly accurate data recording system is required to ensure all the relevant data are sampled, converted and stored. Necessary sensors are constructed to measure state parameters.

Table 2: Main components of CTPC test bench

Component	Type	Specific type	Specification
Pump	Reciprocating plunger	3RC50A-1.7/13	1.7 m^3/h
Preheater	Brazed-plate	SWEP B17x40	1.56 m^2
Gas Heater	S-shaped tube-in-tube	Self-designed	3.09 m^2
Expansion Valve	Needle valve	Self-designed	0-100%
Precooler	Brazed-plate	SWEP B18x40	1.56 m^2
Condenser	Brazed-plate	SWEP B18x40	1.56 m^2
Receiver	Cylinder-shaped	Self-designed	10L

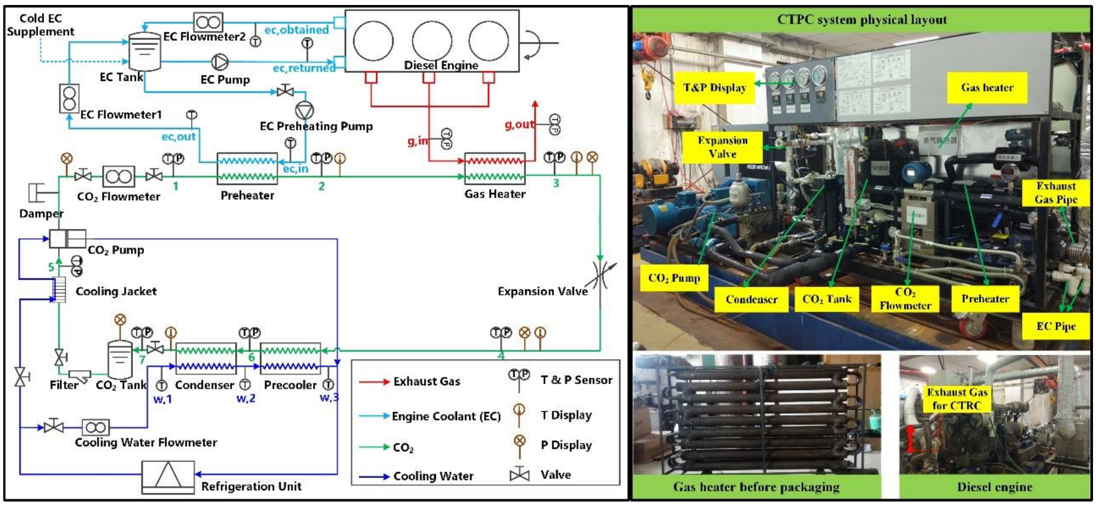

Figure 1: Diagrams and practical photos of test bench

3. EXPERIMENTAL STRATEGY

The objective of current analysis is to gain an understanding of the impact of external engine perturbations on system characteristics and responses in the absence of any control. In this study, The CTPC test bench is designed at magnitude of 4.5 kW power output, thus is much smaller than the full waste heat magnitude of DE. Hence, the DE operates at middle operation conditions to match the CTPC system. In order to compare the dynamic performance of CTPC under different engine operation conditions, the speed of 1100rpm with engine torque ranging from 30%, 40%, 50% to 60% (360N m to 720N m), and the torque of 600N m (50% load) with engine speed varying 1100rpm from 1500rpm are chosen as the engine operation conditions that are shown in Figure 2. The exhaust gas consists of several gas components (H_2O, CO_2, N_2, O_2 et al.). The property of exhaust gas is average calculated by mass proportion of each component. The mass proportion of each component is calculated by the model of prevenient experiment (Shu *et al.*, 2016), which used the same DE by our group.

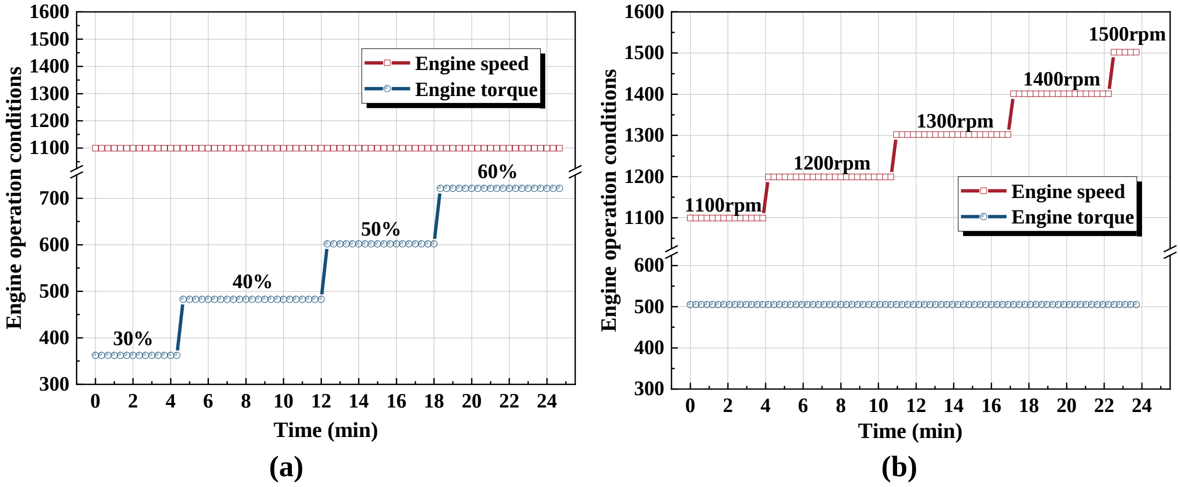

Figure 2: Specific speed and torque selected as engine operation conditions

The CO_2 pump was run by a variable frequency motor whose rotation speed was adjusted by a frequency changer. To provide a similar driven source for the CTRC system, the CO_2 pump was always working at 80 rpm for the dynamic test under selected engine conditions equally. The expansion valve was also set up as a constant suitable opening degree. For safety and reliable consideration, during all the tests, the maximum system pressure is kept below 11 MPa. As the limitation of heat resistance of the plate heat exchangers provided by the supplier is 240 ℃, the outlet temperature of the expansion valve should be less than 240 ℃. For same reason, a threshold value is considered and the maximum temperature is maintained less than 200 ℃ as much as possible.

4. RESULTS AND DISCUSSION

4.1 Heat source characteristics

As mentioned above, the various conditions of engine torque and speed are chosen to study their influence in CTPC, thus the parameters of heat source are determined directly due to the change of engine operation conditions. The inlet temperature of exhaust gas and engine coolant in CTPC is shown in Figure 3 under conditions of fixed torque and speed, respectively.

As is shown in Figure 3, a general increase of temperature of both exhaust gas and engine coolant could be obtained in the whole experimental process. Especially for the exhaust gas, when engine torque was increasing (from 30% to 60%), an evident promotion of gas heater inlet temperature was performed ranging from 270°C to more than 510°C. To further display the trend of heat source temperature, maximum relative difference is calculated between the maximum and minimum valve of inlet temperature, which is defined as:

$$RD_{\max} = \frac{value_{\max} - value_{\min}}{value_{\min}} \times 100\% \tag{1}$$

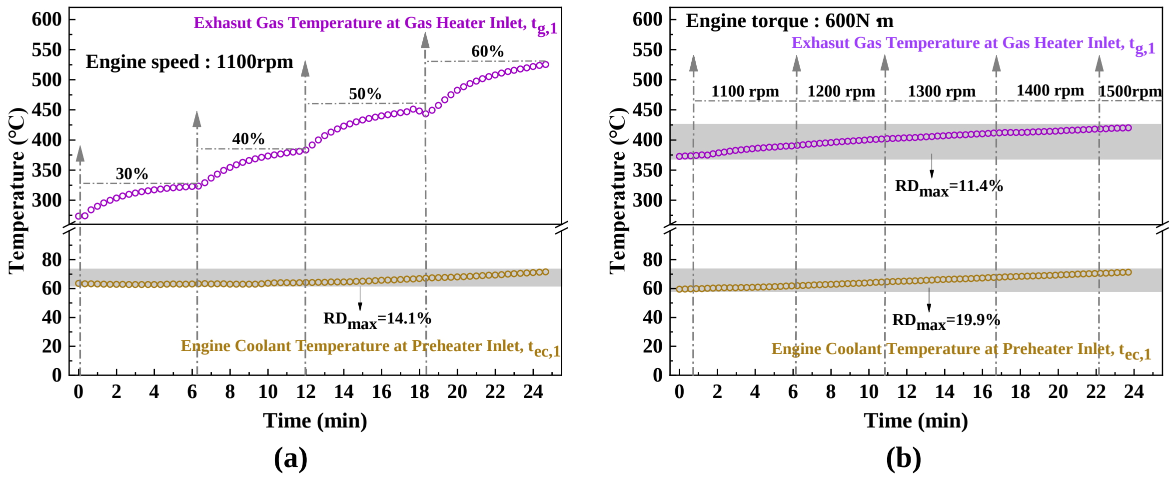

Figure 3: Dynamic response of heat source temperature

It can be seen from Figure 3 (b) that the exhaust gas temperature at gas heater inlet increases not so palpable as the one changing with engine torque shown in Figure 3 (a). The RD_{max} equals to 11.4%, which indicates that it's the engine torque nor the engine speed that affects the exhaust temperature more efficaciously. Nevertheless, the comparison of engine coolant temperature at preheater inlet both showed smooth increasing tendency under varying duty or rotational speed. Particularly from Figure 3 (a), the temperature of engine coolant shows a constant level (around 64°C) at first then increases. The reason is that with a constant mass flow of engine coolant, the auto radiator could satisfy the engine cooling requirement at first. Thus, there is a constant coolant temperature at the outset. Then the temperature increases for the reason that a higher engine torque, a higher cooling requirement is indispensable. To conclude, the temperature of both heat sources improves with the increase of engine duty or speed. Besides, the engine coolant is considered as a relatively stable heat source due to lower RD_{max} under both operation conditions.

4.2 Dynamic performance of CTPC

For a reasonable comparison of performance under various engine conditions, the key system parameters including CO_2 pump speed, expansion valve opening degree, et al, were maintained. Other external perturbations to the systems including heat sink (cooling water) flow rate and inlet temperature were also kept unchanged. To obtain the inherent properties, the response curves have been reprocessed for each step change. For further analysis about system performance, the estimated net power and thermal efficiency are selected as the target.

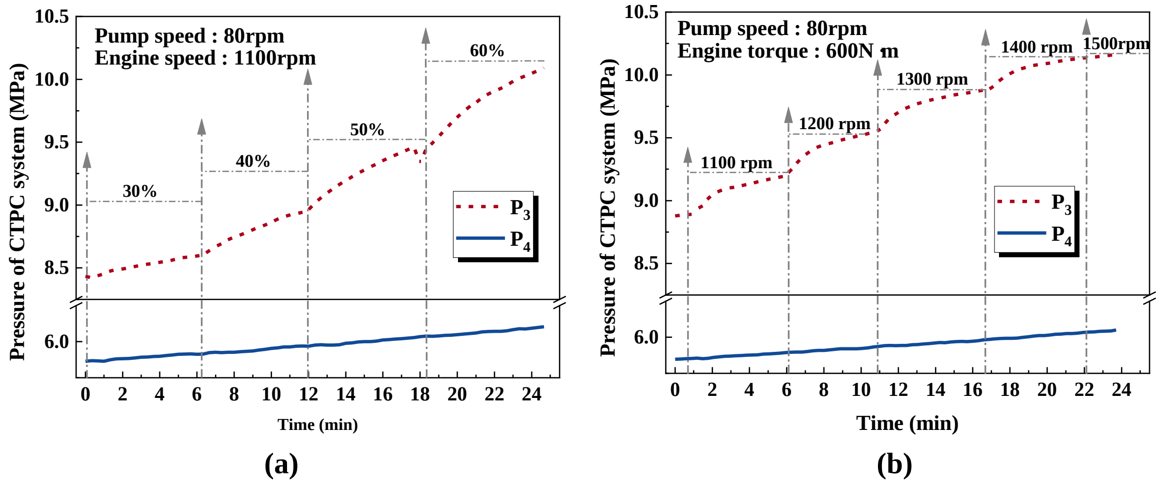

Figure 4: Dynamic responses of pressure with step changes of engine conditions

Figure 4 presents the pressures at the inlet (*P_3*) and outlet (*P_4*) of the expansion valve. Both of high and low pressure are improved with an increasing engine speed and engine torque mainly due to a higher temperature of working fluid. The reason is that a higher heat transfer rate in preheater and gas heater leads to a higher average temperature of working fluid. The maximum operation pressure both surpassed 10.1 MPa. The low operation pressure of test bench increased from 5.8 MPa to 6.1 MPa mainly due to an increasing heat sink requirement. Likewise, the pressure ratio (P_4/P_3) increases from 1.4 (fixed engine speed) and 1.5 (fixed engine torque) to 1.7. Through a comprehensive comparison of P3 under both cases, the increase of high pressure becomes more evidently with engine torque ranging from 30% to 60% while the characteristic of more stable increase of P_3 along with engine speed is found out.

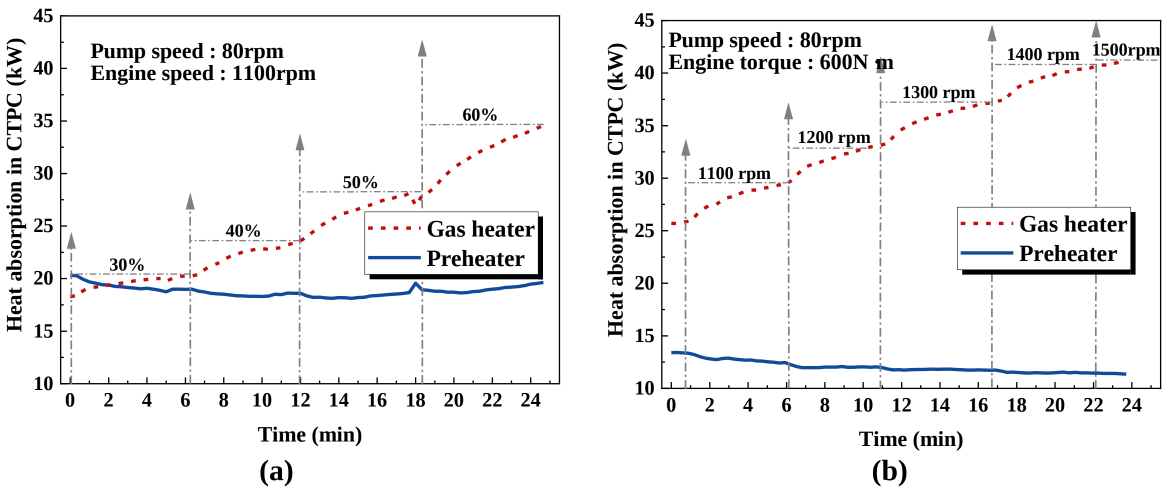

Figure 5: Dynamic responses of heat absorption with step changes of engine conditions

Figure 5 illustrates the dynamic responses of heat absorption from preheater and gas heater, respectively, which is calculated by:

$$Q_{pre} = m_f \cdot (h_2 - h_1) \tag{2}$$

$$Q_{gh} = m_f \cdot (h_3 - h_2) \tag{3}$$

$$Q_{tot} = Q_{pre} + Q_{gh} \tag{4}$$

Heat transfer in gas heater goes up under both increasing engine conditions while the one in preheater diversifies. As shown in Figure 5 (a), a slight decrease of heat transfer in preheater (from 20.3 to 18.1 kW) is shown at first then goes up to 19.6 kW moderately with the increasing engine torque. However, with a constant engine torque, the heat transfer in preheater shows a continuous decline (from 13.4 kW to 11.4 kW) even when the engine speed increases from 1100rpm to 1500rpm, which can be seen from Figure 5 (b).

Due to an expansion valve has been used as a device of pressure control, the expected net power output and thermal efficiency is necessary to evaluate a thermodynamic system which are defined as:

$$W_p = m_f \cdot (h_1 - h_5) \tag{5}$$

$$W_{t,est} = m_f \cdot (h_3 - h_{4,est}) \tag{6}$$

$$W_{net,est} = W_{t,est} - W_p \tag{7}$$

$$\eta_{th,est} = \frac{W_{net,est}}{Q_{tot}} \tag{8}$$

where the subscript est represents the value estimated.

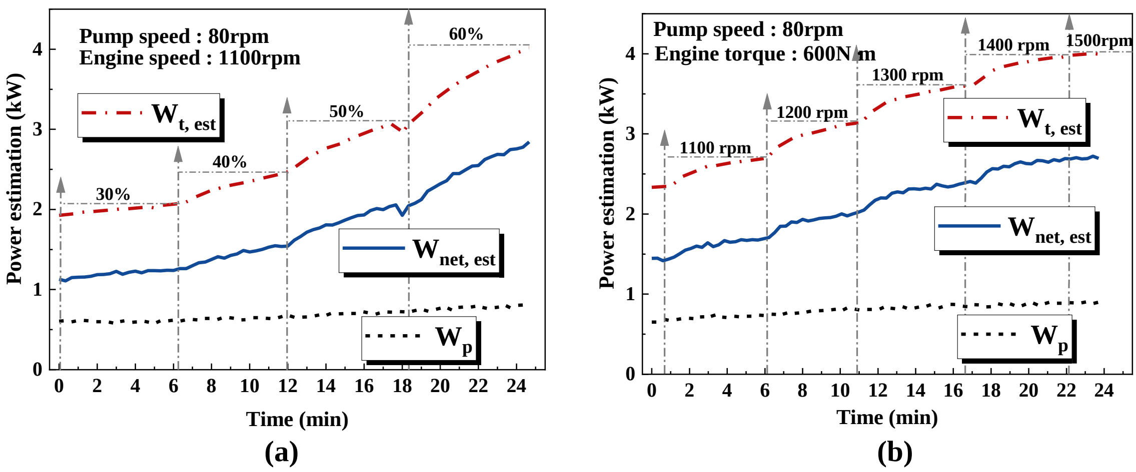

Figure 6: Various power estimation with step changes of engine conditions

With higher pressure ratio (P_4/P_3) and heat absorption mentioned above, Figure 6 (a), (b) describe the system performance of power unit including pump, expansion valve and expected net power. Considering a controlled speed of pump, namely, a constant flow of working fluid could be achieved shown in Figure7 and 8. Thus, similar tendency as pressure of $W_{t,est}$ decided by enthalpy difference can be obtained ranging from 1.9 kW to 4.0 kW and 2.3 kW to 4.9kW in regard to fixed engine speed and torque, respectively. Besides, W_p is as good as stayed with a slight increase mainly derived from a slight raise of pump inlet temperature. The increase of average net power is not even comparing the gradient of both situations. The net power is promoted by 22.6% to 33.1% from step to step during fixed engine speed where a constant increase of 15.6% can be calculated when engine torque remains unchanged.

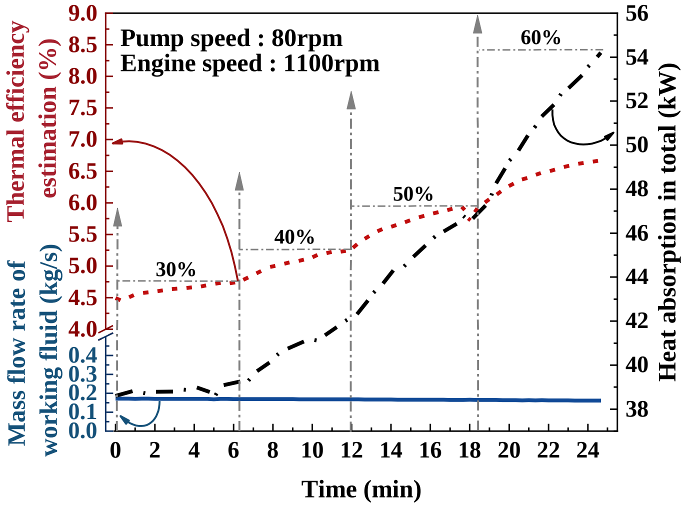

Figure 7: Various power estimation with step changes of engine load

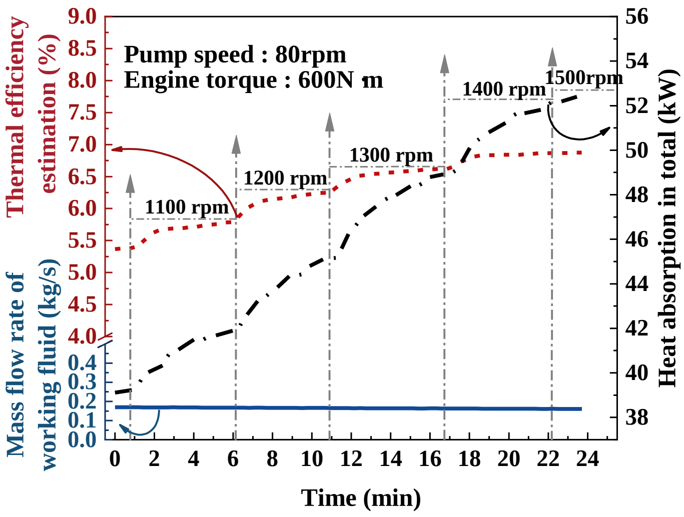

Figure 8: Various power estimation with step changes of engine speed

Analogous regulations can be found in thermal efficiency and heat absorption in total which are demonstrated by Figure 7 and 8. In this research, the test bench expected to obtain a maximum thermal efficiency of 6.7% (when engine load reaches 60%) and 6.9% (when engine speed reaches 1500 rpm), respectively.

5. CONCLUSIONS

This current work conducts the dynamic responses and CTPC characteristics on the effects of external engine perturbations including speed and torque. Investigations are completed based on experimental study. The performance of CTPC system considering net power and thermal efficiency is the target of this study. Main conclusions are summarized as follows:

- The test bench performance including net power expected and thermal efficiency estimation of CTPC presents a consecutive increase when engine speed and torque increases due to a higher pressure ratio and total heat transfer into bottoming cycle system. The maximum estimated thermal efficiency reaches 6.7% and 6.9% under fixed engine speed and torque, respectively, with the net power estimation of 4 and 4.9 kW in maximum.
- When adopting CTPC in engine waste heat recovery, the engine coolant is considered as a more stable heat source with a lower temperature and thermal energy fluctuation to hold smooth running status considering safety issue in comparison with exhaust gas especially with the change of engine load.
- Different impact trend from engine conditions on CTPC performance is discovered. The engine torque has a gradual sensitive effect on operation pressure, heat absorption and CTPC performance promoted by 22.6% to 33.1% (net power estimation) from step to step while a constant influence (15.6% promotion of net power estimation) is found out when engine speed increases.

NOMENCLATURE

η	efficiency	(%)
h	enthalpy	(kJ/kg)
m	mass flow rate	(kg/s)
P	pressure	(MPa)
Q	heat flow rate	(kW)
T	temperature	(°C)
W	power	(kW)

Subscript

ec	engine coolant
est	estimation
f	working fluid
g	exhaust gas
gh	gas heater
in	inlet
out	outlet
pre	preheater
t	turbine
th	thermal

Abbreviation

CTPC	CO_2-based Transcritical Power Cycle
ORC	Organic Rankine Cycle
SWEP	company name, a Swedish supplier of brazed plate heat exchangers

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