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ORIGINAL RESEARCH
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Analysis of CO₂-based trans-critical power cycle in waste heat recovery

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ABSTRACT

Carbon-dioxide-based trans-critical power cycle is a novel technology for waste heat recovery. This technology can handle the high-temperature exhaust gas and can be built in a compact size, which is an important feature for the auxiliary equipment for an internal combustion engine. To obtain the best output, four configurations were constructed: the basic system; one with preheater, another with regenerator and a fourth with preheater and regenerator. Special features of supercritical CO₂ make these cycles able to recover more energy than the traditional organic Rankine cycle. According to this study, heat regeneration increases thermal efficiency while preheating influences the net power output. Thus, it is beneficial to add both regenerator and preheater to the basic cycle.

KEYWORDS

supercritical fluid, anomalous properties, Widom-line, trans-critical power cycle, waste heat recovery

1. INTRODUCTION

The emissions of greenhouse gases and other pollutants caused by the combustion of fossil fuels have an undeniable effect on the climate, including ozone depletion. As a result of fuel costs and environmental concerns, a recent study has focused on developing and deploying renewable energy sources (solar heat, geothermal energy, and different forms of waste heat). Among the retrieving of waste energy technologies, the promising heat-to-power conversion method is considered as a power cycle, which has sparked a lot of research interest due to its higher system integration, consistency, and versatility. Each year, about 60–70% of hydrocarbon fuel is consumed by engine sectors in an industrial country. However, under most operating conditions, the engine coolant and exhaust source could waste more than half of the heat produced by fuel combustion in an engine, resulting in waste of energy and emissions problems. Recycling of Waste Heat (RWH) is a technique for recuperating the energy from waste that can be transferred to the environment to attain some valuable functions. It can also minimize the fuel consumption in many cases that need additional fuel energy input. The main target of WHR is to produce additional work, and a large quantity of waste heat can be converted into work from a high-quality heat source. Generally, Organic Rankine Cycle (ORC) system [1, 2] and Steam Rankine Cycle (SRC) system [3, 4] are contemplated as the most fruitful RWH techniques in the domain of low-grade heat and power generation stations, although other methods, like hot air (Stirling) engines can be also good for this purpose [4].

However, regarding waste heat of engines, ORC and SRC cannot give satisfactory results due to some particular characteristics of the engine. One of the significant drawbacks of the ORC system is its characteristic of persistent evaporation temperature, which is not adjustable for the sensitive misused heat sources [5, 6]. Among different heat sources, exhaust-gas-engine is the major recovery target of waste heat. It possesses a high quantity of heat/energy and a

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high degree of temperature region. The temperature of this targeted source can be reached up to 773.15 K for the diesel engine and 1073.15 K for the gasoline engine, respectively. Most importantly, vehicles have limited space layouts, which is a big problem for the ORC and SRC. Thus, many authors proposed CO₂-based Trans-critical Power Cycle (CO₂TPC) technology to cover up these pressure and temperature-related issues. The major focus of this research was to compare the performance of Basic system CO₂-based Trans-critical Power Cycle (B-CO₂TPC), Preheater CO₂-based Trans-critical Power Cycle (P-CO₂TPC), Regenerator based CO₂-based Trans-critical Power Cycle (R-CO₂TPC) and Preheater and Regenerator CO₂-based Trans-critical Power Cycle (PR-CO₂TPC) from various aspects.

Subcritical cycle means (see, for example, by Jahar-Sarkar et al. [7]) that the process is performed at subcritical pressures when both addition and rejection of heat occurs; also, the temperatures also stay below the critical temperature in all steps (except in cycles with super-heaters). When heat inclusion and exclusion occur at supercritical pressure, the process is known as supercritical power cycle. In this case, the temperatures are above the critical temperature all along the cycle. At very high pressures and temperatures, the supercritical fluid will be similar to gas [8]; therefore, the supercritical cycle (under those conditions) technically - although not thermodynamically - will be identical to a Brayton cycle. When heat addition occurs at supercritical pressure and heat rejection occurs at subcritical pressure, the cycle is trans-critical, although sometimes it is also referred as supercritical Rankine cycle [5]. Due to the so-called supercritical anomalies (like Widom-lines, pseudo-boiling [8, 9] a trans- or supercritical CO₂ cycle can give better efficiency than other, similar gas cycles (like Brayton cycle).

In the recovery of waste heat for Internal Combustion Engines (ICE), it is significant to inspect the performance of four CO₂TPC configurations, giving a reference of applications for the system. Currently, a number of theoretical analyses about carbon dioxide trans-critical power cycle systems can be found in the literature [10, 11], which concludes that CO₂TPC is more and more appealing in the engine sector (ICE). According to Chen et al. [12], approximately one-fifth of the energy in the exhaust gas can be extracted into work by a model developed using the CO₂TPC system's bottoming period. Autier and Kouadri-David [13] initiated an optimum study of a CO₂TPC with or without a regenerator using exhaust gas from a fishing boat diesel engine to reduce fuel dependency. The two optimized parameters in this study were pressure and temperature, and results stated that a convenient temperature had existed for each pressure in CO₂TPC system only with the regenerator. A 250 kW CO₂TPC system with a regenerator completed by Echogen Power System (EPS) in a pilot project, with temperature varying from 473.15 to 813.15 K, which also suggests the probability of recycling the waste heat of engines exhaust gas in an initial test [14]. Pan et al. [15], used a rolling piston expander to experiment on a carbon dioxide-based CO₂TPC power loop. At high pressure (11 MPa) and low pressure (4.6 MPa), the thermal efficiency was 5%, and

the corresponding maximum electric power reached about 1,100 W. Furthermore, Farzaneh-Gord et al. [11] conducted a comparative study of the four CO₂TPC configurations, finding that PR-CO₂TPC would attain a maximum net power output of 18 kW, implying that adding a preheater and regenerator could theoretically improve the net power output. Influential previous research led by Shu [10] convinced us to select four CO₂TPC configurations. Three selection maps were created for various states depending on three attributes: net power generation, energy efficiency, and the cost of electricity production. The result of his research revealed that for a 43.8 kW gasoline engine, the net power output was 9.1 kW, the exergy efficiency was 0.48, and the economic performance was 0.431 \$/kWh with PR-CO₂TPC being one of the most advantageous options with outstanding thermodynamic performances [10].

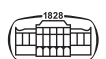
2. SYSTEM STRUCTURE AND METHODOLOGY

2.1. CO₂TPC structure

The graphical representation of the CO₂TPC system a diesel engine is illustrated in Fig. 1a. In the system, mainly three parts are described: CO₂TPC system, exhaust source (a diesel engine), coolant source loop. The topping system in this work is a diesel engine. Major technical parameters of an engine (with turbocharger) can be found in [10, 16]. Let discuss PR-CO₂TPC (Fig. 1b) as an example of CO₂-based systems. The process starts with a pump and ends with a condenser; each point is marked by letters. In the first step, the working fluid (CO₂) is pumped to a supercritical pressure (p-q), where the pump acts as a drive component. Secondly, the engine coolant in the preheater at the pump outlet (q-a) warms up the pressurized CO₂ fluid, the point (a-b) marked as the turbine outlet in the regenerator of CO₂ fluid. The high-temperature exhaust gas enters the heating coils of the evaporator (b-x). After that, the heated CO₂ fluid (already in a fully supercritical state, where both pressure and temperature are above the critical values) flows in the turbine, and the expansion work is generated (x-y). CO₂ fluid exits the turbine and reaches the heat exchanger called regenerator, which serves as regenerated thermal storage (y-z). The working fluid gets access to the condenser while coming out from the regenerator and condenses the fluid into a liquid state (z-p), making the structure p-q-a-b-x-y-z. Similarly, the flow structure of the B-CO₂TPC system is q-x-y-p, P-CO₂TPC is q-a-x-y-p, and R-CO₂TPC is q-b-x-y-z-p. On a mass basis, the mass fractions of CO₂, H₂O, and N₂ in exhaust gas compounds measured were 19.84%, 8.26%, and 71.49% [16], respectively. The temperature of the acid dew point of the engine exhaust was set at 393.15 K, and the exhaust temperature had to reach this limit in operation. The engine coolant fluid is considered in this paper as pure water.

2.2. Methods

MATLAB was used to create the mathematical model, while RefProp was used to test the fluids' thermodynamic



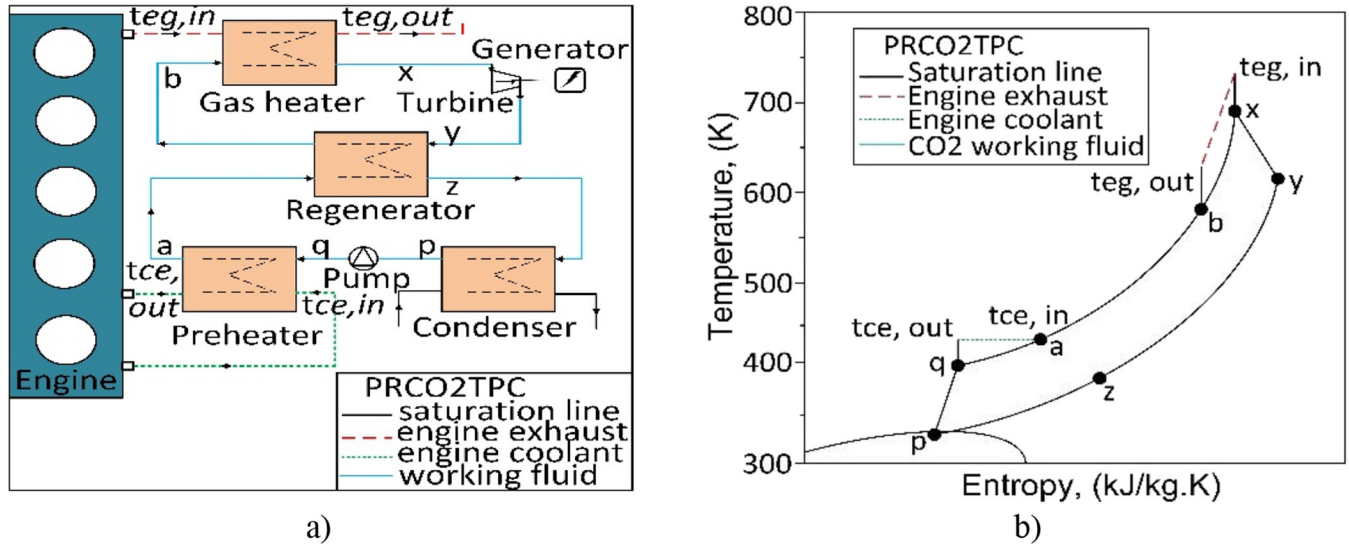


Fig. 1. a) Schematic diagram of PR-CO₂TPC system; b) T-s diagram of the system

properties. In order to build this waste heat recovery model, a few assumptions were taken, similarly to other studies on this field [9, 16, 17, 18]. Furthermore, stable states, steady-flow systems and equilibrium conditions were assumed during the process. The pressure drops in the pipe were neglected as well as heat loss by the radiation. The generator's efficiency was set to 0.9, the turbine's isentropic efficiency was set to 0.7, and the pump's efficiency was set to 0.8, with the condensation temperature set to 298.15 K, which allows CO₂ to be converted to a saturated liquid state.

The determination of the thermodynamic condition of each point is the basis of this model; hence, it is important to quantify the mass flow rate of the working fluid in advance for this case. The usual method to calculate the mass flow rate m_{fr} and thermodynamic state condition is called "Temperature Difference at Pinch Point (TD@PP)" [18]. For the heat exchange in the evaporator, the TD@PP is set at 30 K.

The calculation approach used for both ORC and CO₂TPC had been developed by other authors [17, 18]. Generally, PR-CO₂TPC and P-CO₂TPC contained two waste heat sources, which mean there are two mass flow rates. One is exhaust gas ($m_{fr,1}$), and another is engine coolant ($m_{fr,2}$) which can be obtained by TD@PP method when both are supposed to approach maximum heat recovery rate. Generally, two mass flow rates can be represented by the following Eqs [17, 18]:

$$m_{fr,1} = \frac{m_c \cdot C_{p_{ec}} \cdot (t_{c,in} - t_{c,out})}{(h_a - h_q)}, \quad (1)$$

$$m_{fr,2} = \frac{m_g \cdot C_{p_{eg}} \cdot (t_{g,in} - t_{g,out})}{(h_x - h_b)}. \quad (2)$$

where m is the mass, c_p is the isobaric specific heat, t is the temperature, h is the enthalpy and concerning indices, fr is the flow rate (1 for coolant, 2 for exhaust), c is the coolant,

in and out is the inlet and the outlet, a , q , x and b are states marked in Fig. 1b.

From Eqs (1) and (2), when $m_{fr,1}$ and $m_{fr,2}$ is equal, this quantity of mass flow rate fulfils both the highest usage of exhaust gas and the engine coolant. But, generally, $m_{fr,1}$ and $m_{fr,2}$ are often dissimilar. Therefore, choose the lowest number between $m_{fr,1}$ and $m_{fr,2}$ as the final mass flow rate to meet one complete usage among the two waste heats. On the other hand, this mass flow rate adjustment is unnecessary for the regenerator-based system and the basic system that uses exhaust gas as the only heat source.

2.3. Calculation by 1st law of thermodynamics

In this study, the cycle components' performance and energy stability are examined according to the first law of thermodynamics. Coding was done based on the system's mathematical model by using the MATLAB software. The equations of each component for the entire scheme are shown in Eqs (3)–(9),

$$W_{pmp} = m_{fr} (h_q - h_p) = m_{fr} \cdot (h_{qs} - h_p) / \eta_{pmp}, \quad (3)$$

$$Q_{ph} = m_{fr} (h_a - h_q) = m_c \cdot C_{p,c} \cdot (T_{c,in} - T_{c,out}), \quad (4)$$

$$Q_{rg} = m_{fr} (h_b - h_a) = m_{fr} (h_y - h_z), \quad (5)$$

$$Q_{ghe} = m_g c_{p_g} (t_{g,in} - t_{g,out}) = m_{fr} (h_x - h_b), \quad (6)$$

$$Q_{cnd} = m_{fr} (h_z - h_p), \quad (7)$$

$$Power = m_{fr} (h_x - h_y) \cdot \eta_{gen} = m_{fr} \cdot (h_x - h_{ys}) \cdot \eta_{trb} \cdot \eta_{gen}, \quad (8)$$

$$W_{npo} = Power - W_{pmp}, \quad (9)$$

$$\eta_{te} = W_{npo} / (Q_{ghe} + Q_{ph}), \quad (10)$$

where, in Eq. (3) W_{pmp} represents the pump, the subscript qs means the outlet state of the pump after isentropic compression, the subscript fl is the CO_2 fluid, h_q and h_p is the enthalpy difference between q and p . Equation (4) Q_{ph} represents the heat in preheater, subscript c means the engine coolant and c_{in} and c_{out} are the state of inlet and outlet of preheater, respectively. In Eq. (5) Q_{rg} represents the heat in the regenerator, in Eq. (6) Q_{ghe} is the heat in gas heater, subscript g_{in} and g_{out} are the entry and outer state of exhaust gas in gas heater. In Eq. (7), Q_{cnd} represents the condenser. In Eq. (8) the $Power$ represents turbine and generator, the subscript ys is the outlet state of turbine after isentropic expansion. In Eqs (9) and (10) W_{npo} is the net power output and η_{te} is the thermal efficiency.

However, a percentage of waste heat recovery consumption is also measured to assess the degree of waste heat recovery. According to Shu and Shi [15], if the exhaust gas temperature is reduced to the acid dew point and the engine coolant temperature is lowered to the engine coolant's return temperature, the exhaust gas and engine coolant may be used properly. That is why the utilization rates were also investigated in this work.

3. RESULTS DISCUSSION

The main simulation work output and performance comparison among the four CO_2 TPC system configurations (e.g. B- CO_2 TPC, P- CO_2 TPC, R- CO_2 TPC and PR- CO_2 TPC) was discussed in this part. Comparative performance of net power production and efficiency in terms of heat has been evaluated with some other indicators. In addition, turbine inlet pressure (P_3) had been considered as the variation input (ranging from 10 to 20 kPa), while turbine inlet temperature (T_3) was chosen as 500 K and 700 K for all the discussions.

3.1. Waste heat utilization

Reduction and utilization of waste heat in general can significantly reduce the power consumption. It is important to calculate the utilization rate due to its positive impact on the environment. This could be possible when the exhaust gas utilization rate is maximal, resulting in a lower exhaust temperature. The cooling load of the engine can be minimized by using engine coolant (U_{ce}) at a higher rate, as this allows the engine cooling system to hit a lower temperature. The total use of exhaust heat source and coolant source of the engine is defined as the temperature of the exhaust gas being reduced to the acid dew point and the temperature of the coolant being reduced to the returning temperature of the engine coolant. Among four CO_2 TPC systems, basic configuration (B- CO_2 TPC) is not included in this analysis due to its structure. In this system, the consumption rate of both exhaust gas U_{eg} and engine coolant U_{ce} expanded as turbine inlet pressure increased (with U_{eg} increasing and U_{ce} decreasing). The distinction was made between the exhaust and coolant (R- CO_2 TPC and P- CO_2 TPC with the PR- CO_2 TPC) configurations for constant turbine inlet

Table 1. Summary of waste heat utilization rates by exhaust and engine coolant supply

Configurations	Utilization rate by U_{eg} , U_{ce}			
	550 K, 10 kPa	550 K, 20 kPa	700 K, 10 kPa	700 K, 20 kPa
R- CO_2 TPC (U_{eg})	0.93	0.935	0.55	0.66
PR- CO_2 TPC (U_{eg})	0.35	0.89	0.25	0.58
P- CO_2 TPC (U_{ce})	0.86	0.27	0.51	0.17
PR- CO_2 TC (U_{ce})	1.00	0.38	0.64	0.23

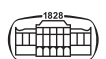
temperatures of 700 and 550 K. Table 1 summarizes the utilization rate results for all scenarios. Overall, R- CO_2 TPC obtained higher U_{eg} than PR- CO_2 TPC system in both conditions. But, PR-based CO_2 system showed higher U_{ce} than P-based system for the same conditions.

3.2. Thermal efficiency and net electricity output

In Fig. 2a, the net power output of all devices increases as the turbine inlet pressure improves, which is attributed to an increase in the enthalpy gap at the expander. Here, PR- CO_2 TPC, R- CO_2 TPC, P- CO_2 TPC, and B- CO_2 TPC initially shows around 12.80 kW, 11.50 kW, 10.0 kW, and 7.0 kW (for $P_3 = 10$ kPa) for turbine inlet temperature state 700 K. Consecutively, for $P_3 = 20$ kPa they reached around 23.05 kW, 22.5 kW, 18.0 kW and 15.6 kW, respectively, where PR- CO_2 TPC dominated over other configurations in the whole pressure range. In the case of the low-pressure range, R- CO_2 TPC was slightly higher than the other CO_2 TPC configurations, and this was caused due to low turbine inlet pressure where the gas heater mainly used up the exhaust gas. In addition, the mass flow rate intensity of the working fluid slowly decreases by proceeding forward. As a result, the net power output steadily declines. On the other hand, the variance of thermal efficiency demonstrated an upward trend with the increase of turbine inlet pressure. According to Fig. 2b, the thermal efficiency acquired by R- CO_2 TPC is 0.229, while PR- CO_2 TPC achieved 0.197, and both B- CO_2 TPC and P- CO_2 TPC remain constant throughout the range. Here, R- CO_2 TPC dominates due to adding the regenerator; also, the percentage of the recovered heat is elevated with the increase of pressure. Moreover, the regenerator itself can be able to recover some internal heat, which is the main reason for its higher thermal efficiency output. Despite the fact that PR- CO_2 TPC had a greater net power yield, but it adsorbs the heat energy from the cooling jacket and gets inferior to R- CO_2 TPC in terms of thermal efficiency. Interestingly, both the B- CO_2 TPC and P- CO_2 TPC remain constant as both possessed similar m_{fr} (mass flow rate) of the working fluid (CO_2) on the basis of their configurations.

3.3. Performance of CO_2 TPC under various engine conditions

When adopting waste heat recovery systems on the particle application, the operation status of the engine differs most of the time. The different engine conditions may cause a great effect on the waste heat recovery system. Thus, a detailed



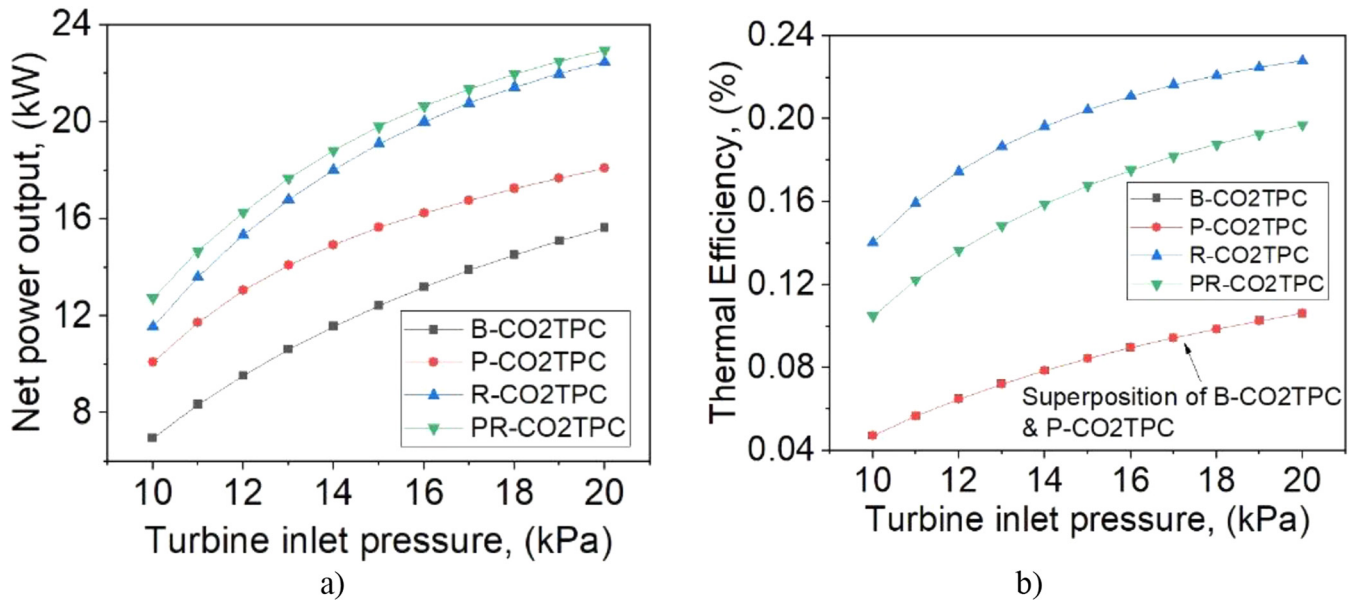


Fig. 2. a) Variations of net power output and b) thermal efficiency with turbine inlet pressure

analysis should also be considered when working conditions of engine changes. The detailed parameters of engine conditions are 211, 316.2, 422, 526, 631.2, 736.4 and 841 Nm (torques) and 48.16, 72.36, 96.74, 120.72, 145.02, 169.08 and 193.38 kW (powers), respectively. With the increase of engine torque (from conditions 1–7), both engine coolant and exhaust gas have increased thermal power, ranging 40.0–190.5 kW with the engine torque changing from 211 to 841 Nm. In contrast to exhaust gas, the rise in heat energy of the engine coolant is more unstable. The temperature of the exhaust source reaches around 370°C with a constant rising as engine torque increases, while the temperature of the engine coolant at first increases randomly then decreases to nearly 81°C.

Considering the discussion above, the relative CO₂TPC performance had analyzed based on first law of thermodynamics. The power output from the turbine increases with the increase of engine torque (Fig. 3a). Here, PR-CO₂TPC owned the highest turbine power output, while the basic one performed the worst. When engine torque increases from 211 to 841 Nm, the power output in the turbine also increases from 6.6–25.9 kW, mainly due to the increase of m_{fr} of CO₂ as well as enthalpy difference.

Meanwhile, in Fig. 3b, it can be seen that the thermal efficiencies of B-CO₂TPC and P-CO₂TPC are equal under the entire working range of the engine. The explanation for this is that in B-CO₂TPC and P-CO₂TPC, the mass flow rate of the operating fluid is determined by the exhaust gas, and

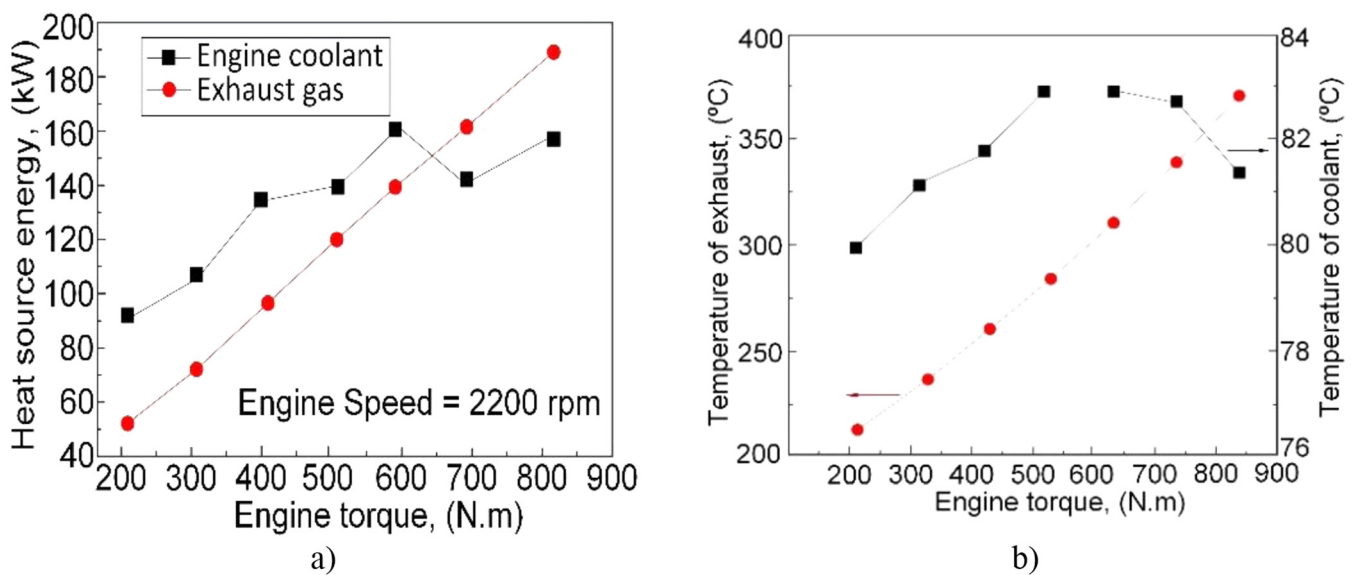


Fig. 3. Variance of a) net power output, b) thermal efficiency (%) with the engine torque



the engine coolant consumption rate has reached its limit. Another interesting point is that the thermal efficiency of R-CO₂TPC are higher than for PR-CO₂TPC although the PR-CO₂TPC attains the maximum net power output.

4. CONCLUSIONS

The theoretical performances among the four CO₂-based configurations, namely: basic, preheater, regenerator, and preheater-regenerator-based system compared extensively in the field of waste heat recovery of internal combustion engines, especially in the case of heavy-duty diesel engines. The study of analysis focuses on the rate of waste heat consumption and net power generation, and the performance of CO₂TPC under various engine operating conditions was compared. The following major conclusions can be distinguished:

- The waste heat utilization rate was investigated for both the exhaust (U_{eg}) and engine coolant (U_{ce}). In the case of exhaust gas, R-CO₂TPC outraced PR-CO₂TPC ($0.66 > 0.58$, for 700 K, 20 MPa). Whereas, for engine coolant (U_{ce}), PR-CO₂TPC achieved higher than P-CO₂TPC ($0.23 > 0.17$, for 700 K, 20 MPa);
- PR-CO₂TPC has obtained the highest net power output among the four CO₂TPC configurations. It achieved 23.05 kW under 700 K and 20 MPa, which was followed by R-CO₂TPC, P-CO₂TPC and B-CO₂TPC, respectively. But at lower T_3 (500 K) and P_3 (10 MPa), R-CO₂TPC owned slightly higher than PR-CO₂TPC;
- The performance of CO₂TPC configurations on various engine's factors had also been evaluated. In other words, PR-CO₂TPC demonstrated higher attributes in the turbine's overall productivity and working fluid's mass flow rate. In terms of thermal efficiency, however, R-CO₂TPC outperformed the other devices;
- Finally, PR-CO₂TPC has the most profound characteristics compared to the basic system in the field of WHR of internal combustion engines. However, this system can be improved in further study under sophisticated supervision by switching trans-critical to supercritical state or modifying components pump or turbine, reducing system size etc.

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