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Performance of Heat Recovery Cycle in order to Enhance Efficiency and its Mutual Effect on the Engine Performance with the Aid of Thermodynamic Simulation

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A B S T R A C T

Considering that the heat required for the Waste heat recovery (WHR) cycle of the engine is provided from two parts of the exhaust gas and the cooling system, the mutual influence of the WHR cycle on the engine performance is undeniable. Therefore, in this numerical study, an attempt has been made to thermodynamically evaluate the effect of the implementation of the WHR cycle on the engine efficiency. For this purpose, the 16 cylinder MTU 4000 R43L heavy diesel engine was simulated and a comparison was made between numerical and experimental results. Finally, the SRC heat recovery cycle was designed and applied in the simulated model according to the desired limits and the temperature range of the engine operation. At low speed with the application of the WHR cycle, the output net power did not drop much, but at the maximum speed and power, a power loss of about 4% is observed. At 1130 rpm, the power did not increase much. At 1600 rpm, the power increase is reduced to about 2.3%. At 1800 rpm, due to the significant increase in exhaust gas temperature, the total power value increased by about 4%.

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INTRODUCTION

Heavy diesel engines are used in power generation, commercial vehicles and ship propulsion (1). According to the United States EPA report in 2014, about eight billion metric tons of CO_2, equivalent to 24% of total greenhouse gas (GHG) emissions, are produced by the transportation sector (2). Environmental researchers and policymakers have identified greenhouse gas emissions from heavy-duty engines as a growing problem and one of the main causes of environmental pollution (3). Currently, approximately 50% of combustion heat in internal combustion engines is discharged to the environment along with exhaust gases and cooling system fluid, etc. (4). By recovering this part of the heat, the fuel consumption and therefore the production of greenhouse gases will be reduced in proportion to the produced power (5). In this regard, there are two ways to reduce fossil fuel consumption. One of the approaches is the development of renewable energy sources such as biofuel (6), geothermal energy (7), solar energy (8) the use of clean working fluid or CHP and CCHP systems (9). In a case study, Singh and Pedersen (10) investigated various heat recovery technologies for heavy duty diesel engines. This technology uses waste heat from diesel engines with the help of steam Rankine cycle (SRC), organic Rankine cycle (ORC), Kalina cycle (11), absorption refrigeration and their combination and converting it into electrical energy, thermal energy or mechanical energy. Steam Rankine cycle is a conventional technology of WHR cycle (12), but its thermal efficiency is very low at temperatures below 200 °C. However, the ORC cycle uses waste heat at lower temperatures (13). Therefore, to implement a heat recovery cycle, the temperature range of exhaust air and engine cooling water is very important.

Yang and Yeh (14) investigated the characteristics of an ORC cycle for a marine diesel engine. They evaluated

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the performance of various fluids; The results of their studies showed that the operating fluid R1234yf has the best performance. Feng et al. (15) investigated how the Rankine cycle works to recover waste heat from the exhaust gas. Their study focused on evaluating the additional power achievable in internal combustion engines as a heat source with a simple heat recovery unit design. They also evaluated the effect of different operating fluids on the heat recovery cycle efficiency. Also, to determine the optimal working fluid, R123, R245ca, R245fa, water and butane were analyzed numerically. Their studies showed that the use of water as the operating fluid increases the efficiency of the heat recovery cycle. By using other mentioned working fluids, cycle efficiency decreases with increasing expander inlet temperature. The results of investigating the thermodynamic characteristics of working fluids show that among these fluids, if the temperature of the exhaust gas is high, water has the highest efficiency; But when the outlet temperature is low or variable, organic liquids are more suitable. Bu Tao Liu and his colleagues (16) analyzed the performance characteristics of ORC cycle using different working fluids; The results of their investigation show that the thermal efficiency for different liquids depends on the critical temperature. The efficiency of the heat recovery cycle increases with an increase in the temperature of the heat loss source and decreases if operating fluids with a lower critical temperature are used. Wei et al. (17) designed a medium temperature WHR system to use of the diesel engine waste heat. Also, they investigated the effects of engine parameters on the organic Rankine cycle. In this study, R123 was chosen as the operating fluid for ORC cycle. The results of their study showed that that under normal load conditions, the ORC cycle is more efficient. They also showed that the Rankine cycle system achieves a significant waste heat recovery efficiency of 10-15% for the design of the optimal heat exchanger, but the Rankine cycle has no effect on the waste heat recovery at a temperature of less than 300 °C of the exhaust gas. Anderson et al. (18) compared Rankine steam and organic Rankine cycles. In this article, two SRC and ORC systems were compared based on a turbine with equal efficiency. The steam Rankine cycle reaches a higher output power for high engine loads, but the ORC cycle reaches a higher power for the engine operating mode with a lower load. SRC will be able to achieve higher efficiency for high engine loads; therefore, its process is more complicated than ORC.

According to the previous studies conducted in the field of WHRS of internal combustion engines, in all these studies, the effect of various cases and engine operating conditions on the performance of the recovery cycle was investigated. Considering that the heat required for the WHR cycle of the engine is provided from two parts of the cooling system and the exhaust gas, the mutual influence of the WHR cycle on the engine performance is undeniable. Reducing the temperature of the exhaust gas and absorbing its heat for the recovery cycle causes a pressure drop in the exhaust pipe and affects the performance of the engine. Therefore, in this numerical study, an attempt has been made to thermodynamically evaluate the effect of the implementation of the heat recovery cycle on the performance of the engine. In this regard, the 16 cylinder MTU 4000 R43L heavy diesel engine was simulated and a comparison was made between numerical and experimental results. Finally, the SRC heat recovery cycle was designed and applied in the simulated model according to the desired limits and the temperature range of the engine operation.

MATERIAL AND METHODS

In this research, GT-Power software is used for engine simulation. The basis of the analysis in GT-Power software is the one-dimensional and thermodynamic analysis of the equations of conservation of mass, energy, torque difference and momentum (equations 1 to 5). Due to the one-dimensional nature of the analysis, all parameters are calculated as average values along the flow (19).

$$\frac{dm}{dt} = \sum \dot{m} \tag{1}$$

$$\frac{d(me)}{dt} = -\frac{dV}{dt} + \sum(\dot{m}H) - hA_s(T_{fluid} - T_{wall}) \tag{2}$$

$$\frac{d(\rho HV)}{dt} = \sum(\dot{m}H) + V\frac{dP}{dt}hA_s(T_{fluid} - T_{wall}) \tag{3}$$

$$\frac{d\dot{m}}{dt} = \frac{dPA + \sum(\dot{m}u) + 4C_f\frac{\rho u|u|}{2}\frac{dxA}{D} - C_p\left(\frac{1}{2}\rho u|u|\right)A}{dx} \tag{4}$$

$$dT = \frac{|\sum(T+) - \sum(T-)|}{2 \times min[T+, T-]} \tag{5}$$

In the above equations, the solution variables are pressure (P), mass flow rate ($\dot{m}$), temperature (T), total enthalpy (H), torque ($\mp$ T), convective heat transfer coefficient (h), velocity (u) and the coefficient of friction (C_f).

Combustion model

Vibe model is used for combustion simulation; in which the amount of energy released inside the cylinder is calculated by calculating the flame speed (20). In this model, ignition delay is 4, Premixed Fraction is 0.14, Tail Fraction is 0.1, Premixed Duration is 5, Main Duration is 27 and Tail Duration is 35. In Tables 1 and 2 the specifications related to engine injector and the calculation formulas in Vibe model are presented, respectively.

Considering that in this engine, intercooler is used to decrease the temperature of the engine's intake air, so to check the performance of the intercooler in the software,

Table 1. Injector specifications (21)

Specifications	Value
Hole diameter	0.270
Number of holes	8
Discharge coefficient	0.70
Parameter	Value

Table 2. Calculation formulas in Vibe model (21)

FM	Main Fraction	$F_M = (1 - F_P - F_T)$
WCP	Wiebe Premix Constant	$WC_P = \left[\frac{D_P}{2.302^{1/(E_P+1)} - 0.105^{1/(E_P+1)}}\right]^{-(E_P+1)}$
WCM	Wiebe Main Constant	$WC_P = \left[\frac{D_M}{2.302^{1/(E_M+1)} - 0.105^{1/(E_M+1)}}\right]^{-(E_M+1)}$
WCT	Wiebe Tail Constant	$WC_P = \left[\frac{D_T}{2.302^{1/(E_T+1)} - 0.105^{1/(E_T+1)}}\right]^{-(E_T+1)}$

the air mass flow rate and its temperature are simulated. The air passing through the compressor is used as input to the simulation module. Cooling was considered. Intercooler simulated model in GT-Power software is shown in Figure 1. Also, according to Equation 6, the outlet temperature of the cooler was calculated.

$$Air\ to\ Air - InterCooler\ Efficiency = \frac{T_{outCompressor} - T_{outIntercooler}}{T_{outCompressor} - T_{Ambient}} \quad (6)$$

In this research, the considered engine was completely simulated by considering all the accessories of the engine. Figure 2 shows the half view of engine simulation model.

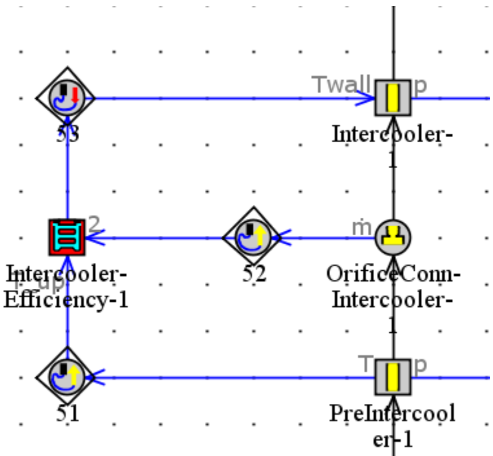

Figure 1. Intercooler simulated model in GT-Power software

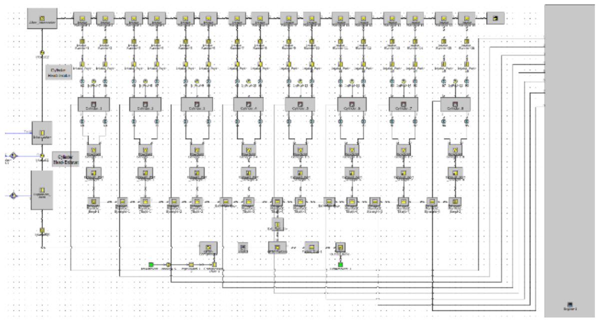

Figure 2. Half view of engine simulation model

Validation

Considering that the aim of the current research is to evaluate the possibility of improving the power and marine use of the MTU4000 R43L engine; therefore, at first, values obtained from numerical simulation were validated. this engine is a heavy-duty 16-cylinder V-type diesel engine used in rail applications. The engine characteristics are summarized in Table 3. The maximum speed is 1800 rpm and its maximum output power is 2400 kW. Figure 3 and Table 4 are related to the schematic and parameters related to this engine, respectively.

Table 3. Engine characteristics (21)

Parameters	Value
Number of cylinders	16V
Power	UIC 2400 kW
Engine rated speed	1800 RPM
Configuration	90°
Bore/stroke	170.0/210.0 mm (6.70/8.30 in)
Cylinder volume	4.77 l (291 cu in)
Total displacement volume	76.3l0 (4656.0 cu in)
Fuel characteristics	DIN EN 590

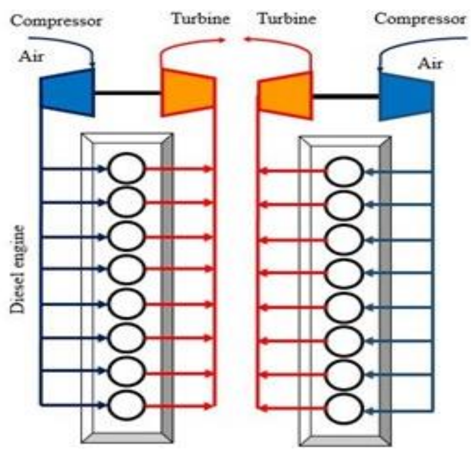

Figure 3. Schematic picture of the intended engine

Table 4. One-dimensional calculation error (21)

Parameter	Simulation Error
Power	1.6%
BMEP	3.26%
BSFC	2.32%

To check the validity, the numerical simulation results were compared with the experimental results at different engine speeds between 600 and 1800 rpm. Therefore, the characteristics of the considered diesel engine such as power, average brake pressure and fuel consumption of special brakes are taken into account. Figures 4, 5 and 6 show the comparison of power, BMEP and BSFC with the experimental results of the base engine. The calculation error of this simulation is shown in Table 5. It can be concluded that this simulation is valid.

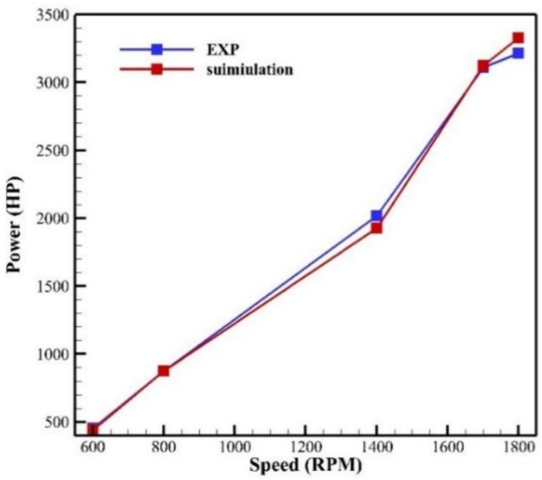

Figure 4. Checking the reliability of the simulated engine braking power

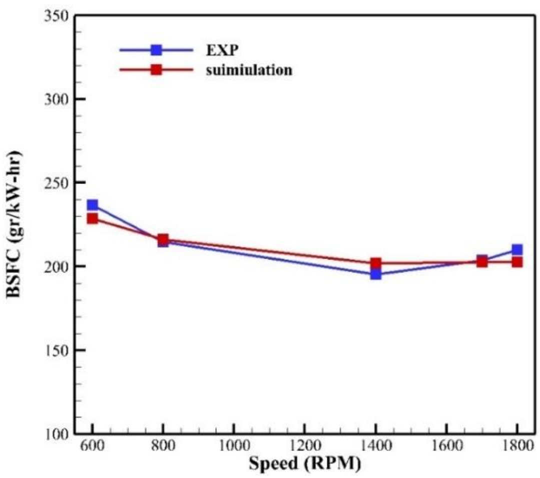

Figure 5. Checking the validity of BSFC values

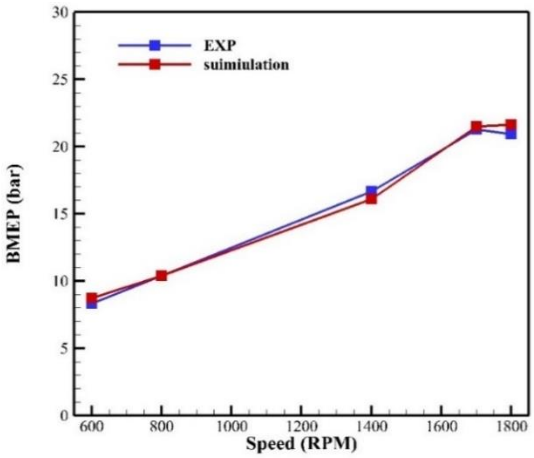

Figure 6. Checking the validity of BMEP values

It is worth noting that in the current study, waste heat of the outlet air and cooling system fluid (water) were used to implement the heat recovery cycle, so the circuit of the engine cooling system was also modeled and connected to the engine. In the next step, the tools and equipment of the heat recovery system were designed according to the temperature limit. According to Kocsis et al. (22) , the amount of back pressure of the engine in power of 50 kW, 50-500 kW and more than 500 kW is equal to 40 kPa, 20 kPa and 10 kPa, respectively. The lowest possible temperature for exhaust after WHRS is given in Table 5.

On the other hand, in order to prevent damage to the engine, the temperature limit of the engine cooling system was considered in such a way that the temperature of the water entering the engine block and the intercooler before and after applying the heat recovery system are exactly the same.

RESULTS AND DISCUSSION

In the present numerical study, an attempt has been made to evaluate the results of using the heat recovery cycle on the thermodynamic performance of the engine.

Table 5. Exhaust air temperature limitation based on back pressure

Speed (RPM)	Gas pressure at the turbocharger outlet (Kpa)	The minimum temperature of the exhaust gas (°C)
600	100.041	29.01
800	100.1634	29.12
1180	100.604	29.71
1500	101.547	74
1600	102.1865	150
1800	102.949	192

Therefore, first, the effect of adding the WHR cycle on the net output power and engine torque has been investigated (Figures 7 and 8).

According to the figure, by applying the thermal recovery cycle, the power of the engine has not decreased significantly at low speed, but at maximum speed and power, a power loss of about 4% is observed. This issue can have various reasons, including the increase in the temperature gradient of the exhaust outlet air. In the following, various parameters are analyzed to evaluate this power reduction.

Considering that the engine speed is equal in two modes with and without the heat recovery cycle, as shown in Figure 8; the torque value has been reduced in turn with the decrease in power.

The speed of the pump used in the regenerator cycle changes according to the engine crank speed, that is why the best performance mode for choosing the pump is the condition where the temperature of the charge air at the maximum speed and power of the engine along with the regenerator cycle is equal to the value of the intake air temperature of the base engine. According to the connections of the heat recovery cycle on the cooling system, it can be seen that the charge air temperature is lower at low speeds. But, when the engine speed increases, the charge air temperature is equal in both engine modes. By increasing the temperature of the charge air, power and the formation of premature ignition increase. However, due to the fact that the charge air temperature has not increased from the base state, therefore, the net power loss of the engine cannot be considered as the result of the effect of the WHR cycle on the intercooler efficiency. The effect of WHR system on charge air temperature and exhaust gas pressure drop are shown in Figures 9 and 10, respectively. The decrease in charge temperature at high speed is less than 2 degrees Celsius, which has a little effect on engine performance.

One of the important points in the implementation of the heat recovery cycle is the amount of pressure drop in the exhaust pipe after heat recovery. As the exhaust air temperature gradient increases, the pressure drop inside the exhaust increases. As seen in Figure 10, with the increase in power, the temperature of the exhaust outlet

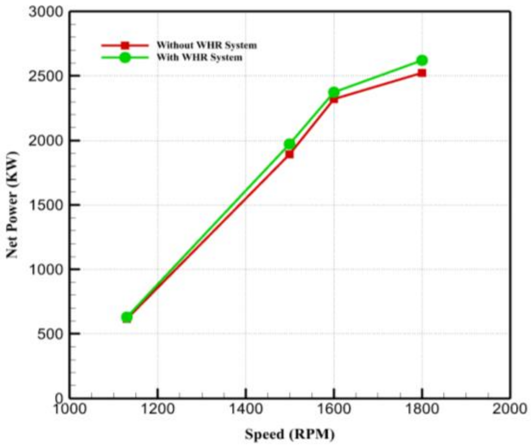

Figure 7. The effect of WHR system on net power

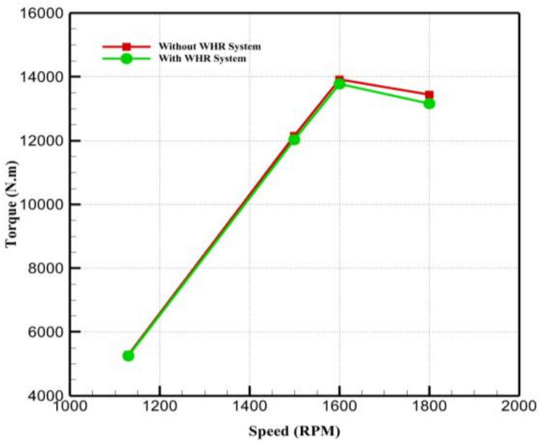

Figure 8. The effect of WHR system on engine torque

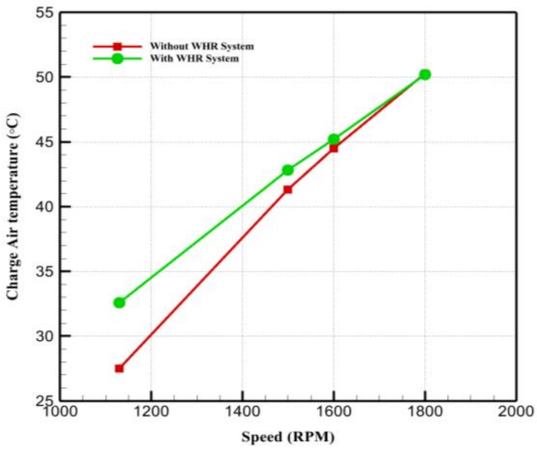

Figure 9. Effect of WHR system on charge air temperature

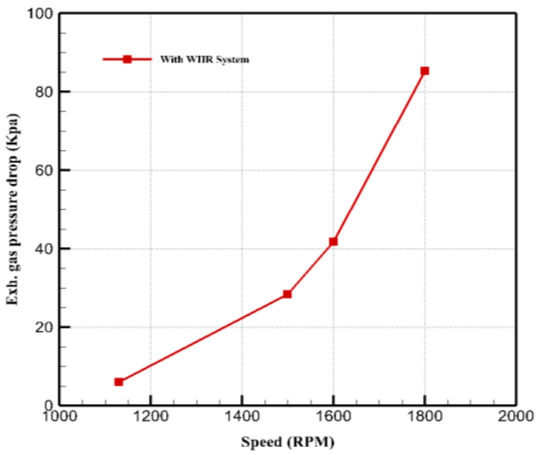

Figure 10. The effect of WHR system on exhaust gas pressure drop

air increases and in case of heat extraction, the temperature gradient increases. At the maximum power of the engine, this pressure drop is about 85 kPa. The main reason for power reduction of the engine if the heat recovery cycle is added is the pressure drop in the engine exhaust pipe, which disrupts the exhaust gas discharge process and reduces the power. Figure 11 shows the pump power used in the Rankine cycle.

In Figure 12, the amount of increase in the total engine power in the case of applying the heat recovery cycle at different speeds has been investigated. Considering that the temperature of the exhaust and cooling system is low at 1130 rpm, in this case, the power increase is not significant. At 1500 rpm, Exhaust gas temperature increases and heat received from the exhaust increases significantly. On the other hand, the pressure drop in the exhaust pipe in this case is around 25 kilopascals, which has little effect on the engine performance. For this reason, at 1500 rpm, the maximum power increase is about 4.3%. At 1600 rpm, according to the amount of power consumed by the pump, the increase in the exhaust pressure drop is almost twice as compared to the previous state and the low temperature of the outlet air, the amount of power produced is less due to the power loss and the amount of power increase is reduced to about 2.3%. In the 1800th round, despite the significant pressure drop of 80 kPa and also the significant amount of power consumed by the heat recovery cycle pump, due to the high exhaust gas temperature, the amount of increase in the flasher increases again and the total amount of power increases by about 4%.

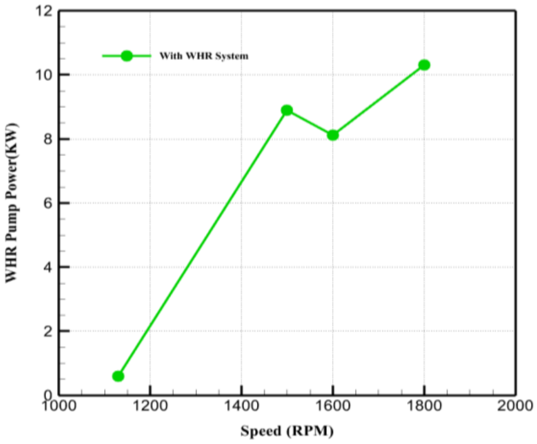

Figure 11. Pump power used in the Rankine cycle

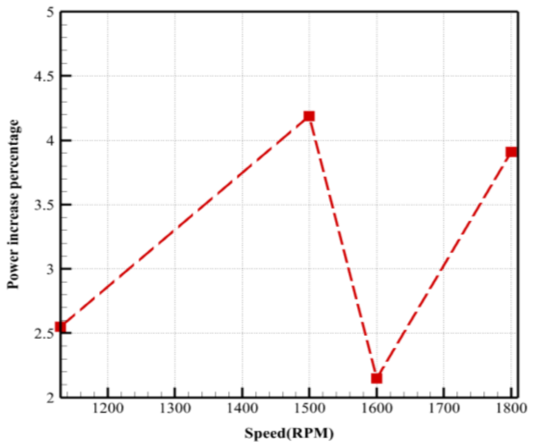

Figure 12. Effect of using WHR cycle on the percentage of engine power increase

CONCLUSION

In this numerical study, an attempt has been made to evaluate the influence of the implementation of the WHR cycle on the performance of the engine thermodynamically. In this regard, the 16 cylinder MTU 4000 R43L heavy diesel engine was simulated in GT Suite software and the values obtained from the numerical simulation were compared with the experimental results. Finally, the SRC heat recovery cycle was designed and applied in the simulated model according to the desired limits and the temperature range of the engine operation.

- At low speed with the application of the WHR cycle, the output power of the engine has not decreased significantly, but at maximum speed and power, a power loss of about 4% is observed.
- After applying the WHR cycle, the charging temperature at high speed drops below 2°C, which has little effect on engine performance.
- by increasing the temperature gradient of the exhaust air, the pressure drop inside the exhaust increases. With the increase in power, the exhaust air temperature increases and in case of heat extraction, the temperature gradient increases. At the maximum power of the engine, this pressure drop is about 85 kPa. The main reason for power reduction of the engine if the heat recovery cycle is added is the pressure drop in the engine exhaust pipe, which disrupts the exhaust gas discharge process and reduces the power.

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Persian Abstract

چکیده

با توجه به اینکه حرارت موردنیاز برای سیکل بازیاب حرارت اتلافی موتور از دو بخش سامانه‌ی خنک‌کاری و گاز خروجی اگزوز تأمین می‌شود، تأثیر متقابل سیکل بازیاب حرارتی بر عملکرد موتور غیرقابل‌چشم‌پوشی است. کاهش دمای گاز خروجی اگزوز و جذب حرارت آن برای سیکل بازیاب، باعث افت فشار در لوله‌ی اگزوز شده و عملکرد موتور را تحت تأثیر قرار می‌دهد. لذا در این مطالعه‌ی عددی تلاش شده است تا اثر پیاده‌سازی سیکل بازیاب حرارتی بر روی عملکرد موتور نیز به‌صورت ترمودینامیکی مورد ارزیابی قرار گیرد. در این راستا موتور دیزل سنگین ۱۶ سیلندر MTU 4000 R43L شبیه‌سازی شد و نتایج شبیه‌سازی عددی با نتایج تجربی مقایسه گردید. درنهایت سیکل بازیاب حرارتی SRC با توجه به محدودیت‌های موردنظر و بازه‌ی دمایی عملکرد موتور طراحی و در مدل شبیه‌سازی‌شده اعمال گردید. نتایج نشان داد در دور پایین با اعمال سیکل بازیاب حرارتی مقدار توان خالص خروجی موتور به میزان قابل توجهی افت نکرده است، اما در دور و توان بیشینه، افت توانی در حدود ۴٪ مشاهده می‌شود. در دور ۱۱۳۰ دور بر دقیقه میزان افزایش توان قابل توجه نیست. در دور ۱۶۰۰ مقدار افزایش توان به حدود ۲/۳ درصد تقلیل می‌یابد. در دور ۱۸۰۰ علیرغم افت فشار قابل توجه ۸۰ کیلوپاسکال و همچنین مقدار قابل توجه توان مصرفی پمپ سیکل بازیاب حرارت، به دلیل دمای بالای گاز اگزوز، مقدار توان کل در حدود ۴ درصد افزایش می‌یابد.