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Power Enhancement of a Heavy-Duty Rail Diesel Engine Considering the Exhaust Gas and ancillary facilities Temperature Limitation: A Feasibility Study

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ABSTRACT

One of the important features of the heavy-duty internal combustion engine is power density in such a way that the limitations created by the engine's features and accessories are the main challenges in evaluating the performance and power enhancement of advanced diesel engines. In other words, the complexity and limited performance of some of these devices do not allow the use of different power enhancement methods. Among these limitations, temperature constraints are one of the main challenges in the power enhancement process. In this study, the feasibility of increasing the power of the R43L MTU4000 heavy rail diesel engine has been considered. In this regard, the limitations of turbocharger inlet temperature as one of the basic performance challenges of the engine have been investigated using a one-dimensional simulation. For validation, the simulation results from the GT-SUITE software are compared with the experimental results. In the results section, the influence of increasing fuel mass, decreasing the compression ratio (CR), and the start of injection timing (SOI) has been investigated. The results show that by raising the fuel quantity by 5%, the power increases by about 7.6%; however, this increase in power leads to an increase in the turbocharger inlet temperature by 20K. Due to the operating limitations of various engine systems, attempts were made to control the rise of exhaust gas temperature by reducing the CR. On the other hand, reducing the CR from 18 to 15 increases the BSFC by 2.5%, but these changes in the CR do not have a significant effect on the output power. Finally, to examine the SOI timing in the enhanced engine at the maximum speed and power, different SOIs are tested and the optimal point is determined.

1. Introduction

Heavy-duty diesel engines are the dominant means of propulsion in the transportation industry, owing to their high efficiency and reliability. The performance and structure of these engines are various in different applications. Therefore, to change the application of internal combustion engines (ICEs), it is necessary to pay attention to their essential features. One of the main characteristics of heavy-duty diesel engines is their high-power density. With the minimum weight and the lowest possible dimensions, these engines provide the power required to propel a vessel [1]. To change the application or increasing the efficiency of heavy-duty diesel engines, first, it is necessary to enhance the engine power based on its dimensions to reach the target power. Nowadays, according to high technology ICEs, to enhance the power, the performance limitations of different engine systems should be considered [2]. Many methods are used in this field [3]. The existing techniques for increasing the engine power depend on changing the mass of inlet air and fuel and adjusting their ratio [4]. Much research has been done to improve the engine intake air condition [5, 6], but it is clear that to enhance the power of engines, more fuel should be injected into the combustion process [7]. By the controlled increase of fuel-to-air ratio in the combustion process, raises the maximum in-cylinder pressure, which leads to higher power output considering the start of combustion timing and engine performance [8]. The rise of in-cylinder pressure may have destructive effects, such as increased nitrogen oxide emissions and higher exhaust gas temperatures. It is noteworthy that to improve the power of heavy engines, due to the existence of unique and expensive equipment such as turbochargers, there are many challenges, including operating temperature limits. These challenges can be addressed through the proper selection and design of engine performance characteristics. In this regard, with the help of various simulation and testing processes, optimum engine performance can be achieved [9, 10]. Several methods have been proposed to control the operating temperature range of the engine [11-15]. Reducing the compression ratio (CR) is one of the best ways to reduce the combustion temperature of the engine and the temperature of the exhaust gases. Mallamo et al. [16] experimentally investigated the effect of two CR values on the in-cylinder temperature and emissions from a diesel-fueled CI engine. They observed that with a decrease in the CR, the exhaust temperature is reduced while the ignition delay is raised. They found that reducing CR or retarding injection timing significantly reduces NOx emissions while it leads to slightly higher CO and HC emissions. To increase the thermal efficiency, Funayama et al. [17] theoretically compared two different CRs of 17 and 26 in a diesel engine. They found that as the CR increases, the unburned hydrocarbon emission decreases, and the rate of heat release increases. Therefore, reducing the CR can reduce the combustion temperature and increase the unburned HC emission in the engine. Omar et al. [18] evaluated the impact of CR on the performance and emissions characteristics of a diesel engine and reported that CR is one of the main parameters to raise the engine efficiency. Jagannath et al. [19] operated experiments on a diesel engine with direct injection and show that the growth of CR from 16.5:1 to 17.5:1 results in the rise of thermal efficiency by 5.41%, 6.59%, and 7.25% with B70, B50, and B0 fuels, respectively. They reported that the main reasons for the increased thermal efficiency are lower ignition delay and higher in-cylinder temperature. Stobart et al. [20] investigated the effect of changing CR in a variable CR (VCR) engine using a 1-D simulation in the GT-POWER software. They found out that to minimize fuel consumption and soot emissions, greater CR is needed. They stated that if the percentage of EGR and the correct position of the variable-geometry turbocharger (VGT) are maintained, the BTE may be improved without increasing the NOx emission. In the subsequent studies on reducing the CR, the effects of fuel injection strategy and changing the CR were examined. The intersection of all these researches was that in addition to the decrease in combustion and exhaust air temperatures, reducing the CR raises the ignition delay [16, 21-23]. Therefore, during the power enhancement process of the ICE, it is necessary to investigate the influence of fuel injection timing to control the start of combustion. It is noteworthy that the change of start of fuel injection timing has various effects on engine power and performance [24-26]. Jayashankara and Ganesan [27] carried out a numerical investigation on a diesel engine to study the influence of fuel injection timing on engine performance and pollution. By performing some alternations compared to the base state, the NOx and CO2 emissions were increased. On the other hand, with earlier fuel injection, the ignition delay decreased while the peak of in-cylinder mean pressure and temperature were increased by around 12.5%. Rosa et al. [28] used a single-cylinder research engine to experimentally determine the effects of fuel injection strategies and injection timings on combustion, emission characteristics, and engine performance. They observed that advanced injection timings lead to higher ROHR in the early combustion stages. With an advanced SOI, the BMEP, and brake thermal efficiency (BTE) were increased, while the brake-specific fuel consumption (BSFC) and exhaust gas temperature were significantly reduced. Also, lower CO2 and unburned HC emissions were observed. In a numerical study using GT-POWER software, Ahmed et al. [29] investigated the effect of SOI on engine performance and emission and determined the optimum operating point for a turbocharged six-cylinder diesel engine. First, the simulation results were compared with the experimental tests, and the calculation results were validated with an error of less than 4%. The simulations were performed at four separate SOI timings (5o, 10o, 20 o, and 25o CA bTDC) and constant engine speed (1800 rpm). According to their finding, a delayed injection reduced NO2 and CO while its increased HC and CO2 emissions. The results also showed that an early injection timing (20o and 25o CA bTDC) reduces the CO2 unburned HC emissions and also increases the thermal efficiency of the engine.

One of the important methods to increasing the power density of IC engines is to reach the target power. Among the various methods to raise the power of heavy-duty engines, increasing the mass of injected fuel is considered the least expensive method. Literature review shows that increasing the mass of injected fuel leads to an increase in the combustion temperature and, of course, growth in exhaust gas temperature, which is one of the main challenges in power enhancement of engines due to the temperature and structural limitations of modern engines. Therefore, in this study, enhancement in the output power of the MTU4000 R43L rail engine was investigated to evaluate the possibility of raising the power density by increasing the mass of injected fuel while considering the temperature and construction limitations of the engine. In this regard, to reduce the combustion temperature of the engine, an attempt was made to make a balance between the reduction of exhaust (and combustion) temperature by reducing the

CR and decrease of engine power to maintain the required output powe. Furthermore, according to the studies on the effect of CR on combustion timing, to prevent the power loss caused by the formation of premature combustion, an attempt was made to evaluate the impacts of the SOI timing on the combustion timing and output power. Due to the large dimensions of the engine, a large number of available cylinders and effective devices, including the turbocharger and intercooler system in the main structure of the engine, and considering the complexity of thermodynamic calculations related to the engine performance, GT-POWER software has been used for numerical modeling of the engine.

2. Method of solution

In this study, GT-suite software was used to simulate the engine. The analysis basis of this software is the 1-D and thermodynamic solution of the mass conservation, momentum, energy, and torque difference equations (Equations 1-5). According to the 1-D solution, all parameters are calculated as the mean value [30].

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

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

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

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

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

In the governing equations, the parameters of mass flow (m ̇), pressure (P), and total enthalpy (H) are considered as the main variables of the solution. Also, T represents the temperature, ∓ T is the torque, h is used to express the convection heat transfer coefficient, u indicates the velocity component, and C_f is friction factor.

2.1. Combustion model

The Wiebe model is used to simulate combustion. In this model, the rate of heat release (RoHR) is calculated by calculating the flame speed [31]. Tables 1, 2, and 3 present the input parameters of the Wiebe combustion model, the specifications of the engine injector, and the computational parameters of the Wiebe method, respectively.

Table 1. The input parameters of the Wiebe combustion model

Parameter	Value
Ignition Delay	4
Premixed Fraction	0.14
Tail Fraction	0.1
Premixed Duration	5
Main Duration	27
Tail Duration	35
Premixed Exponent	0.7
Main Exponent	1.1
Tail Exponent	1.7

Table 2. The specifications of the engine injector

Parameter	Value
Nozzle Hole Diameter	0.27
Number of Holes per Nozzle	8
Nozzle Discharge Coefficient	0.7

Table 2. Computational parameters of the Wiebe method [31]

FM	Main Fraction	$\boldsymbol{F_M = (1 - F_P - F_T)}$
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	$WC_P = \left[\frac{D_T}{2.302^{1/(E_T+1)} - 0.105^{1/(E_T+1)}}\right]^{-(E_T+1)}$

According to Figure 1, the MTU4000 engine uses an intercooler to reduce the temperature of the exiting air from the compressor, so to simulate the performance of this cooler in the GT-Power software, the temperature of the exiting air from the compressor and the mass flow of air passing through the compressor were considered as the inputs of the cooling simulation module. Then, based on Equation 6, the wall temperature at the cooler outlet was calculated.

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

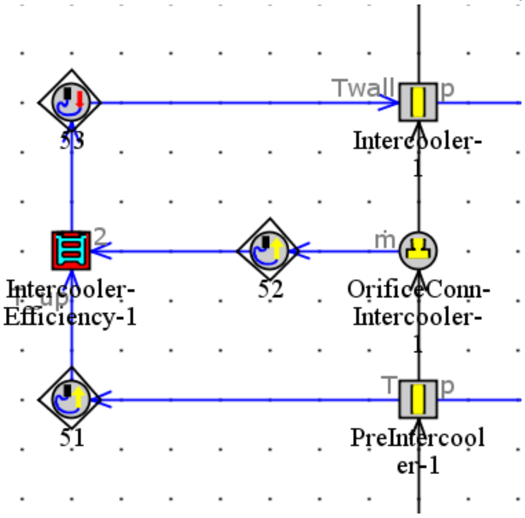


Figure 1. Simulated model of engine inter-cooler

3. validation

In this study, to reduce the calculation error, the considered 16-cylinder engine was completely simulated, taking into account all the engine accessories. Figure 2 shows the general view of the final engine model.

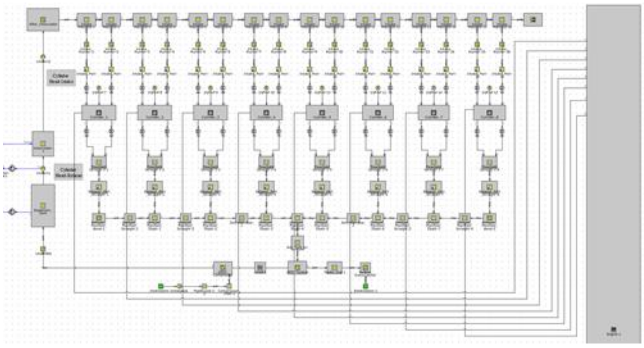


Figure 2. Half of the general view of the engine simulation model

Given that this study aims to evaluate the feasibility of enhancing the power output of the MTU4000 R43L engine, in the first step, by simulating the base engine, the accuracy of the simulation results is verified. The MTU4000 R43L engine is a 16-cylinder V-shaped heavy-duty diesel engine that has applications in rail use. The maximum power of this engine is about 2400 kW, and its maximum speed is 1800 rpm. Figure 3 and Table 4 show the configuration and technical specifications of the engine, respectively.

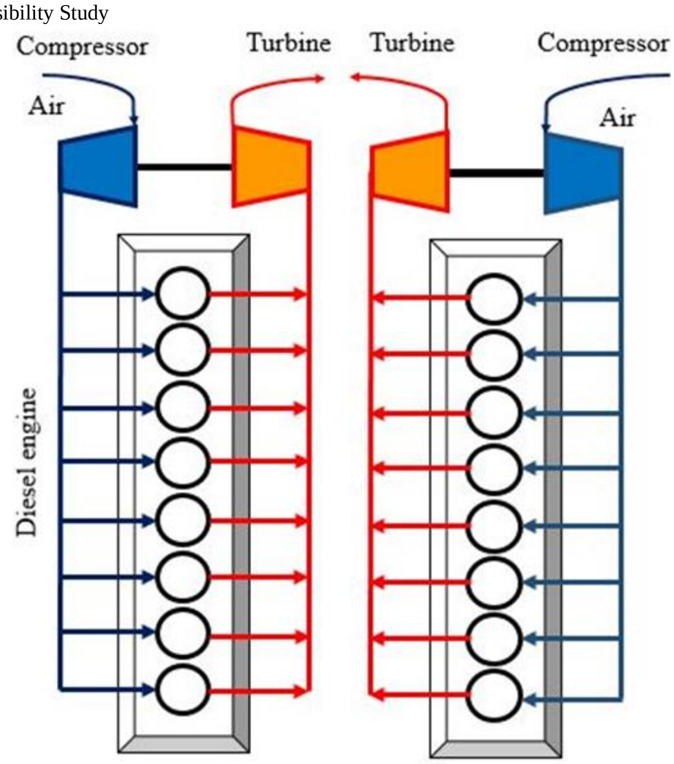


Figure 3. Schematic diagram of the engine

Table 4. The technical specifications of the engine

Parameters	Value
Number of cylinders	16V
Power	UIC 2400 KW
Engine rated speed	1800 rpm
Configuration	90°V
Bore/stroke	170/210 mm (6.7/8.3 in)
Cylinder displacement volume	4.77 l (291 cu in)
Total displacement volume	76.3 l (4656 cu in)
Fuel properties	DIN EN 590

For validation, the simulation results at several engine speeds ranging from 600 to 1800 rpm were compared with the experimental results. In this regard, the performance characteristics of the heavy-duty rail diesel engine (MTU4000 R43L), including power, BMEP, and BSFC have been considered. Figures 4, 5, and 6 show a comparison between the calculated values of output power, BMEP, and BSFC with the available experimental data from the MTU4000 R43L engine, respectively. According to the average computational error presented in Table 5, it can be observed that the simulation results have sufficient validity.

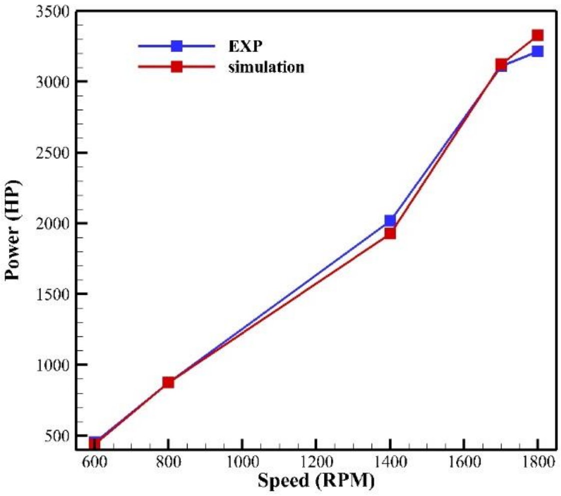


Figure 4. The validation of output power

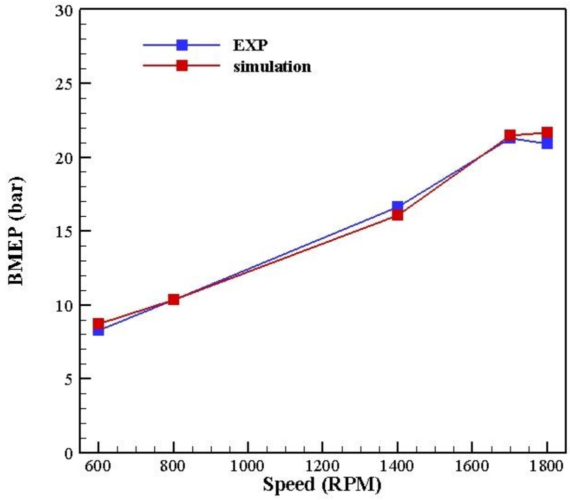


Figure 5. The validation of BMEP

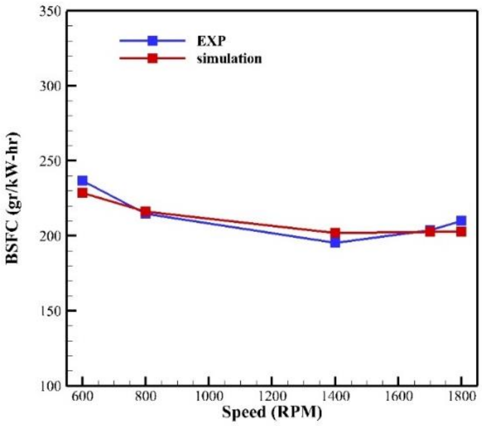


Figure 6. The validation of BSFC

Table 4. Error of validation

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

3. Results and discussion

In this study, the feasibility of enhancing the output power of a heavy-duty diesel engine by increasing its injected fuel mass has been investigated, so it is necessary to consider all the factors influencing the rise in fuel mass in the engine. If the structural and performance limitations of the engine are not taken into account, the increase in fuel mass may be harmful to various parts, including the cylinder block and accessories. For this purpose, in this study, at first, the effect of increasing fuel mass on the engine performance is evaluated, but to control the performance parameters of the engine within the defined limits, the exhaust temperature is examined at each situation, then to control the combustion temperature and efficiency, the engine performance is compared in the different CR. Finally, to prevent knock and loss of engine output power, an attempt is made to modify the start of combustion timing according to the performance of the base engine by changing the SOI timing.

3.1. Turbocharger performance curves

The operating points of the turbocharger compressor are shown on the compressor isentropic efficiency contour in Figure 7. The horizontal axis shows the corrected mass flow rate, the vertical axis shows the pressure ratio, and the drawn contours show the isentropic efficiency of the compressor. As can be observed, the operating points are far enough from the surge and suffocation lines. On the other hand, at high speeds, the operating points are in the range of maximum compressor efficiency. This turbocharger can pass more flow to produce higher power in the engine.

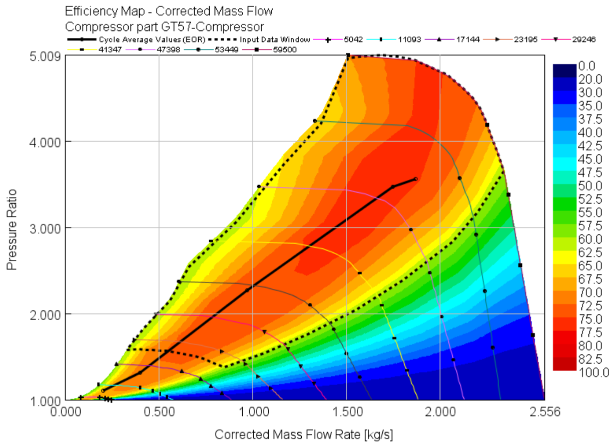


Figure 7. Operating points of the turbocharger compressor on the compressor isentropic efficiency contour

The operating points of the turbocharger turbine on the isotropic efficiency contour of the turbine are shown in Figure 8. The vertical axis indicates the reduced flow rate, the horizontal axis shows the pressure ratio, and the drawn contours are the isotropic efficiency of the turbine. Considering the shape, pressure, and reduced flow rate of the base engine is in the operating range of the engine. It is also observed that it is possible to increase the turbine pressure, but this increase in pressure has an insignificant influence on the turbine efficiency.

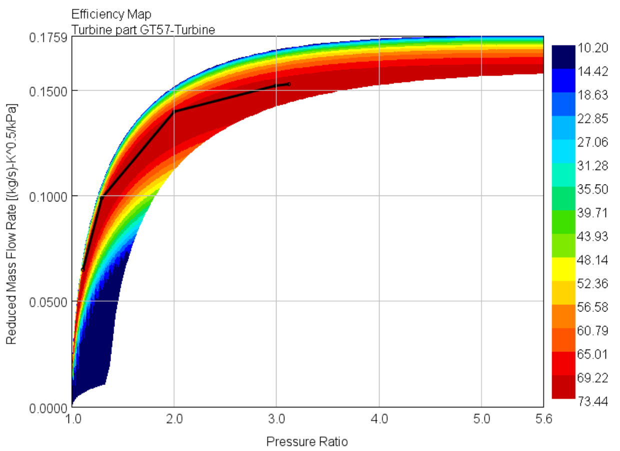


Figure 8. Operating points of the turbine on the turbine isentropic efficiency contour

3.2. Fuel injection rate

To evaluate the amount of fuel injection, the amount of fuel compared to the base condition has been increased by 2, 3, 4, and 5% (Table 2).

Table 5. The technical specifications of the engine

Engine speed (rpm)		600	1000	1400	1700	1800
Injected fuel mass (mg/cycle)	Base	260	296	430	577	580
	+2%	265.2	301.92	438.6	588.54	591.6
	+3%	267.8	304.88	442.9	594.31	597.4
	+4%	270.4	307.84	447.2	600.08	603.2
	+5%	273	310.8	451.5	605.85	609

BMEP is one of the important power characteristics. Equation 7 shows the formula for BMEP calculating.

$$BMEP\ (bar) = \frac{4\pi\ x\ (Torque\ (N.m))}{Displacement\ (L)x100} \qquad (7)$$

As shown in Figure 9, due to the lower operating temperature of the engine at low speeds and loads, by increasing injected fuel mass up to 5%, the BMEP does not rise significantly, indicating that a portion of the fuel did not participate in the combustion process. Therefore, for obtaining the maximum overall efficiency in the engine, the fuel mass should not be increased at low speeds and loads. But at maximum engine load and speed, raising the injected fuel mass by up to 5% increases the BMEP up to 4 bars.

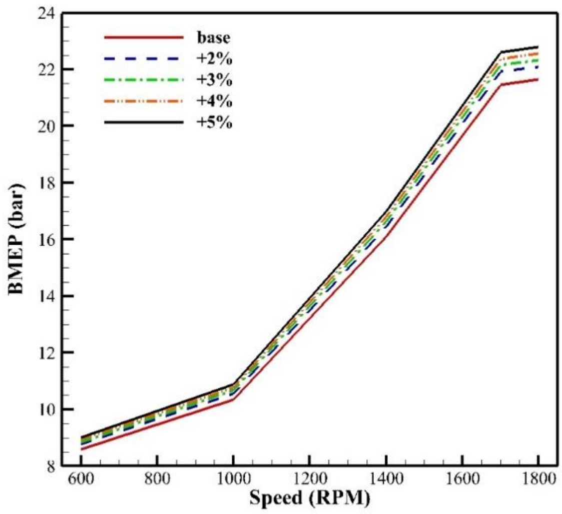


Figure 9. The influence of injected fuel mass on the BMEP

As shown in Figure 10, raising the injected fuel mass by up to 5% raises the engine fuel consumption, but it decreases the BSFC value. This indicates that the engine efficiency increases as the injected fuel mass rises. Because, as it was observed in Figure 9, raising the injected fuel mass increases the output power especially at high speeds. Also, according to Figure 11, at high speeds, with a slight increase in injected fuel mass, the engine torque increases significantly. On the other hand, observations show that at low speeds, increasing the injected fuel mass has almost no effect on the BSFC, because a large portion of the injected fuel exits without participating in the combustion process (especially at engine speeds below 1000 rpm).

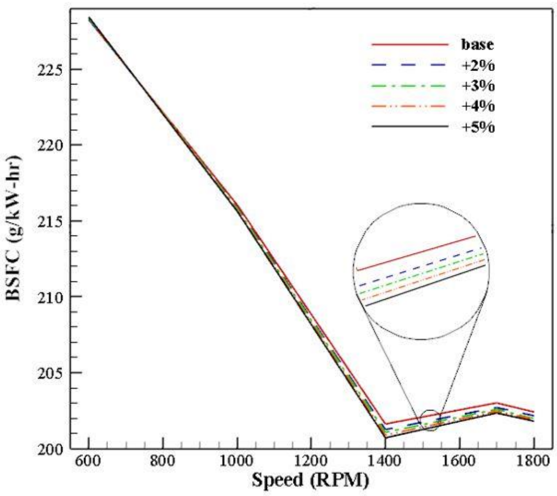


Figure 10. The influence of injected fuel mass on the BSFC

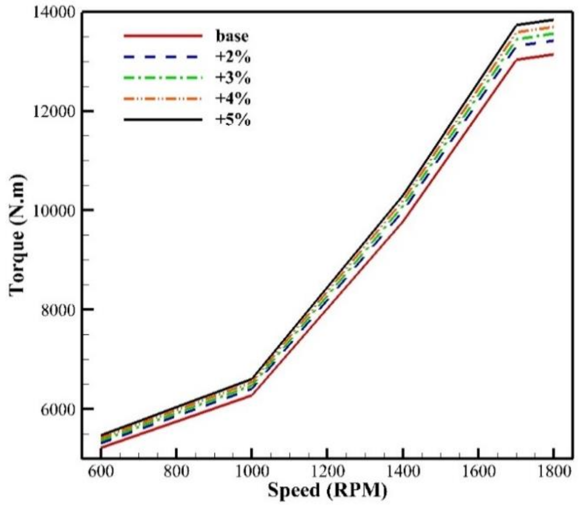


Figure 11. The influence of injected fuel mass on the torque

Figure 12 shows the impacts of raising the injected fuel mass on the output engine power. Increasing the injected fuel mass by 5% at an engine speed of1800 rpm increases the output power by about 7.69%, while there is no considerable change in BSFC.

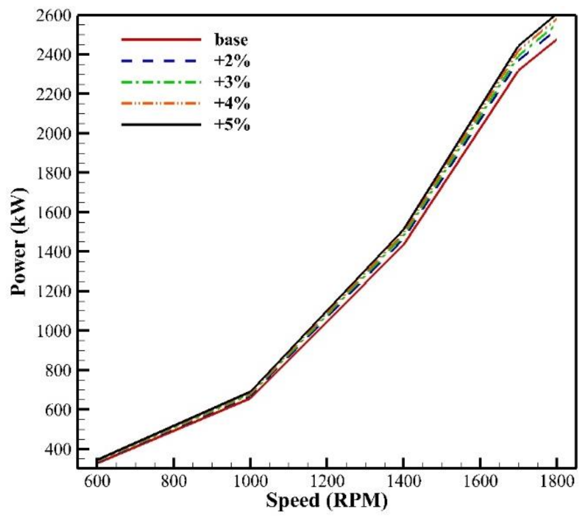


Figure 12. The influence of injected fuel mass on the output power

Figure 13 shows that as the injected fuel mass grows, the air-to-fuel ratio decreases, which is perfectly normal, but it is important to note that the air-to-fuel ratio significantly decreases at low speeds and loads. In this situation, the combustion temperature is much lower than the full load, so as the injected fuel mass increases, a remarkable portion of injected fuel does not participate in the combustion process, and therefore the air-to-fuel ratio decreases sharply. This condition decreases the engine efficiency at low speeds and loads.

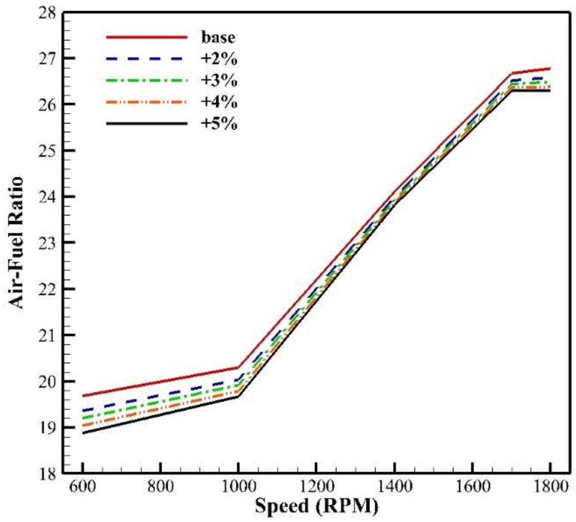


Figure 13. The influence of injected fuel mass on the air-fuel ratio

Figures 14 and 15 present the in-cylinder mean pressure and the exhaust temperature of the engine at 1800 rpm. It is observed that by increasing the injected fuel mass, the maximum in-cylinder mean pressure increases up to 4%, but beyond this point, the maximum in-cylinder pressure does not change considerably. On the other hand, according to Figure 15, increasing the fuel mass up to 5% at 600 rpm increases the turbine inlet temperature by about 10^{O}C, and at 1800 rpm it raises this temperature by about 7 ^{O}C, which is higher than the turbine temperature limitation can damage turbine of turbocharger. Since the efficiency and power output are not much different with the addition of 4 and 5% injected fuel mass, due to the limitation of engine temperature, it is better to increase the fuel mass by about 3% to improve engine power.

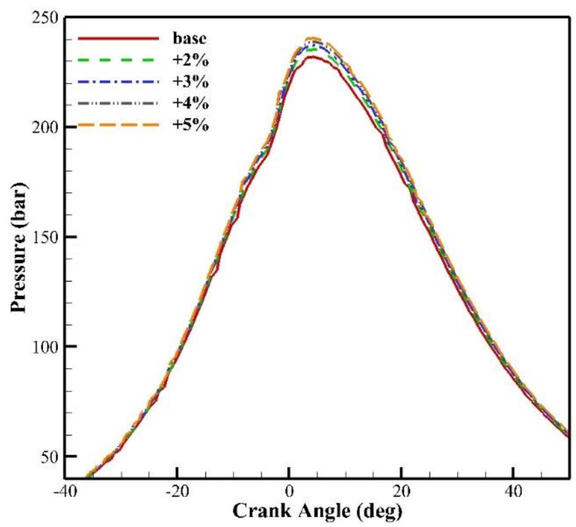


Figure 14. The impacts of injected fuel mass on in-cylinder pressure at 1800 rpm

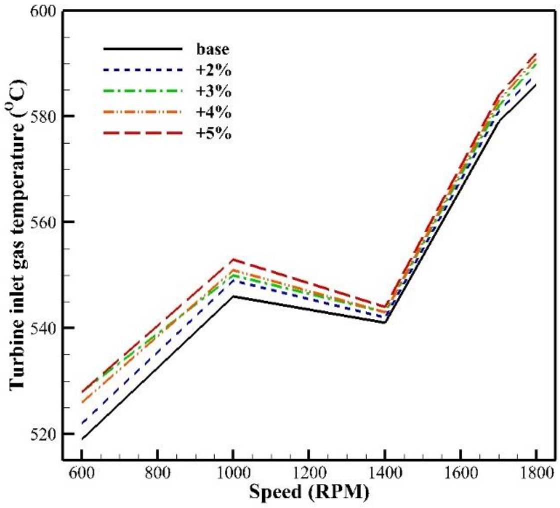


Figure 15. The influence of injected fuel mass on the exhaust temperature

3.3. Compression ratio

To investigate the effect of CR, changes of this parameter in the range of 15 to 18 were evaluated on various engine performance parameters. According to the previous section and the influence of injected fuel mass on different parameters (exhaust temperature, output power, and BSFC), the addition of the injected fuel mass by up to 3% is selected as the best option case, thus in the other power enhancement processes, this level of injected fuel mass is selected.

Figure 16 illustrates the influence of changing CR on the BMEP values. It can be observed that at low speeds and loads (600 rpm) the IMEP level does not change significantly, because in this operating range, the combustion temperature of the engine is very low, and changing the dead volume of the engine does not have a significant effect on the engine power output. On the other hand, at the engine speed of 1400 rpm and above, by the increasing CR, the BMEP value continuously grows. It is important to note that with the addition of 3% in injected fuel mass, the BMEP of the engine was increased compared to the base condition of the engine. This curve also shows that at the medium load range (1000 to 1400 rpm), the BMEP can be increased even with lower levels of rising in the injected fuel mass.

As shown in Figure 17, the most considerable effect of CR is on the engine fuel consumption and changes in the in-cylinder pressure, so that fuel consumption is improved by 3% at best.

A reduction in the BSFC value indicates an increase in engine efficiency, but it is noteworthy that besides the significant increase in efficiency it is necessary to pay close attention to the combustion temperature and exhaust air temperature because there are limits in the operating temperature of the engine based on the ability of the cooling system and engine accessories. Furthermore, at medium loads, in a CR range between 16 and 15, the operating conditions of the base engine improve in terms of efficiency and fuel consumption.

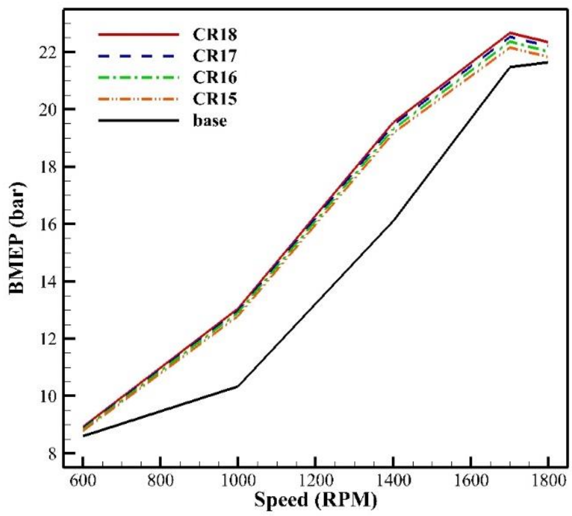


Figure 16. The influence of CR on the BMEP

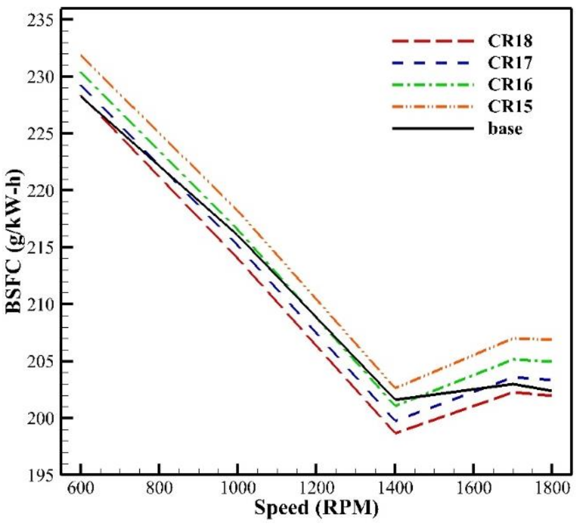


Figure 17. The influence of CR on the BSFC

In Figures 18 and 19, the effect of CR changes on engine power and torque is investigated. As can be observed, reducing the CR decreases the power output and torque significantly at high loads. This power reduction occurs if the injected fuel mass remains constant, so according to Figure 17, reducing the CR increases the fuel consumption considerably.

By decreasing the CR, due to the increase in the dead volume and formation of the combustion process in a larger volume, the maximum in-cylinder pressure decreases. As a result, the maximum temperature inside the combustion chamber and, naturally, the temperature of the exhaust gas decrease. According to Figure 20, by reducing the CR from 18 to 15 for the enhanced engine, the maximum pressure is reduced by about 5 bars. According to Figures 20 and 21, which show the effect of changing the CR on the average pressure and temperature inside the cylinder, this increase in maximum pressure in-cylinder has an adverse effect on the power increase process because increasing the maximum combustion pressure can lead to a significant rise in the exhaust temperature. The results also show that between the CR of 15 to 16, the maximum in-cylinder pressure is observed, which is due to engine performance limitations. Therefore, the best CR for the enhanced engine is this range.

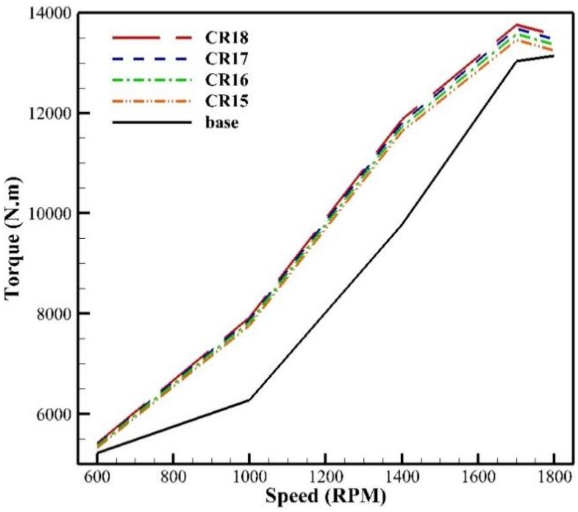


Figure 18. The influence of CR on the torque

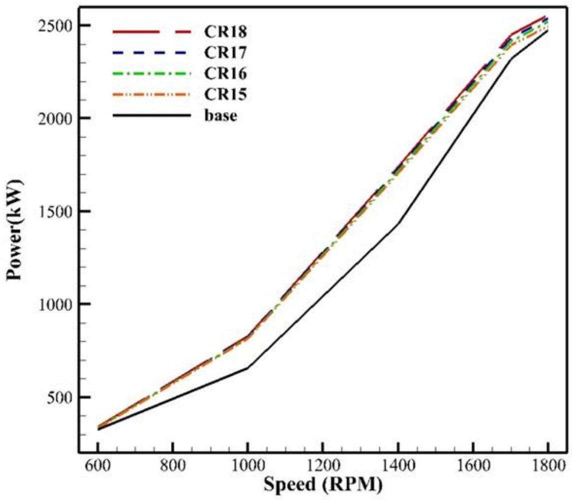


Figure 19 The influence of CR on the output power

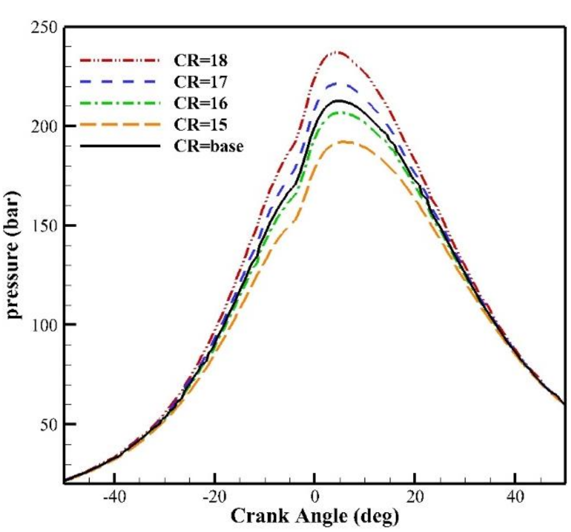


Figure 20. : The influence of CR on the in-cylinder pressure at 1800 rpm

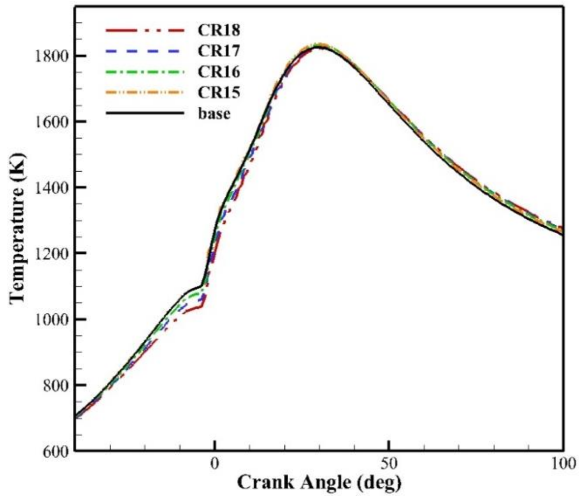
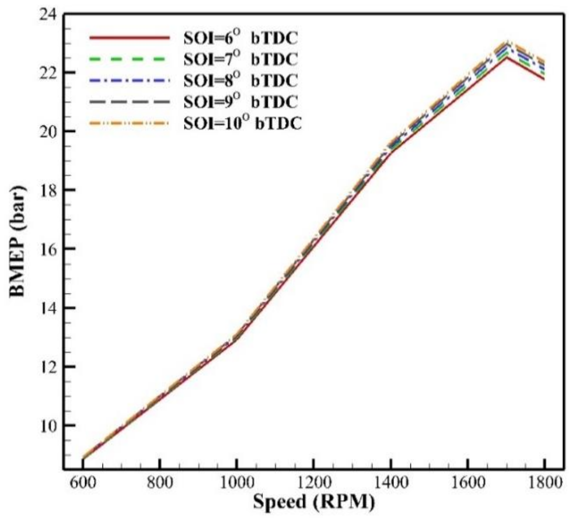


Figure 21. The influence of CR on the in-cylinder temperature at 1800 rpm

3.4. Start of injection timing

To evaluate the effect of the start of injection (SOI) timing, its value was changed from -6 to -10 °CA bTDC.

Figure 22 illustrates the effect of the SOI timing on the BMEP. Again, at low speeds and loads, the SOI timing does not have much effect on engine performance, but at maximum speed, earlier fuel injection up to 4O CA causes the BMEP to grow by about 1 bar. With earlier fuel injection, since the fuel and air have more time to mix, also with earlier fuel injection, the combustion process begins earlier, so the BMEP level rises with earlier fuel injection.



The effect of SOI timing on the BSFC level is presented in Figure 23. The plot shows that with earlier fuel injection from -6 to -10 °CA bTDC BSFC is improved by about 3% at best. The effect of SOI timing on the fuel consumption is higher at higher speeds. Figure 24 shows that the earlier injection of diesel fuel raises the torque. By taking a closer look, it becomes clear that the alternations of these parameters are more

Figure 22. The influence of SOI on the BMEP

insignificant with the SOI timings in the range of 8-°CA bTDC.

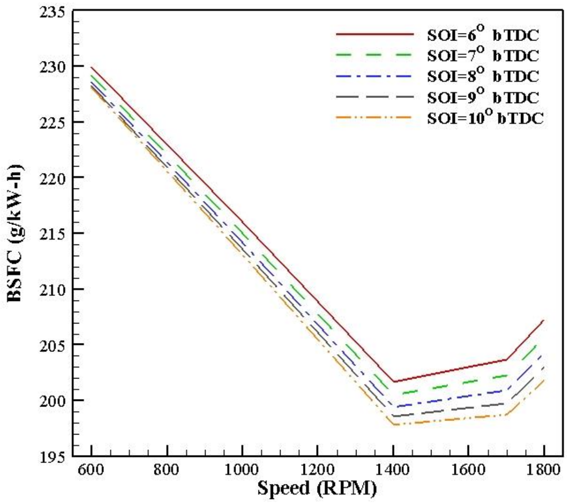


Figure 23. The influence of SOI on the BSFC

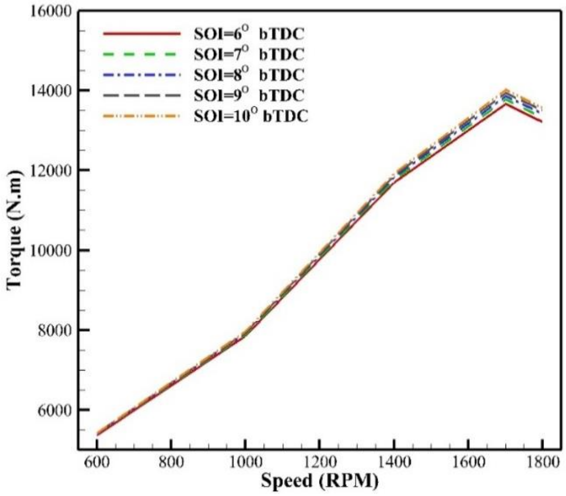


Figure 24. The influence of SOI on the torque

According to Figure 25, at 1800 rpm, with earlier fuel injection, the maximum mean pressure increases by about 10%. Earlier injection of fuel into the combustion chamber results in the formation of a more homogeneous fuel-air mixture, which results in the participation of a higher portion of fuel mass in the combustion process [21].

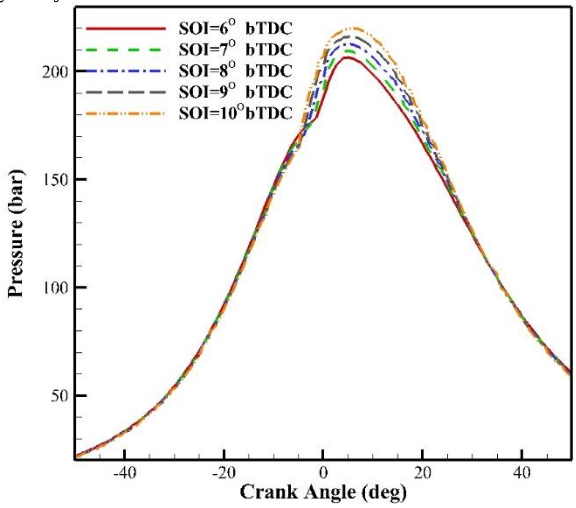


Figure 25. The influence of SOI on the in-cylinder pressure at 1800 rpm

Another important parameter that is affected by the SOI is the exhaust gas temperature. With earlier fuel injection, the temperature of the exhaust gases decreases due to the earlier heat release. The greatest temperature drop of about 20 degrees occurs at low speeds (Figure 26).

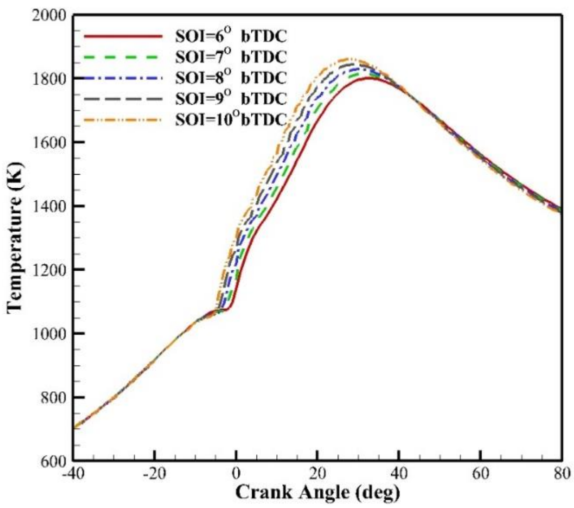


Figure 26. The influence of SOI on the in-cylinder temperature at 1800 rpm

Figure 27 examines the effects of fuel injection timing at different crank angles on the rate of heat release (RoHR) for the enhanced engine. This figure also shows the RoHR diagram for the base rail engine for a better comparison. As well As according to Figure 28 observed that, by changing the CR compared to the base state, the start of combustion gets delayed, in this case, since sudden changes occur in the in-cylinder pressure of the cylinder due to the combustion of fuel in a larger volume of the combustion chamber, the maximum efficiency cannot be obtained. By earlier fuel injection, it ignites at a greater distance from the TDC. This causes a decrease in power and increases the risk of engine knock. As a result, it is necessary to change the SOI timing to set the start of the combustion time. As can be seen in the figure, the SOI for the base engine at 1800 rpm is 6 °CA bTDC. Using this SOI timing for the enhanced engine with a CR of 16 and a 3% increase in injected fuel mass delays the combustion time by about 3 degrees relative to the base engine. By changing the SOI timing it can be observed the best point for fuel injection is at 8 °CA bTDC. In this case, due to the addition of injected fuel mass the maximum heat release rate increases compared to the base engine, on the other hand, according to the red arrow in Figure 28 the combustion duration is the same as the base engine and in the optimal state and the start of combustion is not significantly different compared to the base engine.

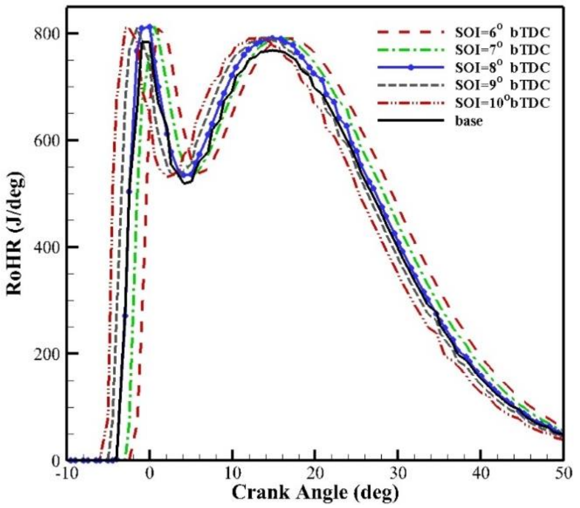


Figure 27. The influence of SOI on the RoHR

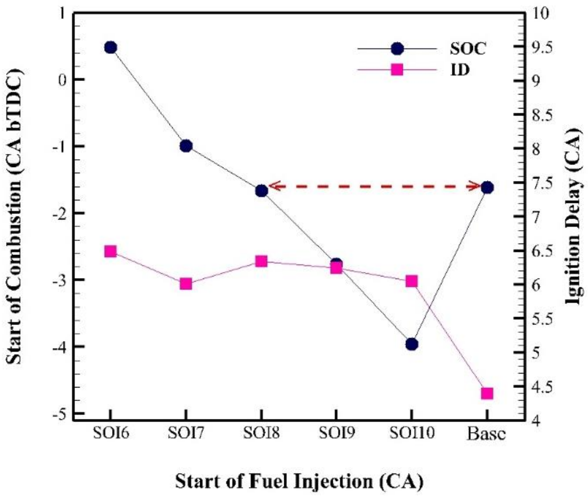


Figure 28. The influence of SOI on ID & SOC

4. Conclusion

In the present study, to investigate the feasibility of increasing the power density of the MTU4000 R43L heavy-duty diesel engine by considering the existing performance and temperature limitations, first, a simulation of the engine, turbocharger, and intercooler was performed the GT SUITE software. The results of power, BSFC, and BMEP of the simulation results were compared with the experimental data, and the results were validated. Finally, the influence of increasing injected fuel mass, decreasing CR, and changing SOI in the simulated model was examined to achieve the ideal conditions. The main findings of the study are as follows:

- To check the influence of injected fuel mass, the amount of fuel was increased by 2, 3, 4, and 5% compared to the base state.
- At 1800 rpm, increasing the injected fuel mass by 5% compared to the base state increased the power by about 7.6%, while the BSFC was significantly changed.
- To control the exhaust temperature, the influence of reducing the CR from 18 to 15 was investigated. The greatest effect of CR was on fuel consumption of the engine and the in-cylinder pressure, so that fuel consumption was improved by 3% at best, but, in return, the maximum pressure was increased by about 20%
- To evaluate the effect of SOI timing, the SOI values were changed in the range between -10 to -6 °CA bTDC.
- The effect of SOI timing on various engine performance parameters was examined. With earlier fuel injection, the BSFC was improved by about 3% at best. The influence of injection timing on fuel consumption is greater at higher speeds.
- According to the studies, upgrading the engine power is possible only if the parameters of compression ratio and the start of fuel injection timing are changed after the fuel mass increasing. Otherwise, the increase in fuel mass may cause damage to the engine, engine accessories, and it causes Inappropriate engine performance.

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