JOURNAL OF MECHANICAL ENGINEERING AND SCIENCES (JMES)
ISSN: 2289-4659 e-ISSN: 2231-8380
VOL. 15, ISSUE 2, 8153 – 8168
DOI: https://doi.org/10.15282/jmes.15.2.2021.15.0640

ORIGINAL ARTICLE

Improving the performance of a diesel engine by changing injectors characteristics after reduction on the compression ratio

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ABSTRACT – In order to repair internal combustion engines, sometimes it is necessary to replace the components of these engines with each other. Therefore changes in engine performance are inevitable in these conditions. In the present study, by changing the conecting rod and the crank of the OM457 turbo diesel-fueled engine with the OM444, it was observed that the performance of the engine decreases. Numerical simulations have been carried out to study the Possible ways to mitigate this reduction. One way to achieve this goal is to change the fuel injector's characteristics such as, fuel injector's nozzle hole diameter, number of nozzle holes, and start time of fuel injection. In this study, the impact of these parameters on the performance and emissions of these engines were analyzed. Another scenario is an increase in inlet fuel and air by the same amount. The results indicate that By reducing the diameter of fuel injector holes and hole numbers, the performance of the engine was increased. on the other hand, the NO_x emissions were increased while the amount of soot emission decreased. The same results were concluded by retarding the start time of injection. Subsequently, a case study of changing fuel injector parameters for mitigation of decreased performance was performed. These parameters were simultaneously applied, and results were compared. The performance of the engine with improved injector's characteristics was close to the main OM457. Similar results were obtained by increasing the amount of inlet air and fuel.

ARTICLE HISTORY
Received: 04th May 2020
Revised: 03rd Oct 2020
Accepted:15th Dec 2020

KEYWORDS
Diesel fuel;
CI engine;
injector characteristics;
performance and emission;
compression ratio

INTRODUCTION

The ability to deliver constant power and economically safe made the Internal Combustion Engines (ICE) suitable for stationary and movable applications. Therefore, these types of engines have been mass-produced in many different types and classes. one of these types of ICE are compression ignition engines, which commonly work with diesel fuels [1, 2].

Recently, legislation has limited the emissions of ICE including heavy-duty diesel engines and studying new strategies is inevitable to reduce the emissions of these engines [3,4]. For reducing the emissions of a diesel engine, the combustion temperature should be decreased [5]. For this purpose, some solutions have been presented, such as alternative strategies in fuel injection and combustion phase or using different fuel-additives [6,7,8]. One of the ways to reduce the combustion temperature and emissions of diesel engines, especially NOx is to reduce the compression ratio (CR) of an engine, which will cause a reduction of in-cylinder temperature, and in return, the NOx emissions will be reduced [9,10,11].

On the other hand, according to Cooney et al. [11] and Yucesu et al. [12], the performance of a specific engine will reduce by lowering the CR due to the reduction of its pressure. To correct this reduction in the performance of an engine, several strategies can be considered. According to Shundoh et al. [13], lowering the diameter of the nuzzle will cause a more homogeneous mixture of air and fuel. Therefore the heat release and the in-cylinder pressure will be increased. In-cylinder pressure has a direct influence on the output power of a cylinder. By increasing the pressure in an engine, the output power and performance of the engine will be increased too [1]. Noguchi et al. [14] reported that by changing the start time of fuel injection in a diesel-fueled engine, the in-cylinder pressure will change. Based on their study, retarding the start time of injection from 200 crank angle before Top Dead Center (TDC), the pressure of the chamber will drop. Although this drop in pressure will reduce the knock, it will reduce the performance of the engine too. Based on Montgomery et al. [15], the impact of geometry and hole numbers of a fuel injector is inevitable on the performance parameters of a heavy-duty diesel-fueled engine. The following results are concluded by their study: as the penetration decrease, fuel droplets' diameter will be reduced, the impact of penetration on fuel consumption is more than droplets' size, and by decreasing the hole numbers, Specific Fuel Consumption (SFC) will be reduced.

According to the mentioned studies, changing the fuel spray characteristics will either increase or decrease the performance and emissions of an ICE. The aim of this study is to study the effect of changing the fuel spray characteristics of an engine with lower CR and the possibility as reaching the same performance of a high-CR engine. Subsequently, in order to improve the engine performance, these changes are applied to the engine simultaneously. For this purpose, all of

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the cases which has been shown in Figure 1 were modeled, and their performance and emissions such as NOx and SOOT were compared. The characteristics of these engines are illustrated in Table 1.

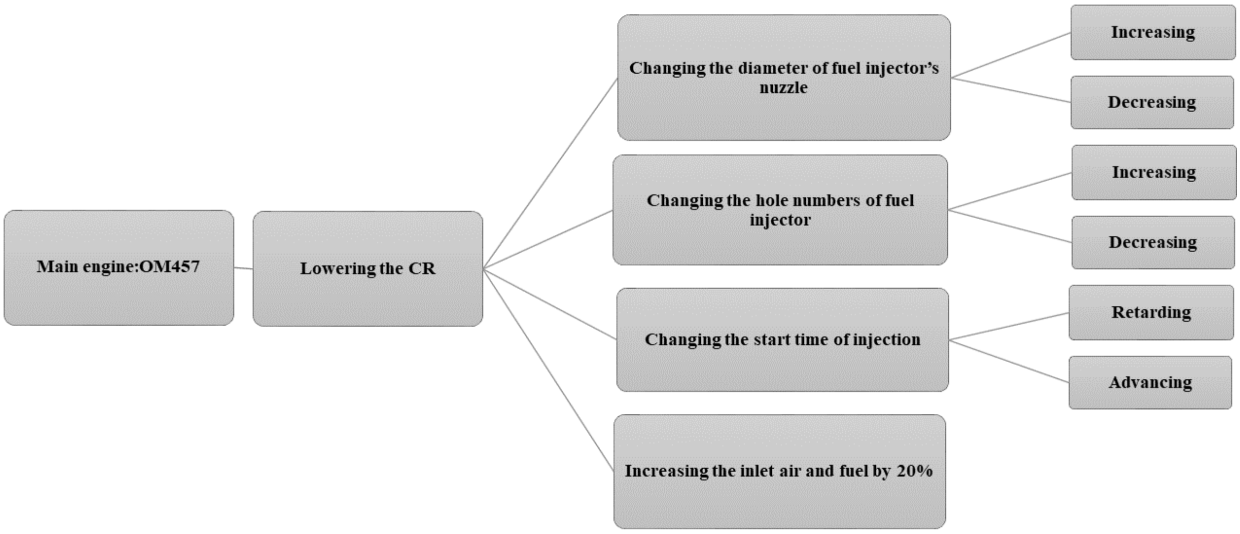


Figure 1. Modeling approach of the present study

Table 1. OM457's general characteristics [16]

No. of cylinder	Displacement (Liter)	Compression Ratio	Bore (mm)	Stroke (mm)	Connecting rod length (mm)	Rated power @2000 RPM (kW)
6	11.97	17.75:1	128	155	250.9	260

NUMERICAL PROCEDURE

For performing the computational modeling of the present study, CONVERGE CFD software [17] is employed. For modeling procedure, some sub-models have been used. These sub-models are briefly described below.

Spray Model

For spray modeling, which is built on the Eulerian-Fluid and Lagrangian-Drop (LDEF) framework, is divided into the physical sub-models such as break-up droplet collision, wall interaction, and vaporization. For the early fuel jets just after the nozzle's exit, we assumed the diameter of fuel droplets to be equal to the diameter of the nozzle holes. For modeling the fuel jets breakup in this study, Kelvin-Helmholtz and Rayleigh-Taylor (KH-RT) models were used. This model separates droplets break up into two steps. By using the Kelvin-Helmholtz instability for the first step, described by Reitz and Bracco [18], the fuel spray break up will be predicted, and the Rayleigh-Taylor model is used for predicting the secondary break up of the droplets [19, 20]. The interaction of liquid drops with solid surfaces such as the cylinder wall was modeled by O'Roruke model [21]. For modeling the collision in this study, the No Time Counter method (NTC) has been used. Schmidt and Rutland [22] showed that this method is very similar to the gas dynamic's methods for Direct Simulation Monte Carlo (DSMC) calculations. For modeling the convertion of liquid droplets to gaseous vapor, Frossling correlation [23] was used. This correlation determines the time rate of change of droplet size.

Combustion Modeling

After the break up of the fuel jet and mixing with air, an acceptable model should be used for combustion modeling. In the present study, SAGE detailed chemical kinetics solver [24] has been used for this purpose. This solver uses elementary reactions and calculates the reaction rates for each primary reaction. As a result of this detailed approach for combustion modeling, the computation time will be higher. However, the result will be much closer to the experimental, and it can be used in many combustion regimes in different conditions. This chemical kinetics solver was applied and integrated with a multi-zone model [25] to decrease the time of simulations.

Emission Modeling

NOx model

Nitrogen oxides are one of the secondary products in the diesel fuel high-temperature combustion. This high temperature will cause the N2 to break down, and after that, synthesizing with oxygen will form one of the most threatening emissions in ICE combustion, NOx.

In the present study, because of the high temperature, the rapid transient formation of NO_X (prompt NO_X) can not be used [26], therefor the extended Zel'dovich mechanism has been used. This mechanism was presented by Heywood [1] and was given by three different reactions with different rates, which can calculate NOx by assuming an equilibrium reaction.

SOOT Modeling

The soot mass production in a cell can be determined by a competition among mass formation and mass oxidation rate of the soot based on the Hiroyasu and Kadota [27] , which is known as the Hiroyasu model. In the present study, by assuming the soot particles as uniform in size and spherical, Nagle and Strickland-Constable (NSC) [28] model were used to simulate soot oxidation, and after coupling it with the Hiroyasu model, the rate of soot formation was calculated. For computing production of soot mass in a cell, a single-step competition between the soot mass oxidation rate and the soot mass formation rate was used. This competition is formulated by Hiroyasu and Kadota [27] as Eq. (1) :

$$\frac{dM_s}{dt} = \dot{M}_{sf} - \dot{M}_{so} \tag{1}$$

The formation rate is calculated by:

$$\dot{M}_{sf} = SF\ M_{form} \tag{2}$$

and,

$$SF = A_{sf} P^{0.5} \exp(-\frac{E_{sf}}{R_u T}) \tag{3}$$

While M_s (g) is the production of soot mass, $\dot{M}_{sf}$ (g/s) is the soot mass formation rate, $\dot{M}_{so}$(g/s) is the soot mass oxidation rate, M_{form} (g) is the mass of the soot formation species, P(bar) and T (K) is the cell pressure and cell temperature, R_u *(cal/(K g mol))* is the universal gas constant, E_{sf} *(cal/g mol)* is the activation energy, and A_{sf} *(1/(s bar$^{0.5}$))* is the Arrhenius preexponential factor [29].

Turbulence Modeling

For solving Navier-Stokes equations in a turbulence field, many models are suggested. Reynolds-Averaged Navier-Stokes (RANS) is one of them. Based on Han and Reitz [30] and Krishna et al. studies [31] RNG k-ε model was used in this study due to its reasonable accommodation with the experimental data. This model uses a statistical technique known as the renormalization group, which leads to a variable turbulence Prandtl number. As a result of this modification, more accurate results can be extracted compared to the other RANS turbulence models [32][32].

Case Setup and Validation

Case setup

The real piston of OM457 is shown in Figure 2. After some measurements, the geometry of this piston, which is sketched by some CAD software, is illustrated in Figure 2(b). After that, the fluid volume captured in the cylinder was extracted for CFD modeling as Figure 2(c).

After importing the geometry to the CFD software by using the data from Table 1 and Table 2, the case setup has been made.

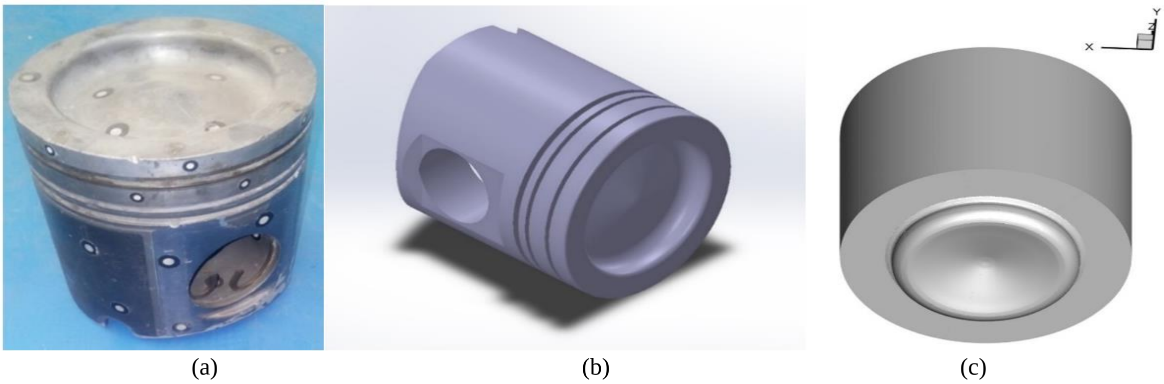


Figure 2. The geometry of piston head used for modeling: (a) actual model, (b) CAD model, (c) fluid volume extracted for CFD modeling

Table 2. Valves and injector characteristics of OM457

IVO/IVC (deg)	EVO/EVC (deg)	Start of injection (deg)	Numbers of fuel injecture's holes	Diameter of fuel injecture's holes (mm)	$\dot{m}$ of fuel (gr/cycle)
336/-144	116/387	-13.7 TDC	7	0.259	0.152

Boundary Conditions

For modeling a study case in a CFD software, the boundary conditions should be defined by the user.

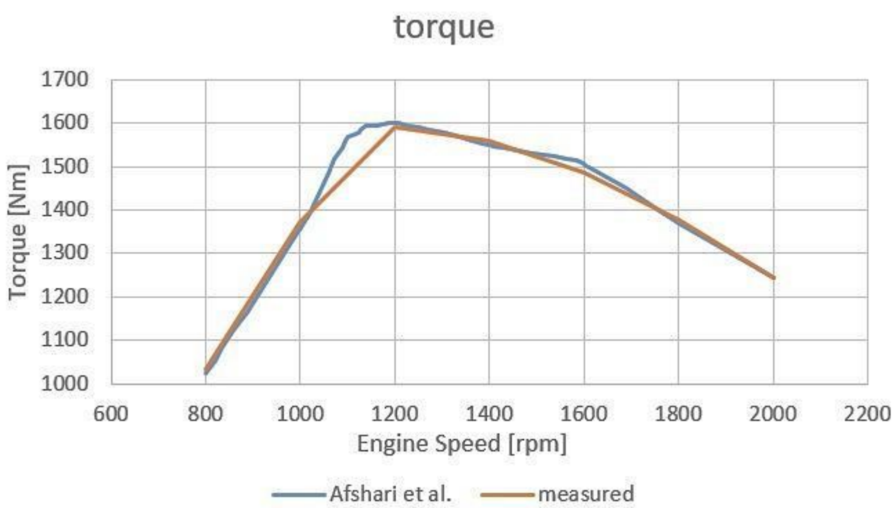
As in Afshari et al. [33] study which they used the same OM457 turbo diesel-fueled engine, and the technical data of OM457[16] the boundary condition are as in Table 3.

Table 3. boundary condition for modeling the OM457

Cylinder wall's temperature (K)	Cylinder head's temperature (K)	Piston's temperature (K)	Inlet air's temperature (K)	Inlet air's pressure (Pa)
433	523	553	336	267130.0

Validation

After setting up the case, a series of CFD modeling was done to check the accuracy of the present study's case setup. In the study by Afshari et al. [33] the torque and the amount of NOx had been reported and compared with experimental tests for various engine speeds, and their results were matched with experimental data. Therefor for validation porpuses, the torque and amount of NOx emission were calculated in different engine speeds (800-1000-1200-1400-1600-1800 and 2000 rpm). The results were compared with those of study as mentioned earlier shown in Figure 3.



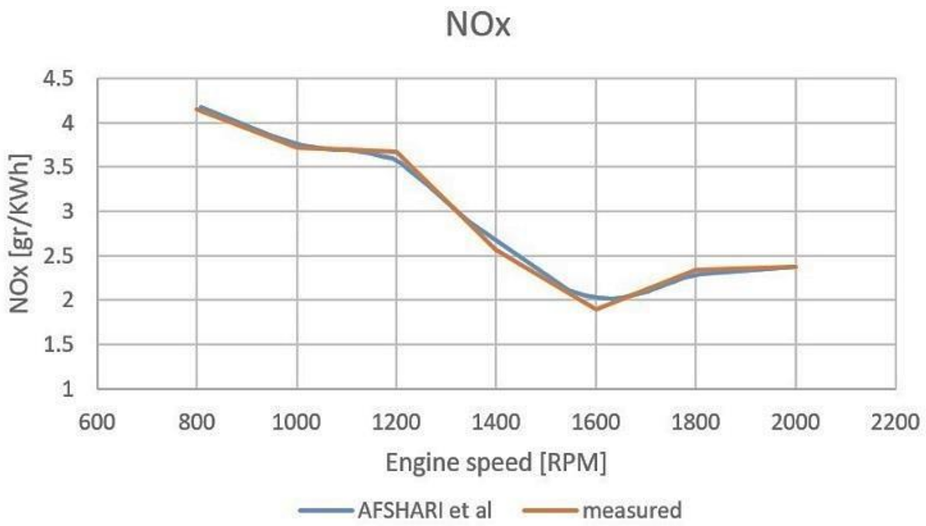


Figure 3. Comparison of Afshari et al. [33] data with the present study's modeling for different engine speeds: (a) NOx, and (b) torque

The results of this modeling are in reasonable accommodation with the previous study in those engine speeds. The maximum error was in the engine speed of 1600 rpm, in which the difference was less than 2% for torque and NOx emission.

On the other hand, the results of this CFD modeling in the engine speed of 2000 rpm were so near to those presented in the OM457 technical data [34] such as IMEP, which the error was less than 0.1 % (13.03 to 13.04 bar).

RESULTS AND DISCUSSION

Lowering the Compression Ratio

After surveying the validation of this modeling approach, the compression ratio of OM457 has been decreased. This reduction is because of changing the connecting rod and crankshaft with the ones from OM444 presented in Table 4 (this change has been done due to the maintenance of the engine).

Table 4. OM457 and OM444's Connecting rod length and crank radius

Engine	Connecting Rod Length	Crank Radius
OM457	250.906 mm	77.36 mm
OM444	256.21 mm	70.80 mm

After these changes, the CR can be measured by simple equations given by Heywood [1]. The new CR is equal to 14.52:1, which equals to 18.2% reduction in CR. Based on the Malaquias et al. [35] this CR for a diesel engine is practically sufficient for combustion. In Figure 4, the difference between these two engines is shown in the top dead center (TDC).

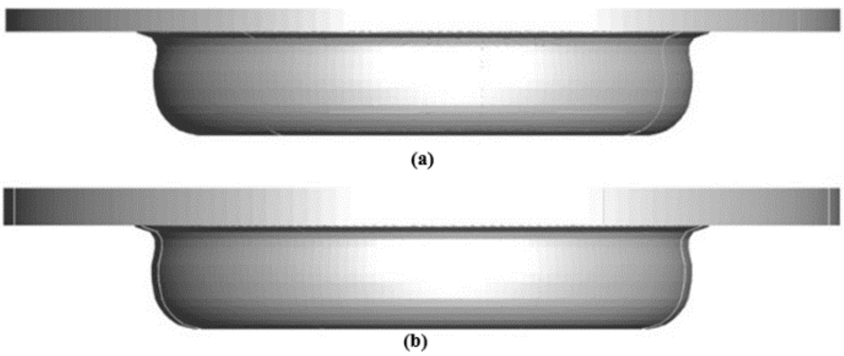


Figure 4. Clearance volume: (a) main OM457 and (b) new engine with lower CR

As it is apparent, the new engine with lower CR has more clearance volume. This increase in the clearance volume will reduce the total pressure of the cylinder shown in Figure 5 at TDC. As in Figure 5, by reducing the CR, the temperature will grow due to the flame's higher speed and acceleration in the start time of ignition (Figure 5).

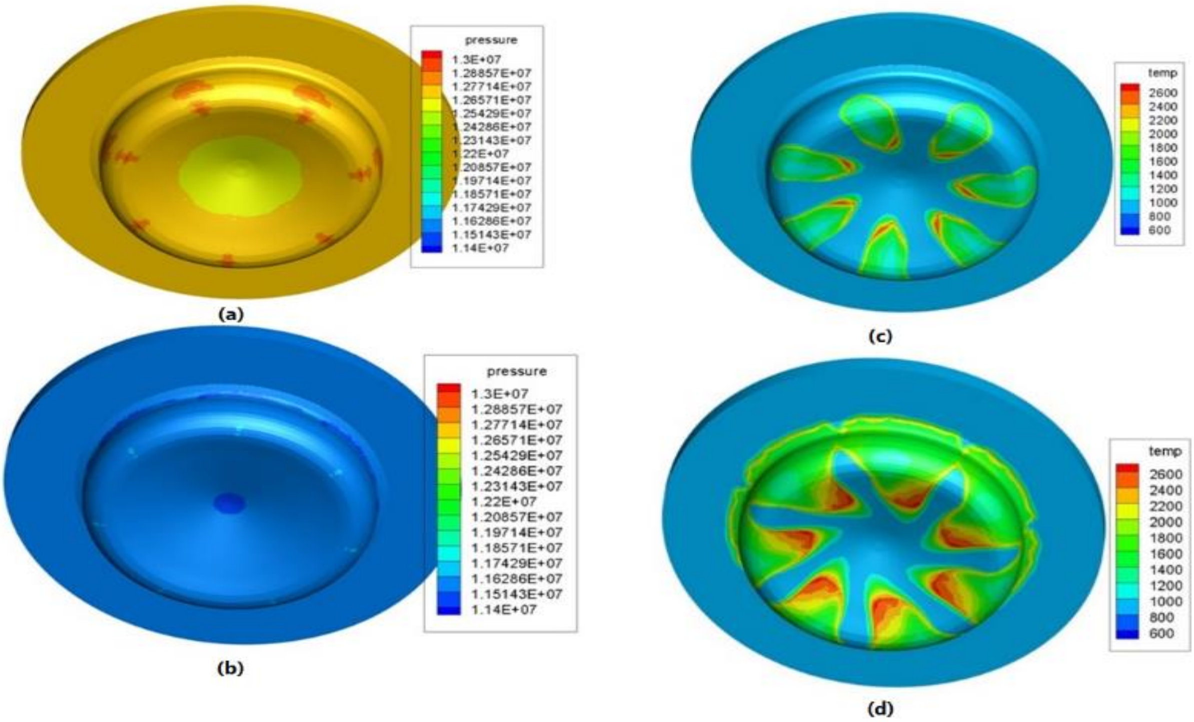


Figure 5. Contours of pressure and temperature for fluid volume at TDC: (a) contour of pressure for OM457, (b) contour of pressure for the new engine with lower CR, (c) contour of temperature for OM457 and (d) contour of temperature for the new engine with lower CR

The performance and emission of these two engines have been illustrated in Figure 6 and the pressure, NO_x, and Soot emission of these two engines is compared.

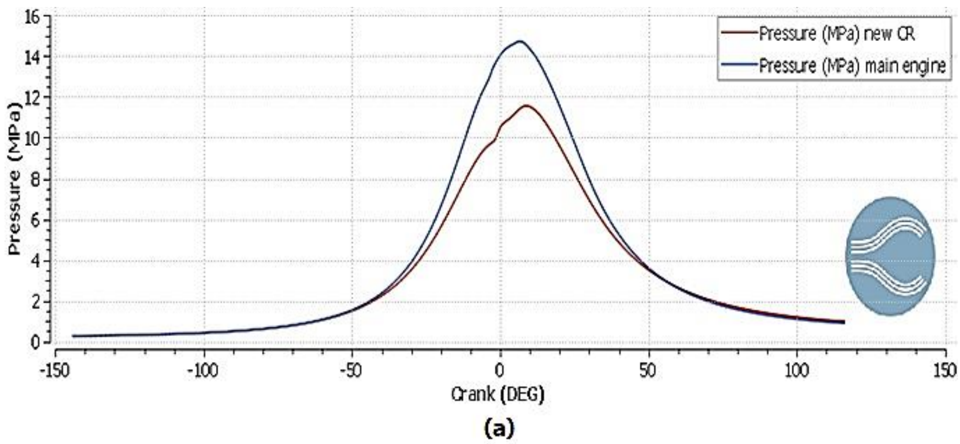
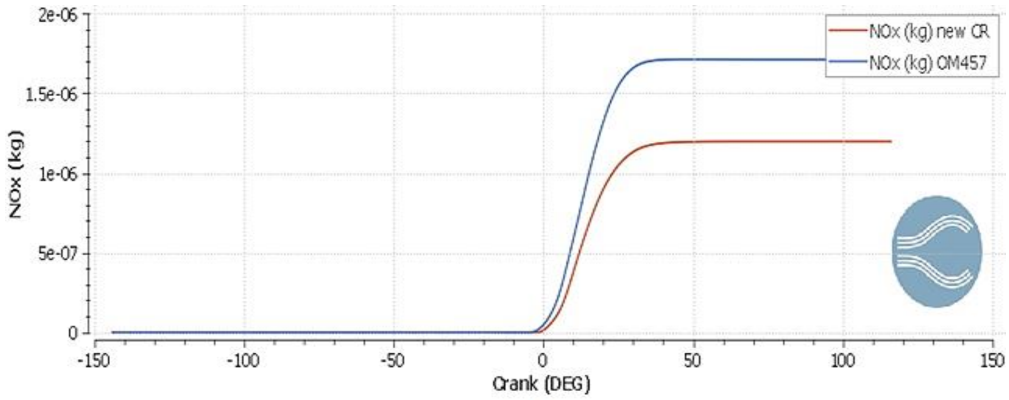
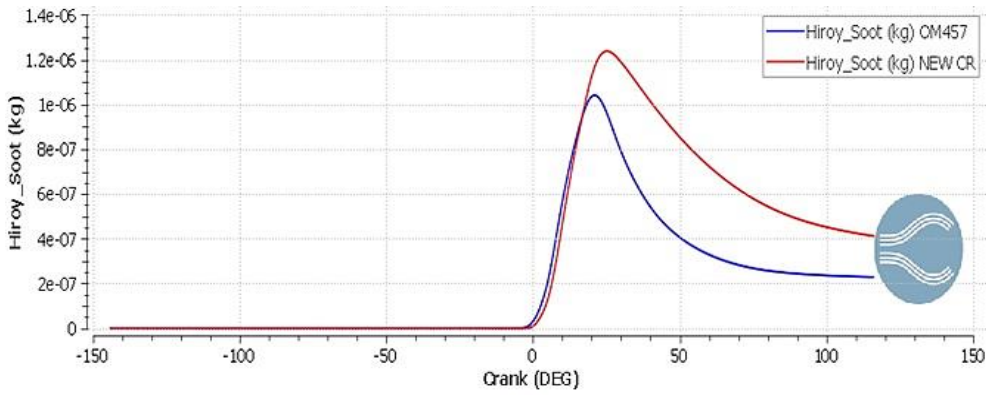


Figure 6. Comparison of the OM457 with the new engine with lower CR at different crank angle, (a) pressure, (b) NOx, and (c) Soot



(b)



(c)

Figure 6. Comparison of the OM457 with the new engine with lower CR at different crank angle, (a) pressure, (b) NOx, and (c) Soot (cont.)

As it is evident in graphs of Figure 6, by lowering the CR, the pressure will drop, and the NOx emission will be reduced by 30.24%, but the Soot emission will increase. After calculating the net work by p-θ diagram, the torque and output power of the engine can be calculated by Eqs. (4) and (5).

$$P_i(W) = \frac{W_{c,i}(J)N\left({}^{rev}/_{S}\right)}{n_R} \tag{4}$$

$$P(kW) = 2\pi N\left({}^{rev}/_{S}\right)T(N.m) * 10^{-3} \tag{5}$$

Where P_i is the indicated power and $W_{c,i}$ is indicated work, N is the engine speed, which in this study equals to 33.333 rev/s and n_R for a four-stroke engine equals 2.

Using the above equations, the performance for each cylinder of these two engines will be calculated, which shows a reduction in torque and power of each cylinder by 13.8%. Results of this calculation are shown in Table 5:

Table 5. Performance comparison of OM457 and the new engine with lower CR

	OM457	New Engine
MEP (Pa)	$1.304\times10^{+6}$	$1.228\times10^{+6}$
Net work (J)	2597.03	2238.41
Power (kW)	43.283	37.303
Torque(N.m)	206.67	178.115

Changing the Diameter of Nozzle Holes

The impact of changing the diameter of fuel injector's nozzle holes is intense as it directly impacts on the depth of penetration or the shape and behavior of the fuel jet [23,24]. However, the impact of this parameter on the performance and emission is not the same in different studies. Thus in this study, surveying the impact of this parameter on the new engine with lower CR is inevitable.

The diameter of the nozzle holes in the leading case was 0.259 mm. After changing the diameter by $\pm 0.02mm$ (which led us to the new diameters equal to 0.239 and 0.279 mm), some runs were done to check the differences on the performance and emission. The results are shown in the graphs of Figure 7.

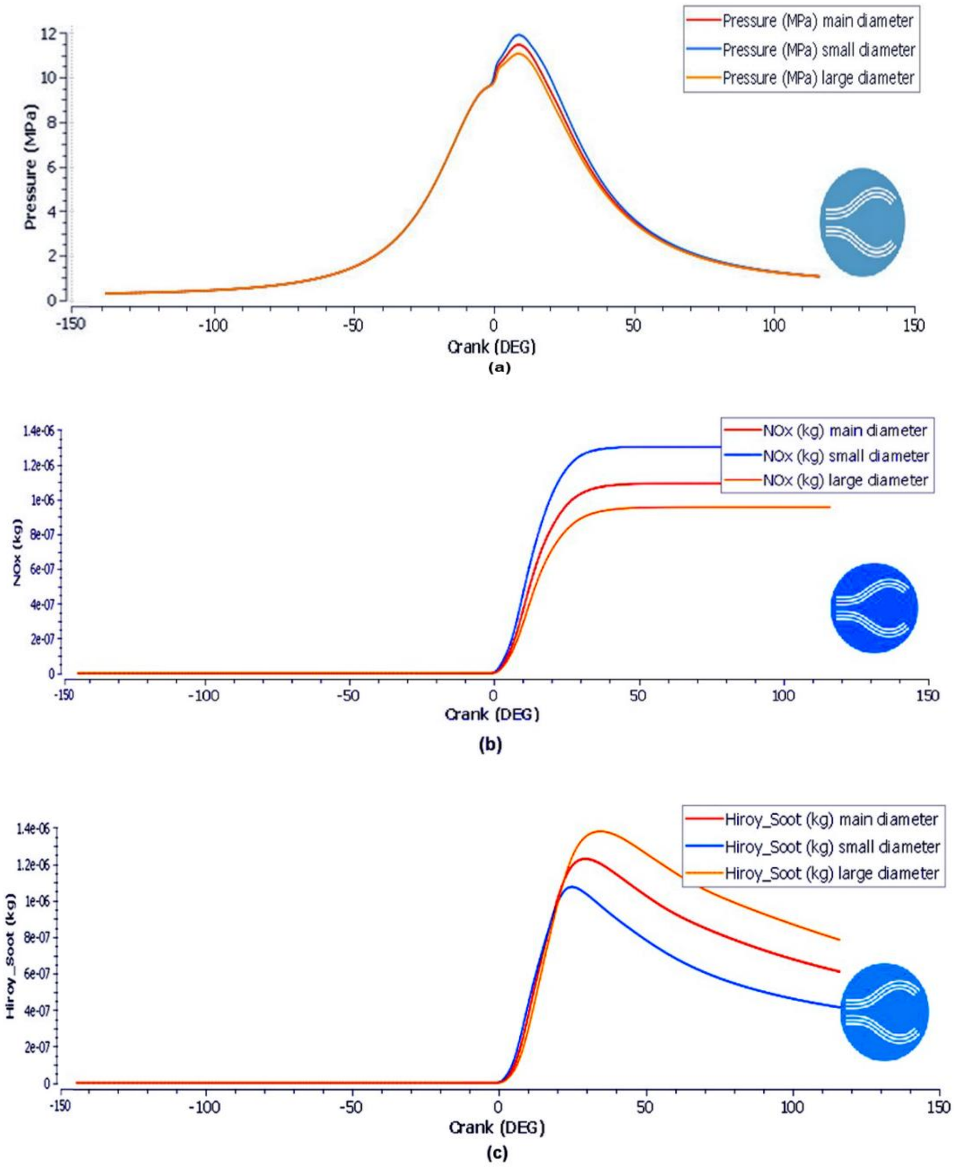


Figure 7. Comparison of the new engine with lower CR by different nozzle diameters, at different crank angle, (a) pressure, (b) NOx, and (c) Soot

As is evident in Figure 7, by decreasing the diameter, the pressure will be increased slightly, and the NOx will increase by 15% (NOx changes by 14.5% when increasing the diameter), but the soot emission will be reduced. As the fuel rate is constant, the pressure of injection will be increased by decreasing the diameter, so these changes are mainly because of the increase in the penetration tip of the fuel jet As shown in Figure 8 at four crank angle before TDC, which is in a complete agreement with Pontoppidan et al. [38]. This higher tip penetration, as Montgomery et al. [15] had shown, will cause a more homogeneous mixture of fuel and air. By this better mixture, the soot will reduce. As the whole domain will ignite properly, the temperature of the in-cylinder will increase, which leads to an increase of NOx emission.

In Table 6, the performance characteristics of the new engine with changing the diameter of the nozzle holes are calculated.

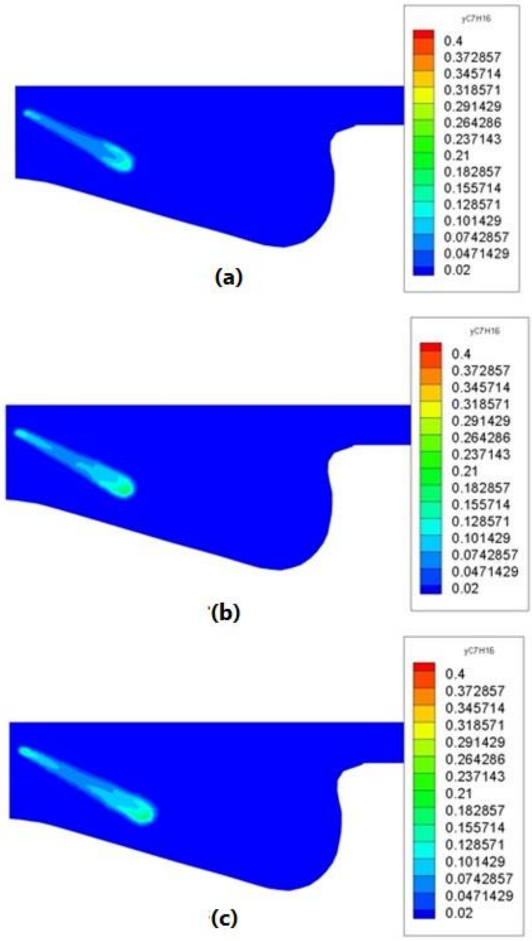


Figure 8. Contours of C_7H_{16} molar fraction at -4 crank angle before TDC: (a) large diameter, (b) main diameter, (c) small diameter

Table 6. Performance comparison of the new engine with lower CR by the different nozzle diameter

	Small Diameter	New Engine	Larger Diameter
MEP (Pa)	$1.277\times10^{+6}$	$1.228\times10^{+6}$	$1.16\times10^{+6}$
Net work (J)	2327.26	2238.41	2113.83
Power (kW)	38.75	37.303	35.2
Torque(N.m)	185.11	178.115	168.075

As in Table 5, by reducing the diameter of the nozzle holes by 0.02 mm, the torque and power will be increased by 4%. On the other hand, for the diameter increase of this magnitude, the torque and power will decrease by 5.5%.

Changing the Numbers of Nozzle Holes

Based on Montgomery et al. [37] and Moon et al. [39] by lowering the nozzle holes numbers in a constant fuel mass rate, the diameter of fuel jet in the chamber will be reduced. Thus, the fuel jet will atomize and vaporize faster. In the present study, the main engine had seven holes in each nozzle, so for surveying this parameter, the hole numbers were changed to 6 and 8, while the fuel rate remained constant. After doing some runs, the fuel jets came from 6 nozzle holes were faster, and they got mixed with the air in the chamber quicker than the other. As a result, the start time of ignition was accelerated in this fuel jet. As shown in Figure 9 at TDC, by reducing the hole numbers, the tip penetration and penetration velocity will increase, which is incomplete accommodation with previous studies [38, 40].

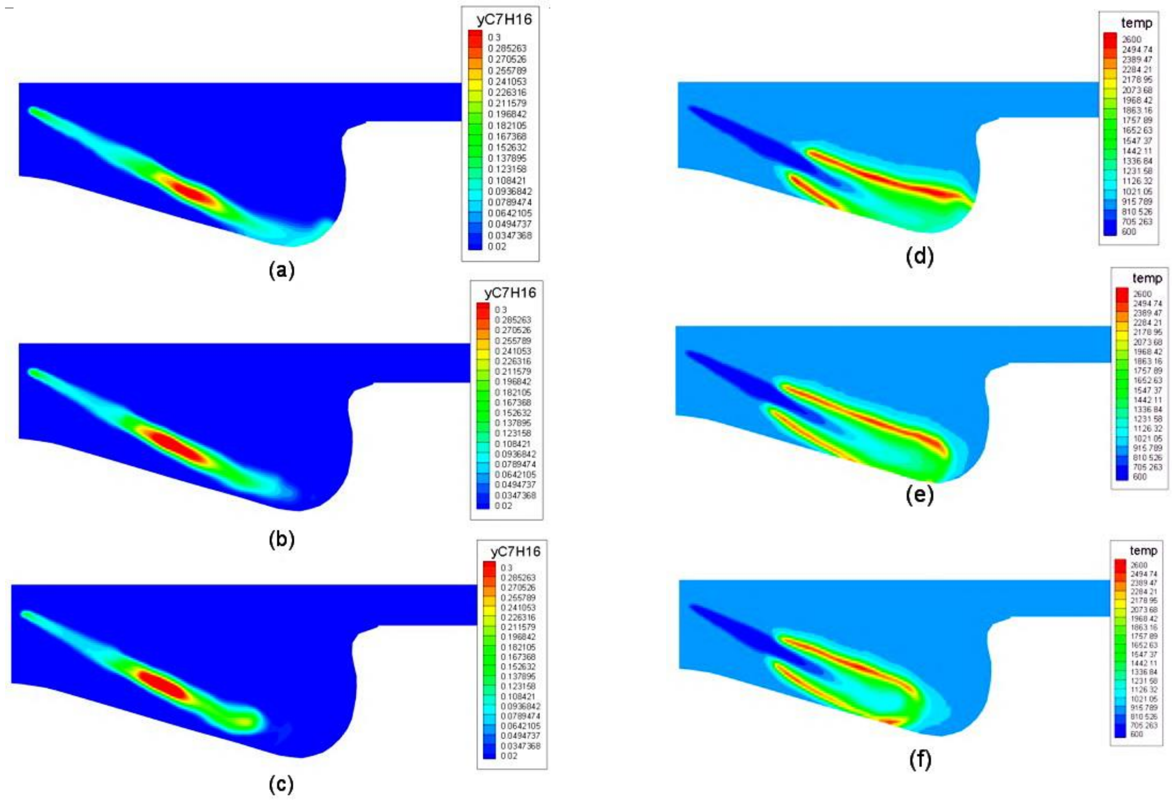


Figure 9. Contours of C_7H_{16} molar fraction and temperature at TDC for the new engine with different nozzle hole numbers: (a) C_7H_{16} molar fraction with 6 holes, (b) C_7H_{16} molar fraction with 7 holes (primitive engine), (c) C_7H_{16} molar fraction with 8 holes, (d) temperature with 6 holes, (e) temperature with 7 holes (primitive engine), (f) temperature with 8 holes

Figure 10 illustrates the comparison of pressure - NOx and Soot due to the changing of this parameter. As in Figure 10, if the number of nozzle holes reduces, the pressure of the chamber will increase, and the NOx emission will increase by 5.9 %. However the soot emission will drop down by a considerable amount. Based on the Hiroyasu and Kadota's [41] these results are mainly because of the increase in fuel jet's pressure and penetration velocity which cause smaler droplet diameter and faster vaporization of fuel droplets. In Table 7, the performance characteristics of the engine by changing the nozzle holes number is presented.

Table 7. Performance comparison of the new engine with lower CR by different hole numbers

	6 holes	**New engine (7 holes)**	**8 holes**
MEP (Pa)	$1.294\times10^{+6}$	$1.228\times10^{+6}$	$1.173\times10^{+6}$
Net work (J)	2359.17	2238.41	2138.11
Power (kW)	39.318	37.303	35.637
Torque(N.m)	187.75	178.115	170.24

According to the data from Table 6, if the holes in each nozzle are reduced by one, the power of each cylinder will increase by 5.4%, and if this number changes to 8, the power will reduce by 4.4%. Therefore, by lowering the nuzzle hole numbers, the performance will increase, and Soot emission will reduce, but on the other hand, the NOx emission will increase.

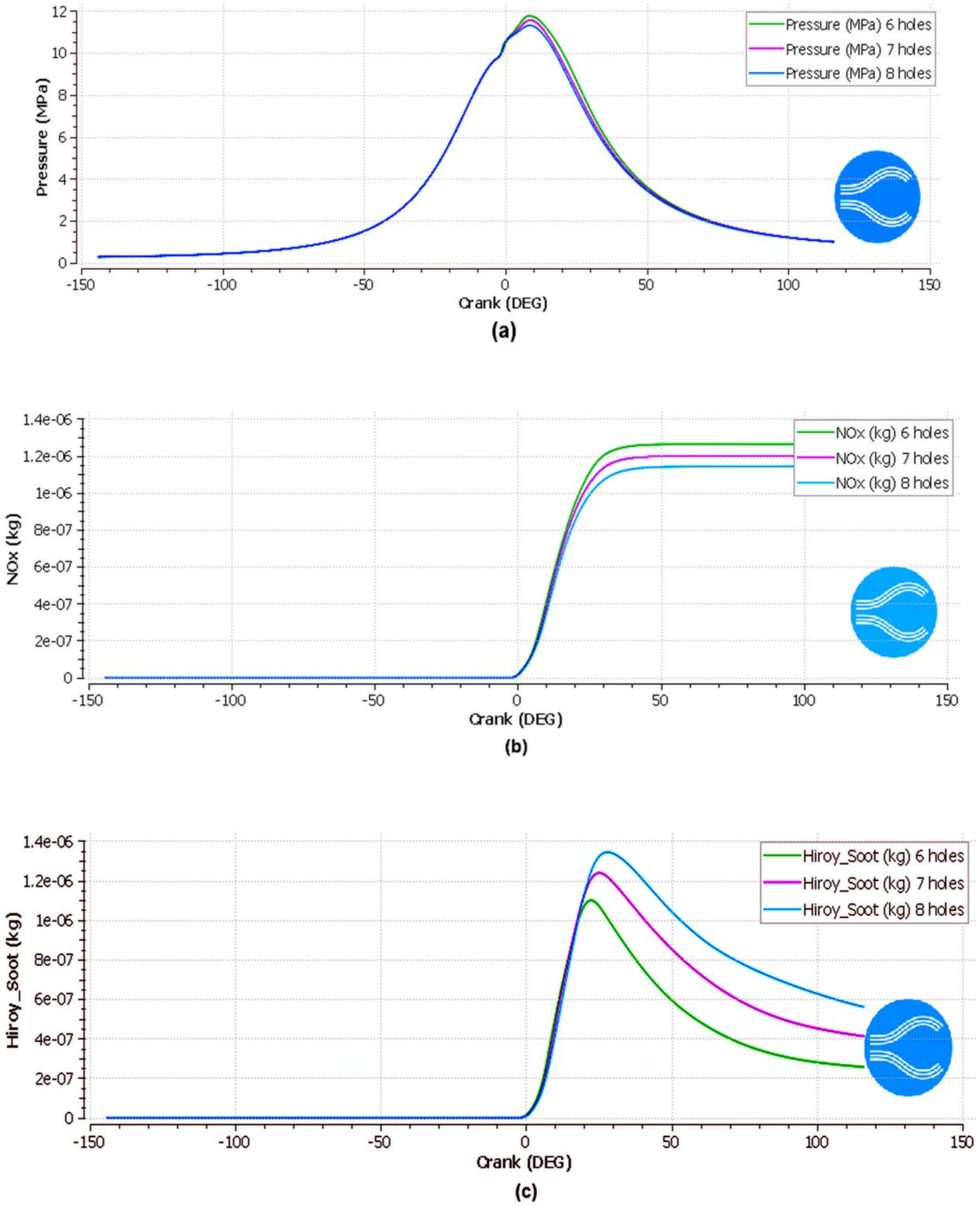


Figure 10. Comparison of the new engine with lower CR by different hole numbers, at different crank angle, (a) pressure, (b) NOx, and (c) Soot

Changing the Start Time of Injection

Advancing or retarding the start time of injection and finding the best time depends on the engine and varies from engine to engine, so for each engine, this should be surveyed individually [42]. In the OM457, the start time of injection is 13.7 crank angle before top dead center (BTDC), so for checking the impact of this parameter, the start time of injection changed to 18 and 10 crank angle BTDC. As its obvious, retarding the start time of injection fuel jet will mix earlier with the air, and the start of ignition will be retarded too (shown in Figure 11 at TDC).

For illustrating the impact of the start time of injection, the graphs of pressure – NOx and Soot are compared for these three different cases, as in Figure 12.

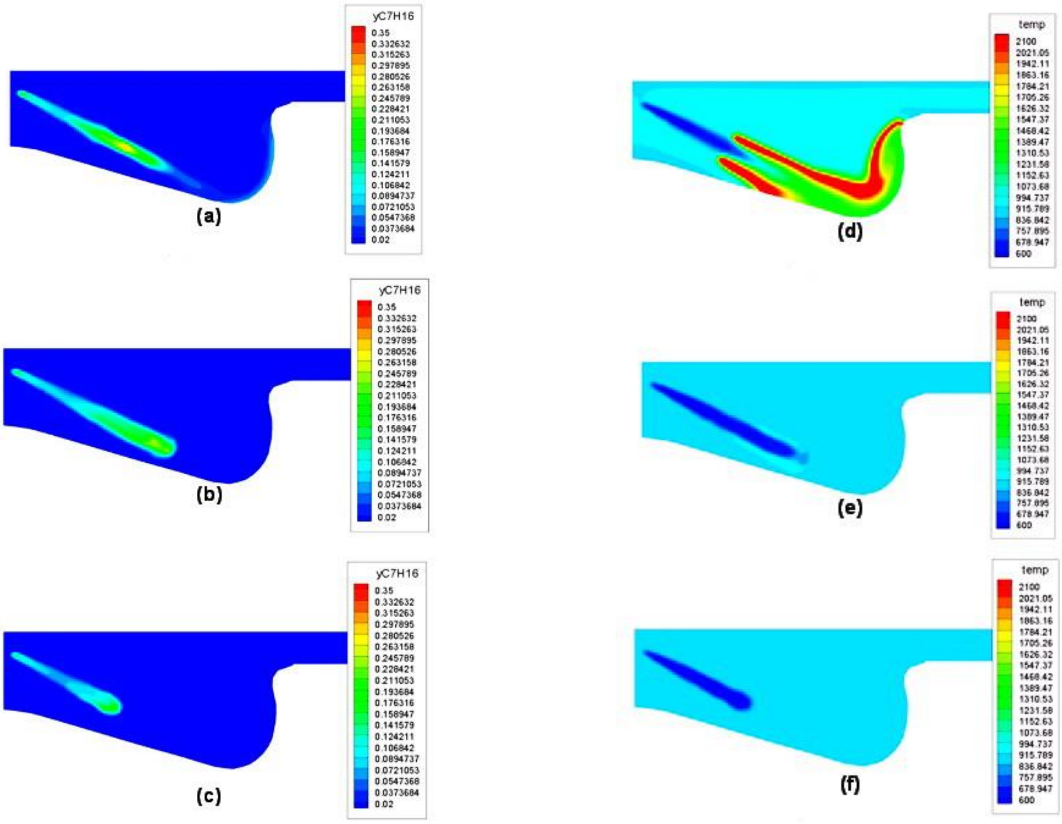


Figure 11. Contours of temperature and C_7H_{16} molar fraction at TDC for the different start time of injection: (a) C_7H_{16} molar fraction for retarded injection, (b) C7H16 molar fraction for the primitive engine, (c) C_7H_{16} molar fraction for advanced injection, (d) temperature for retarded injection, (e) temperature for primitive engine, (f) temperature for advanced injection

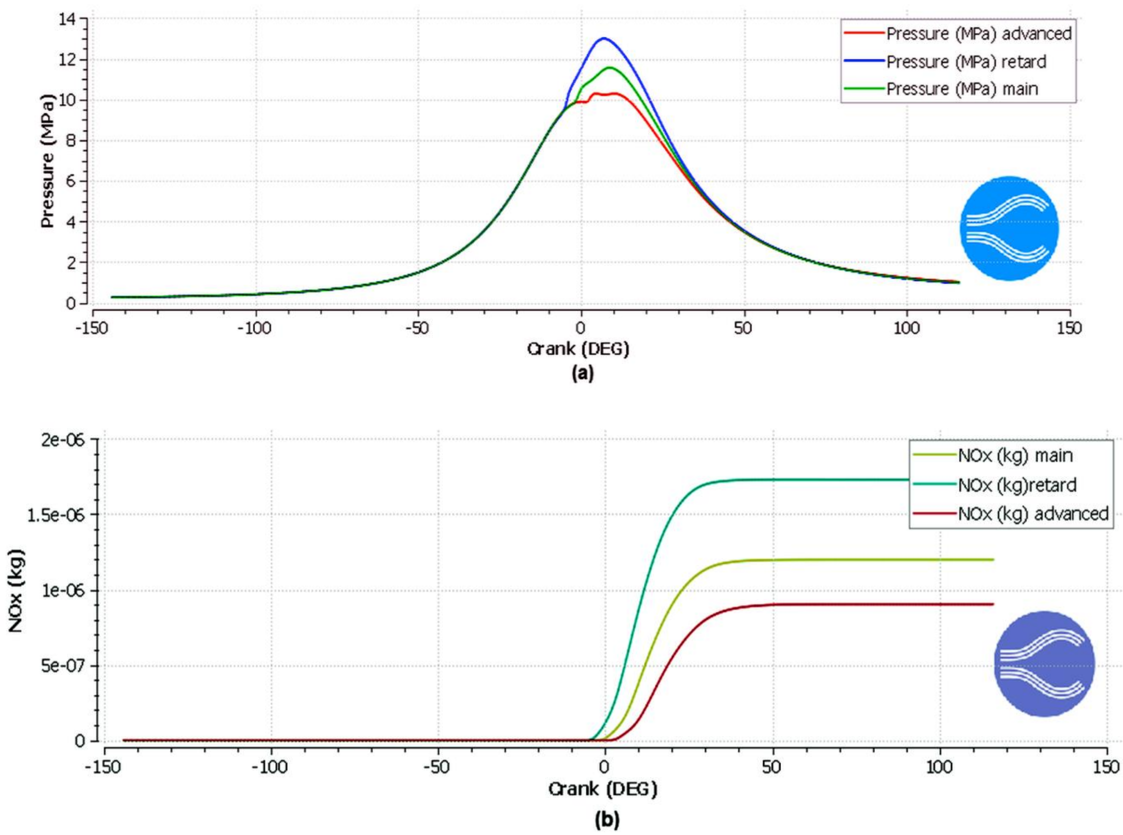
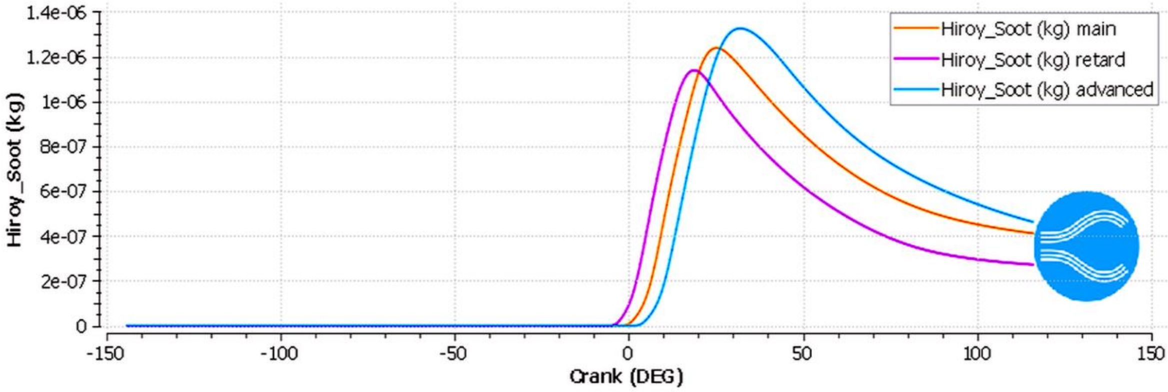


Figure 12. Comparison of the new engine with lower CR by the different start time of injection, at different crank angle, (a) pressure, (b) NOx, and (c) Soot



(c)

Figure 12. Comparison of the new engine with lower CR by the different start time of injection, at different crank angle, (a) pressure, (b) NOx, and (c) Soot (cont.)

As in Figure 12, because the fuel jet has more time for penetrating among the cylinder, retarding the time of injection, better ignition will occur [43], so the pressure will increase. Therefore, the performance will be improved. Soot emission will reduce, but NOx emission will be increased by 35%. For measuring the amount of improvement in performance, in Table 8, the performance characteristics of these three cases were presented.

Table 8. Performance comparison of the new engine with lower CR by the different start time of injection

	Retarded (-18o BTDC)	**New engine (-13.7o BTDC)**	**Advanced (-10o BTDC)**
MEP (Pa)	$1.303\times10^{+6}$	$1.228\times10^{+6}$	$1.168\times10^{+6}$
Net work (J)	2374.5	2238.41	2138.1
Power (kW)	39.575	37.303	35.635
Torque(N.m)	188.97	178.115	170.15

According to the results shown in Table 7, the performance of each cylinder will be increased by 6.1% if the start time of injection retarded to -18 crank angle BTDC.

Increasing the Performance

As mentioned before, by lowering the CR of the engine, the performance is reduced remarkably. In the previous section, it has been shown that by changing some parameters of the fuel injection, the performance of the engine will increase. In this section, by replacing all of those parameters simultaneously (in the way of improving the performance, which was reducing the diameter of the injector's nozzle - reducing the number of holes and retarding the start time of injection) the emissions and performance of the engine have been measured.

By reducing the CR, clearance volume will increase, so the pressure of the chamber will dropdown. If the inlet air and fuel mass rate increase by the same amount, not only the air-fuel ratio will be the same as the main engine (OM457), but also the pressure in the chamber will increase. So it seems that if the inlet air and fuel mass rate increase by 20%, this reduction on the CR will be mitigated. For surveying this idea, the inlet air, and mass fuel rate are increased by 20%. The results after applying these changes are shown in Figure 13 and Table 9 and compared with the main engine and the one with modified spray injection parameters.

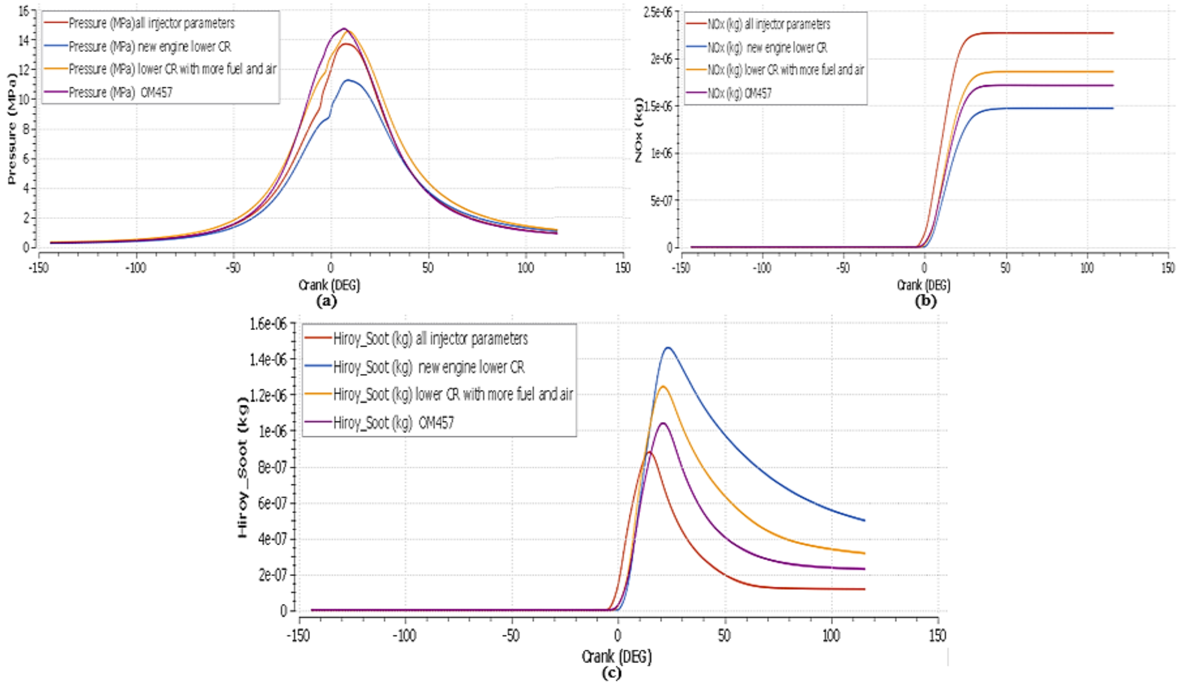


Figure 13. Comparison of OM457 with the new engine (lower CR), improved new engine and the new engine with more fuel and air, at different crank angle, (a) pressure, (b) NOx, and (c) Soot

Table 9. Performance comparison of OM457 with new engine (lower CR) , improved new engine and new engine with more fuel and air

	All Injector Parameters	New Engine Lower CR	Lower CR with More Fuel and Air	OM457
MEP (Pa)	$1.405\times10^{+6}$	$1.228\times10^{+6}$	$1.565\times10^{+6}$	$1.304\times10^{+6}$
Net work (J)	2560.71	2238.41	2852.86	2597.03
Power (kW)	42.6785	37.303	47.547	43.283
Torque(N.m)	203.782	178.115	227.03	206.67

Although changing the injectors parameter, the engine will almost reach the same performance as OM457, the NOx emission will increase by 34% because of higher in-cylinder temperature, which causes more NOx formation (This is evident in Figure 13 and Table 8). Meanwhile, the soot emission will reduce 23% because of better air and fuel mixture. On the other hand, if the inlet air and fuel mass flow increase, NOx emission will increase by 5.7%, but the performance will be increased by 9.8%. It is noticeable that the MEP of these two engines (one with more inlet air and fuel mass rate, and the other one which the injectors characteristics has been improved) will be higher (respectively 27% and 14.4%). As a result of this increase in the MEP, the strength of the material should be re-evaluated.

CONCLUSION

The performance of the engine drops down by lowering the CR of the OM457 heavy-duty diesel engine. Two different strategies were studied for mitigating this reduction on the performance, one was changing the injector's parameters simultaneously, and the other one was increasing the inlet air and fuel rate. For investigating the impact of injector's parameters on the performance and emission, three different parameters, including: nuzzle hole's diameter, nuzzle hole numbers and start time of injection were checked separately, and the impact of these parameters on the emission and performance of the engine has been identified. By changing all these three parameters on the injector, the performance almost reached the main OM457's, while the NOx emission increased, and Soot emission decreased.

In summary, the following conclusions can be listed based on this study:

1. By changing the connecting rod and crankshaft of OM457 with OM444, the CR will be reduced.
2. Reducing the CR will cause a reduction in performance and NOx emission of the engine, but Soot emission will increase.

3. Reducing the diameter of the injector nozzle and lowering the nozzle holes in constant fuel rate scenarios will increase the performance of the engine, but on the other hand, the NOx emission will be increased while the soot emission decreased.
4. In our case, retarding the start time of injection will increase the performance and reduce the amount of soot emission, but the NOx emissions of the engine will increase.
5. For mitigation of reducing the CR (which causes performance reduction), increasing the air and the fuel in the same ratio will be an acceptable solution.
6. By applying those changes for fuel injectors simultaneously, although the performance will be near the same as the main engine and Soot emission will decrease, but on the other hand, the NOx emission will increase.

Declaration of Conflicting Interests

The authors declared no potential conflicts of interest with respect to the research, authorship and/or publication of this article.

Funding

The authors received no financial support for the research, authorship and/or publication of this article.

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