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Full Length Research Paper

Impacts of fuel infusion discharge curve and injection pressure on overhauling power and burning parameters in substantial obligation (HD) diesel motor with computational liquid dynamics (CFD) reenactment

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In this study, the effects of fuel injection discharge curve and injection pressure onpower upgrade of heavy-duty diesel engine by simulation of combustion process in AVL-Fire software are discussed simultaneously. Hence, the fuel injection discharge curve is changed from semi-triangular to rectangular which is usual in common rail fuel injection system. Injection pressure with respect to amount of injected fuel and nozzle hole diameter are changed. Injection pressure is calculated by an experimental equation which is developed for heavy duty diesel engines with common rail fuel injection system. Power upgrade for 1000 and 2000 bar injection pressures are discussed. For 1000 bar injection pressure with 188 mg injected fuel and 3 mm nozzle hole diameter, power is upgraded about 19% in comparison to original state which is semi-triangular discharge curve with 139 mg injected fuel and 3 mm nozzle hole diameter, with no special change in cylinder pressure. On the other hand, both the NO_X and the Soot emissions decreased about 30 and 6%, respectively. Compared with the original state, in the case of 2000 bar injection pressure, with injected fuel and nozzle diameter, 196 mg and 2.6 mm respectively, the power is upgraded about 22%, whereas cylinder pressure has been fixed, and the NO_X and the Soot emissions are decreased to 36 and 20%, respectively.

Key words: Computational fluid dynamics (CFD) simulation, heavy-duty (HD) diesel engine, upgrading power, injection pressure, fuel injection discharge curve, combustion process.

INTRODUCTION

Power upgrade of internal combustion engines is a quantitative feature. Nowadays, many methods are used in this field. In fact, all of the methods which are used consider fuel and air inlet in combustion chamber. By increasing each one of these two parameters (fuel and input air) and by changing the strategies in fuel injection and air intake systems, power upgrade is possible. Hence, many parameters of fuel injection and air intake systems have effect on combustion parameters and output power of engine. Among the air intake system

Table 1. Geometrical and fuel injection data.

Engine parameter	Specification
Number of cylinder	16
Bore	0.215 m
Stroke	0.275 m
Connecting rod length	0.502 m
Rated speed	1000 rpm
Compression ratio	13.5:1
Number of nozzle hole	9
Hole diameter	0.003 m
Cone angle 1	140°
Cone angle 2	20°
Start of fuel injection	20°bTDC
Fuel used	Diesel
Fuel injection quantity	139 mg/cycle
Fuel injection duration	30°
Intake air pressure	4.2 bar
Intake air temperature	370K
Mechanical efficiency	0.905

characteristics, volumetric efficiency is one of the most effective parameter on power of internal combustion engine, because it is related to amount of input air. If the air inlet in combustion chamber does increase, the condition for better combustion and power upgrade is provided. Hence, many different methods are used for increasing the amount of air inlet which three of them are mentioned as follow (Heywood, 1988):

(i) Geometrical changing on input manifold (swirl, tumble…)
(ii) Turbo charging
(iii) Supercharging

In the field of improvement of air inlet conditions to upgrade power, numerous works have been done (Papyri et al., 1996; Justham et al., 2006). Certainly, computational methods are more common than experimental methods because of lower costs (Wu et al., 2004; Gosman, 1999; Nureddin et al., 2007).

Jemni et al. (2011) increased the volumetric efficiency and output power of heavy-duty diesel engine with improvement of the design of the inlet manifold in their computational fluid dynamic method. Numerous works have been done for improving the performance of turbocharger in order to increase the output power in heavy-duty diesel engines (Nishiguchi et al., 1982; Pattas et al., 1992; Lee et al., 1991). Sik Lee and Jung Choi (2002) in their experimental study investigated the improvement of a turbocharged diesel engine by means of injecting air into the intake manifold. The experimental results show that air injection into intake manifold of turbocharged diesel engine lead to the improved combustion characteristics and output power. Moreover, the intake air and fuel injection systems have significant effect on combustion characteristics and output power of internal combustion engines (Goldsworthy, 2012). CeliktenI (2003) investigated the effect of injection pressure on engine performance and exhaust emission in indirect diesel engine experimentally. The injection pressure is changed from 100 to 250 bar and results are discussed in different throttle positions. The results show that by increasing injection pressure, output power of diesel engine increases.

In this study, power upgrade of heavy-duty diesel engine (RK215) through changing the fuel injection discharge curve from semi-triangular to rectangular and increasing injection pressure has been discussed simultaneously. Increasing injection pressure by increasing total mass of fuel and reducing nozzle diameter has been done and also air inlet pressure is fixed. An experimental equation is used for calculating the injection pressure with rectangular discharge curve (Zhao, 2012). This strategy is used for upgrading the power of the diesel engine with constant cylinder peak pressure. Computational fluid dynamic method has been applied for simulation of combustion process in AVL-Fire software.

ENGINE GEOMETRY AND REQUIRED DATA

The engine under study in this research is a heavy-duty diesel engine. The geometrical and fuel injection data of the engine are summarized in Table 1. The AVL-Fire software is used to simulate the preprocessing and postprocessing of the engine. In preprocessing the entire computational domain and moving mesh for 180° to 540° crack angle is created, (Figure 1). Figure 1 illustrates the method of meshing and boundary condition determination. In Figure 2, the various parts of the piston are shown with different colors. In this simulation the thermal boundary conditions are used. Three thermal boundary conditions are represented Table 2.

FUEL INJECTION PRESSURE

Pressure in the nozzle during injection is the most important factor for engine performance, as well as the injector performance in terms of operating speed to allow short and multiple injections and injection repeatability. Injection spray momentum is regarded as an ideal measurement to compare systems for good air-fuel mixing and efficient combustion. Alternatively, Needham and Whelan used mean effective injection pressure (MEIP) which they reported gave reliable measure of average injection pressure and hence injection energy (Zhao, 2012):

$$meip = \left(\frac{0.0426 \times Q \times N}{d^2 \times \theta \times h}\right)^2$$

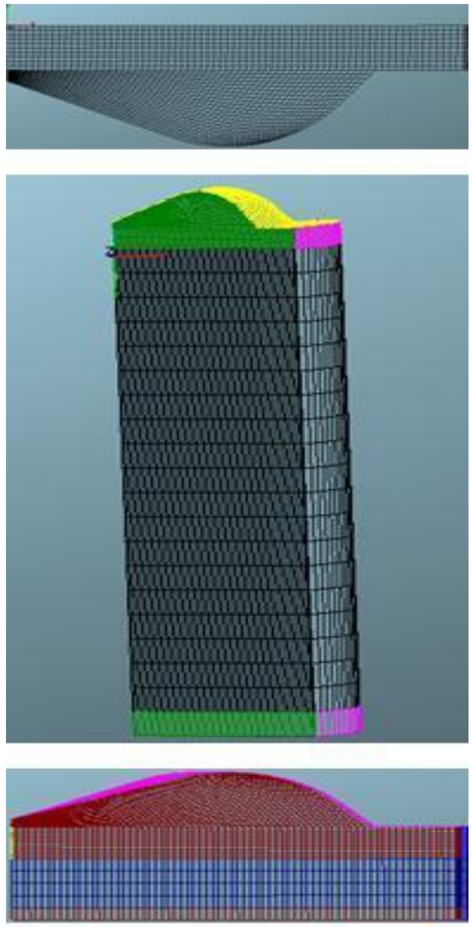

Figure 1. Parts of postprocessing in AVL-Fire software.

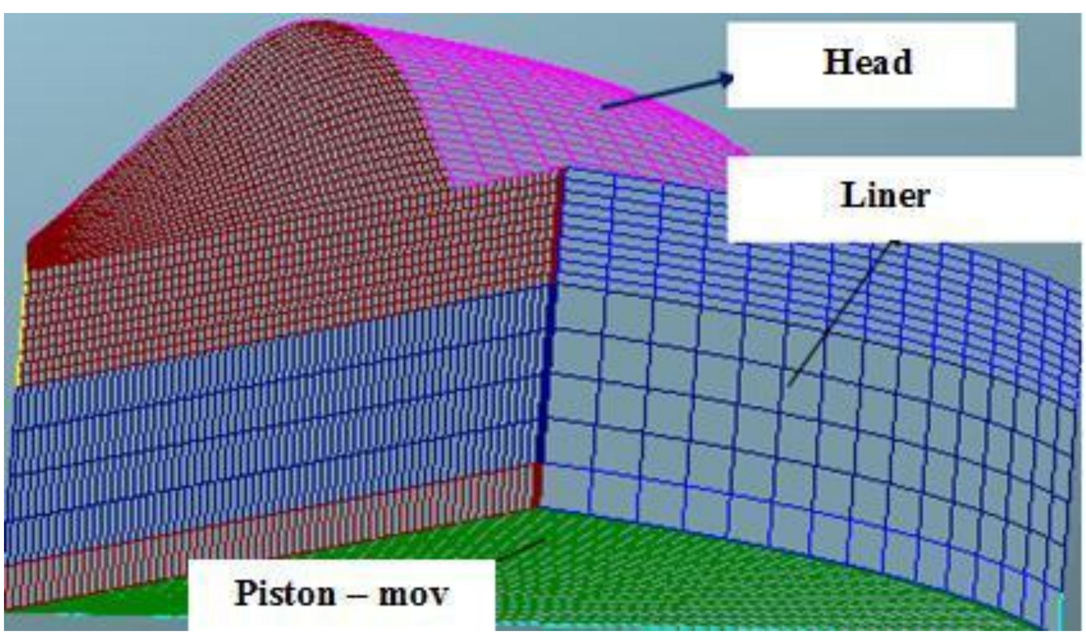

Figure 2. Boundary condition that used in simulation.

Table 2. Boundary temperatures that used in simulation.

Sections	Temperature(K)
Head	553
Liner	403
Piston-mov	593

Table 3. Models for calculation.

Models	Specification
Heat transfer	Dukowicz
Break up	Wave
Wall interaction	Wall jet1
Combustion	Eddy break up
NO	Zeldowich
Soot	Kennedy-Horoyasu-Magnussen

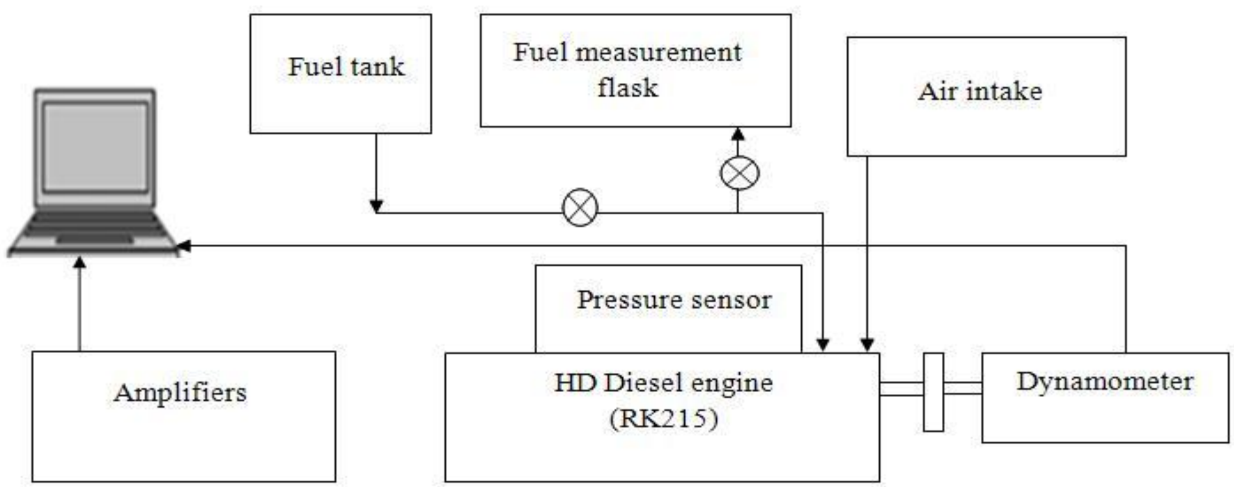

Figure 3. Schematic of test engine.

Where: Q= fuelling ($mm^3/injection$), N= engine speed (rev/min), θ= injection period (deg crank), d= hole diameter (mm), h=number of injector nozzle holes,
0.0426 is a constant which includes the discharge coefficient of typical nozzle hole.

The governing equations introduced in computational fluid dynamic model are used in postprocessing to solve the problems with the computational fluid dynamic method.

COMPUTATIONAL FLUID DYNAMIC MODEL

In this study, computational fluid dynamic method is used for solving the problem. The models which are used in simulations are listed in Table 3. The heat process is described by a model originally derived by Dukowicz (1980) and Emami and Jafarmadar (2013).With the assumption of uniform droplet temperature, the rate of droplet temperature change is determined by the energy balance equation, which states that the energy conducted to the droplet either heats up the droplet or supplies heat for vaporization (Figure 3).

The standard WAVE model is used for simulating the breakup process (Reitz, 1987). This model has two break-up regimes. One of them is important for high pressure injection systems. By varying the characteristic breakup time via the model constant C_2, calculated results can be fitted to measurements or visualization data. For this model the droplet size has to be set to the nozzle hole diameter. The constant C_2 corrects the characteristic break-up time and suitable value of that is

Table 4. Measured peak pressure from each cylinder.

Number of cylinder	Peak pressure
Cylinder 1	166
Cylinder 2	162
Cylinder 3	161
Cylinder 4	162
Cylinder 5	163
Cylinder 6	160
Cylinder 7	160
Cylinder 8	164
Cylinder 9	162
Cylinder 10	163
Cylinder 11	162
Cylinder 12	163
Cylinder 13	160
Cylinder 14	161
Cylinder 15	162
Cylinder 16	163

12 for diesel engine, whereas the value of C_1 should be kept at 0.61 (AVL Manual, 2004). The method that is used for wall interaction is wall jet1 model. This model in principle is based on the spray/wall impingement model of Reitz et al. (1996). In current implemented model, it is assumed that a droplet which hits the wall suffers one of the two consequences, namely rebound or reflection in the manner of a liquid jet, depending on the Weber number. The transition criterion between these two regimes is described by a critical Weber number which is taken to be We_c = 80 (Wei et al., 2013).

In this study, Eddy break up model is used for modeling the combustion process. It is based directly on a physical assumption on the turbulent reaction rate. The instantaneous reaction rate in laminar or turbulent flows can be represented in the form of Arrhenius equation (AVL Manual, 2004).

Finally, for modeling nitrogen oxide and soot emissions, the Zeldowich and Kenndey-Hiroyasu-Magnussen models are used respectively that common for modeling nitrogen oxide and soot emissions in diesel engines (Magnussen, 2005).

EXPERIMENTAL SETUP AND VALIDATION

For this experiment, a HD diesel engine equipped with one piezotron quartz pressure sensor per cylinder and coupled to transient AC dynamometer is used. Installed sensor is from Kistler Company (type7613c). These sensors are used for measuring each cylinder pressure and output power measured with dynamometer. Figure 3 shows the details of experimental setup. Table 4 represents the measured cylinder peak pressure from the engine test while the peak pressure from simulation is about 136.5 bar. In order to complete the validation process, it is necessary to compare the indicated power from simulation with engine test data. The output power from the engine test is about 3815 hp. Considering the mechanical efficiency, the real indicated power is about 4216 hp. Figure 4 shows the cylinder pressure versus volume that is taken from simulating data. The indicated power of simulation resulting from this diagram is 3878 hp. Table 5 represents all results from engine test and simulation, together. The observed difference in indicated power between the simulation state and engine test is about 338 hp. The difference is due to the fact that the simulation is for single cylinder but the test is carried out on the RK215 engine with the 16 cylinders.

RESULTS AND DISCUSSION

The engine test and validation of the simulation has been carried out with a semi-triangular injection curve. With respect to the effect of rectangular discharge curve and injection pressure with increasing amount of fuel and reducing nozzle diameter, power upgrade for two injection pressures (1000 and 2000 bar) has been discussed. Fuel injection discharge curve for 1000 and 2000 bar injection pressures are shown in Figures 5 and 6, respectively. All of these changes for combustion parameters and output power are compared with the original working state of the engine.

Effect of injection discharge curve and 1000 bar injection pressure on power upgrade

The combustion process has been simulated due to investigation of the effect of fuel injection discharge curve and 1000 bar injection pressure on power upgrade. Diagrams which are presented here include: output power, mean cylinder pressure, mean cylinder temperature, NO and Soot mass fraction.

The best condition for upgrading output power is for state with 188 mg fuel total mass and 3 mm nozzle diameter that output power is upgraded about 19%. Figure 7 shows the cylinder pressure versus volume that describes amount of upgrading output power. The mean cylinder pressure and mean cylinder temperature are shown in Figures 8 and 9. At the worst condition the peak temperature increases about 150K whereas peak pressure doesn't have a significant change.

In Table 6, the values of peak pressure and temperature for semi-triangular and rectangular discharge curves with 1000 bar injection pressure are observed. The NO and Soot emissions are shown in Figures 10 and 11. The main parameters affecting NO emission are the cylinder temperature, air concentration and the time required for the reaction (Magnussen, 2005). Increase of the cylinder temperature about 150K resulted in increase of NO emission about 30%. By increasing the amount of fuel without any change in air intake, it is reasonable that the Soot emission increases. This increase is about 6%.

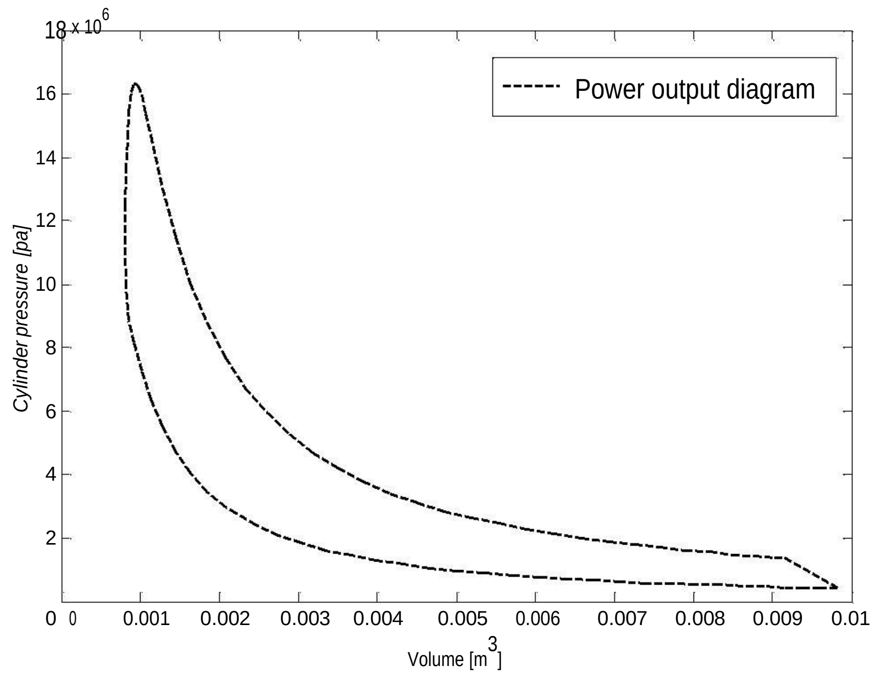

Figure 4. Power output diagram from simulation.

Table 5. Results of engine test and simulation.

Variables	Simulation	Engine test
Cylinder peak pressure [bar]	163.5	160-166
Indicated Power [hp]	3878	4216

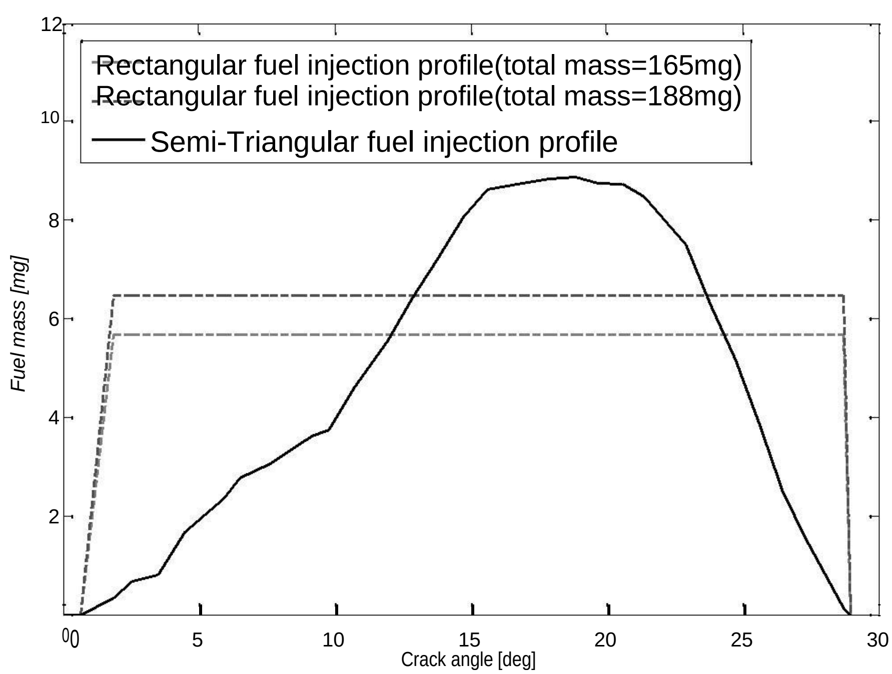

Figure 5. Semi-triangular discharge curve and rectangular with 1000 bar fuel injection pressure.

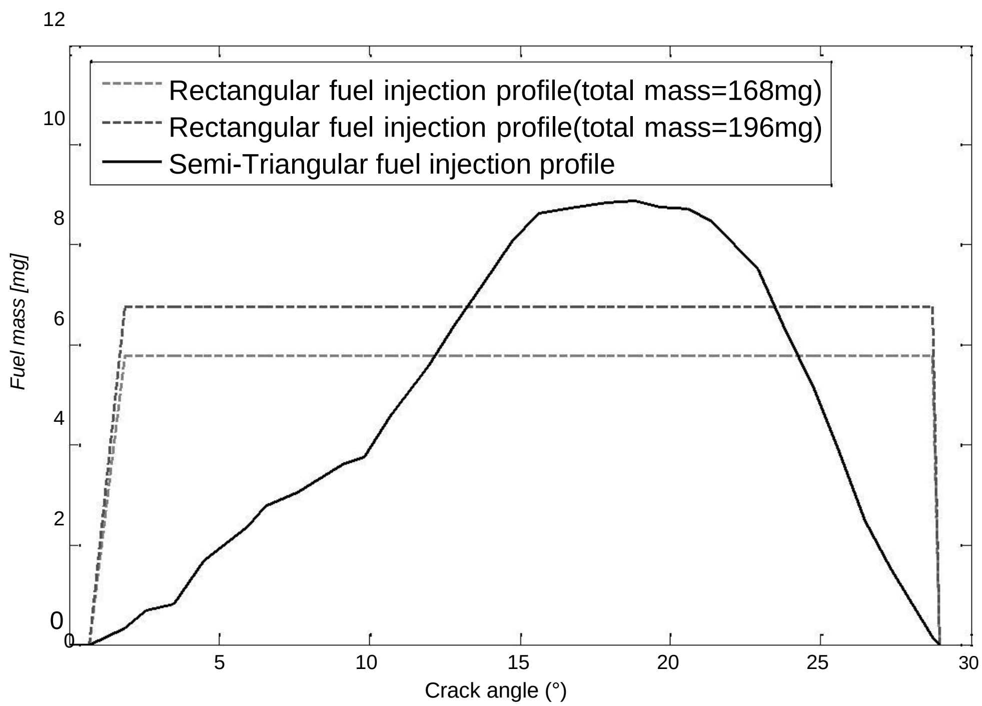

Figure 6. Semi-triangular discharge curve and rectangular with 2000 bar fuel injection pressure.

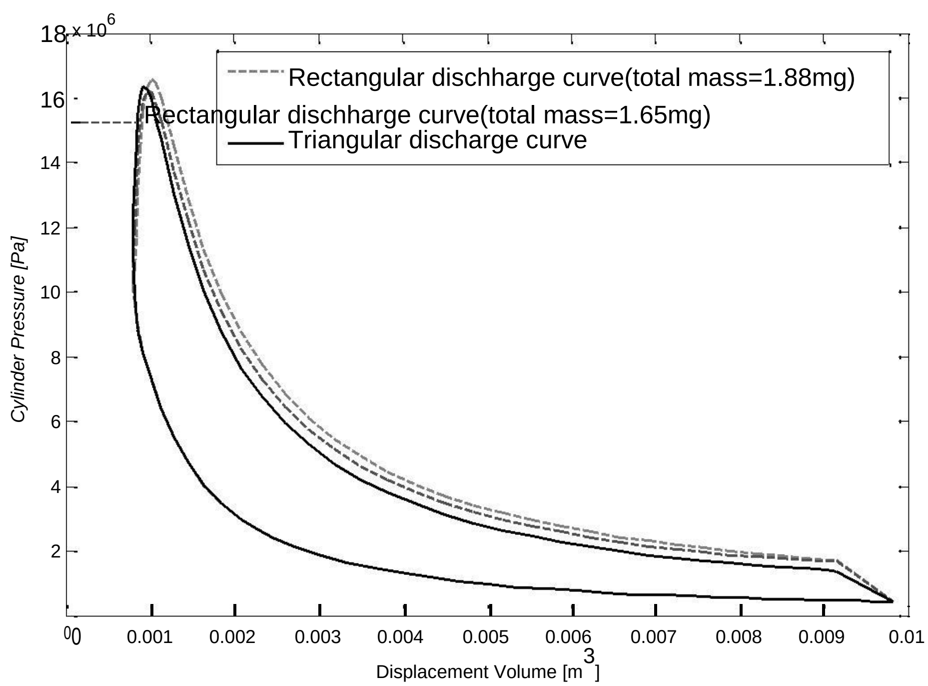

Figure 7. Cylinder pressure versus volume in 1000 bar fuel injection pressure.

Effect of injection discharge curve and 2000 bar injection pressure on power upgrade

To achieve the best condition of power upgrade, combustion process has been simulated with 2000 bar injection pressure, too. Output power of combustion process in this simulation is upgraded about 22%. The increase of output power here is shown in Figure 12. The cylinder pressure and temperature are shown in Figures 13 and 14, respectively. With increase of injection

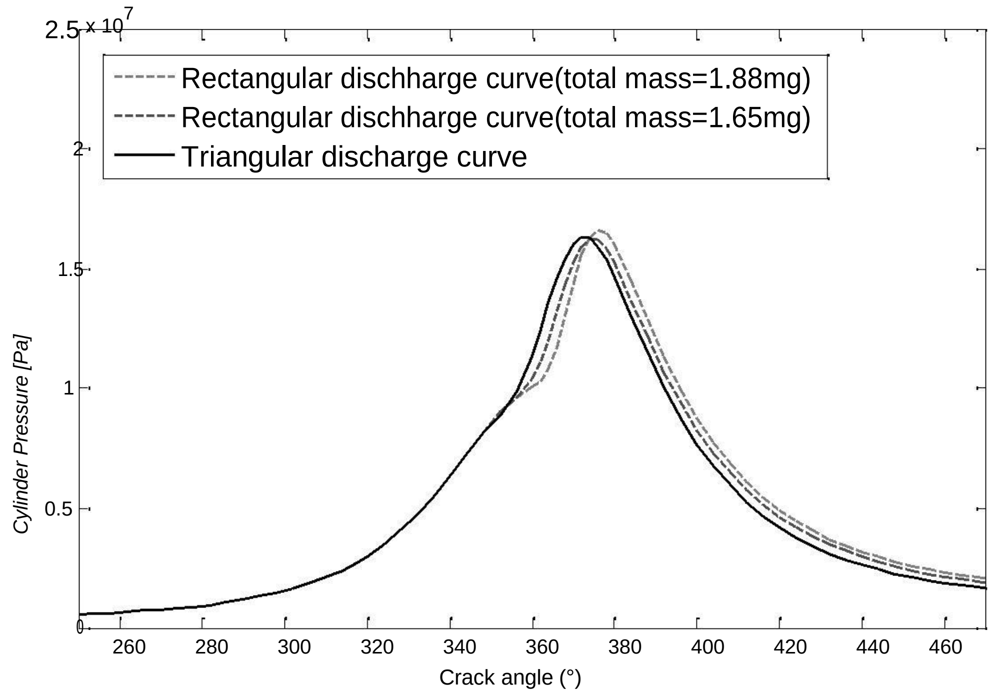

Figure 8. Mean cylinder pressure in 1000 bar fuel injection pressure.

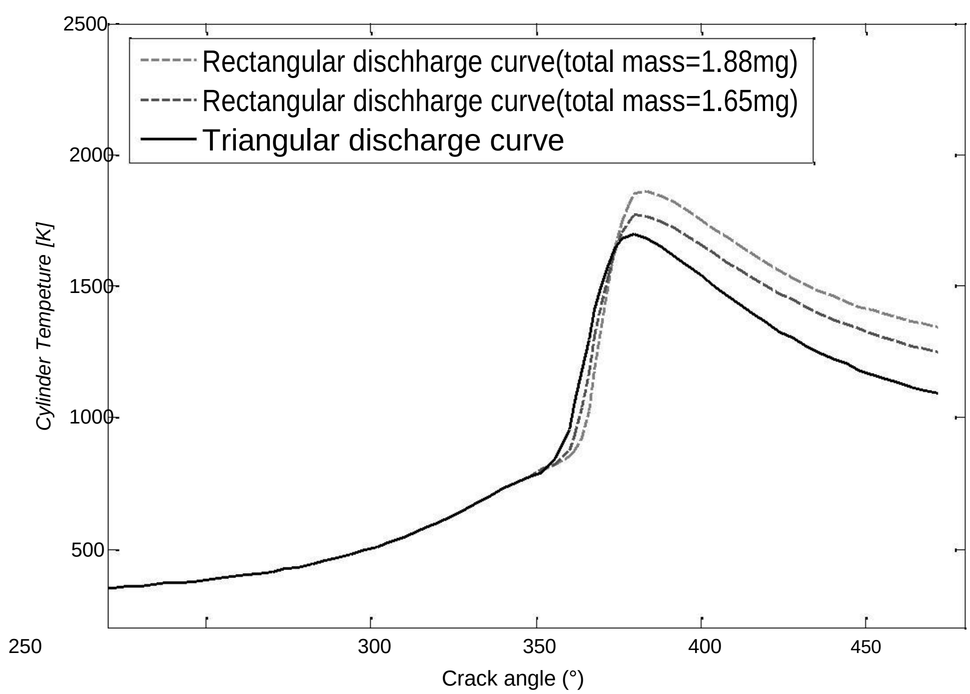

Figure 9. Mean cylinder temperature in 1000 bar fuel injection pressure.

pressure, peak temperature increases but peak pressure has no special change. The amount of cylinder pressure and temperature for semi-triangular and rectangular discharge curves with 2000 bar injection pressure are reported in Table 7.

The NO and Soot mass fraction are shown in Figures 15 and 16, respectively. The best condition for output power is achieved by the total mass of 196 mg and with

Table 6. The amount of peak pressure and temperature for 1000 bar fuel injection pressure.

Discharge curve	Total mass (mg)	Nozzle diameter (mm)	Peak pressure (bar)	Peak temperature (K)
Semi-triangular	139	4	163.5	1700
Rectangular	165	2.8	166	1770
Rectangular	188	3	162	1850

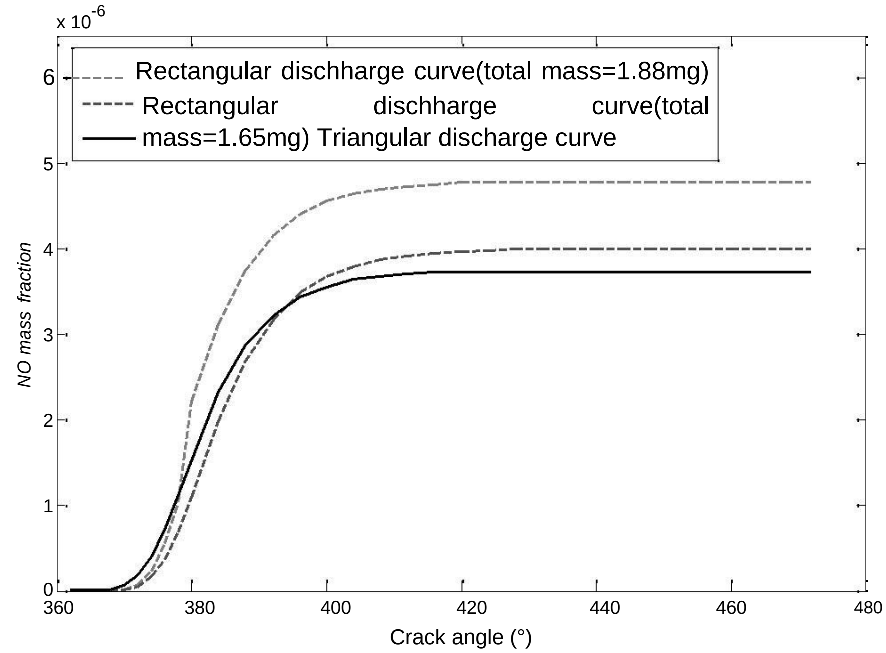

Figure 10. NO mass fraction in 1000 bar fuel injection pressure.

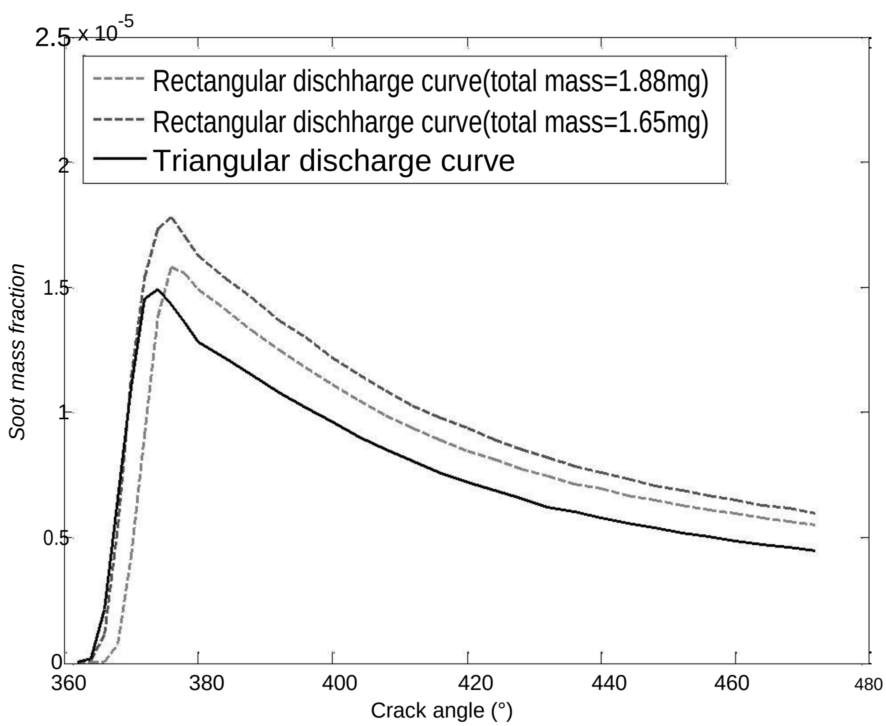

Figure 11. Soot mass fraction in 1000 bar fuel injection pressure.

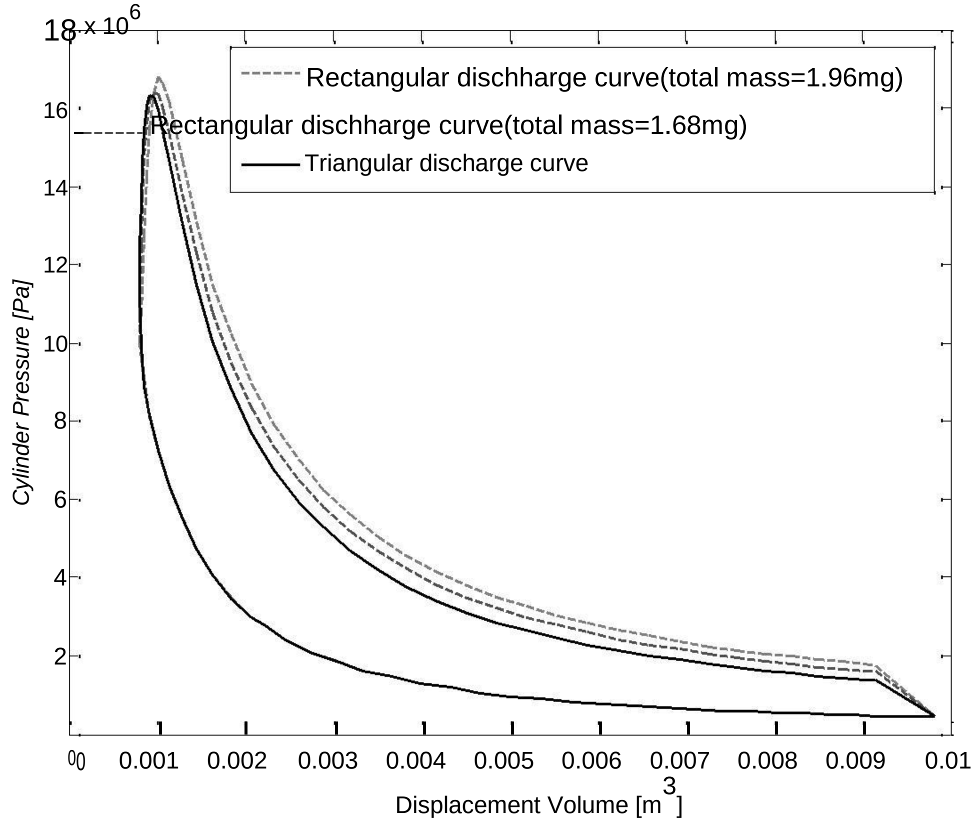

Figure 12. Cylinder pressure versus volume in 2000 bar fuel injection pressure.

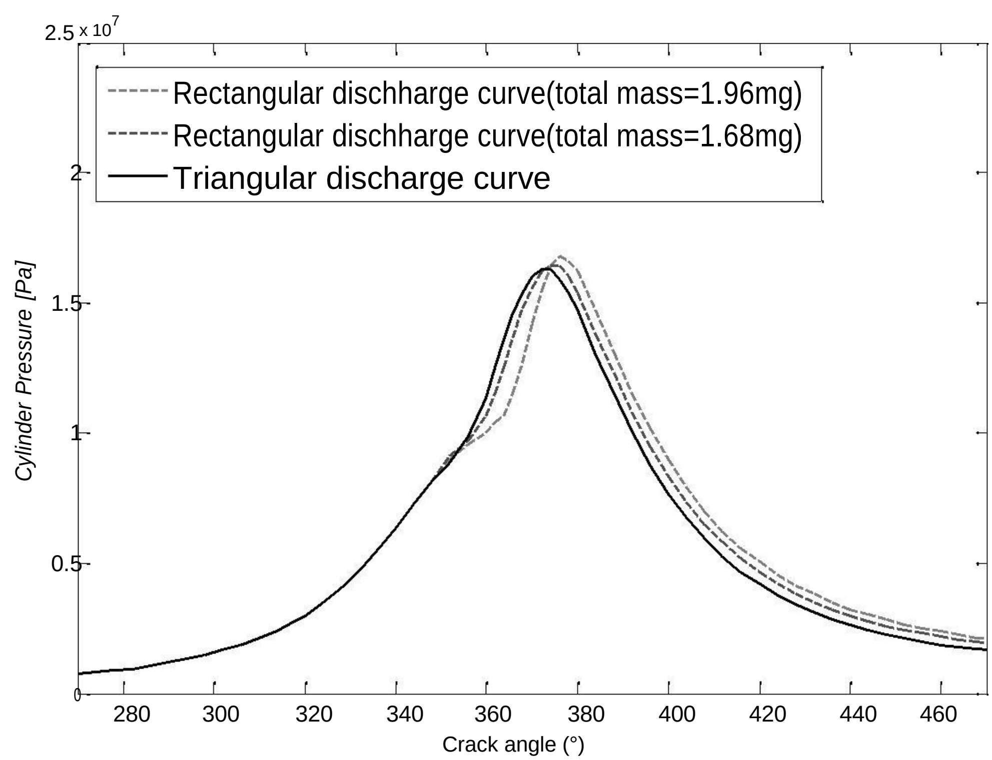

Figure 13. Mean cylinder pressure in 2000 bar fuel injection pressure.

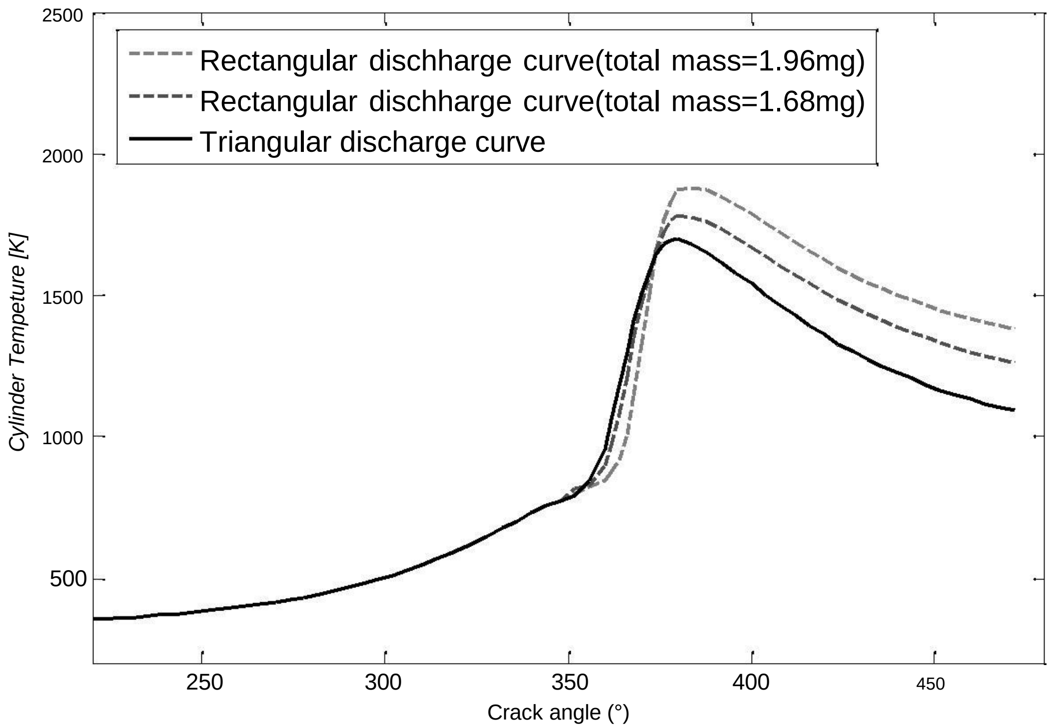

Figure 14. Mean cylinder temperature in 2000 bar fuel injection pressure.

Table 7. The amount of peak pressure and temperature for 2000 bar fuel injection pressure.

Discharge curve	Total mass (mg)	Nozzle diameter (mm)	Peak pressure (bar)	Peak temperature (K)
Semi-triangular	139	4	163.5	1700
Rectangular	168	2.4	164	1780
Rectangular	196	2.6	167	1870

nozzle diameter of 2.6 mm, that the NO and Soot emissions increase by 36 and 20%, respectively.

CONCLUSION

In this research, the effect of change of the fuel injection discharge curve from semi-triangular to rectangular and the effect of injection pressure on combustion parameters and output power have been investigated by simulation of combustion process in AVL-Fire software. The effect of injection pressure on output power with rectangular discharge curve is considered. For validation of the simulation process the engine test is carried out. In the engine test the peak pressure of each cylinder and output powerare measured and compared with results of simulation. The peak pressure measured with the pressure sensor of Kistler Company and output power measured with AC dynamometer. The simulation and the experimental results are in a good agreement.

Rectangular fuel injection discharge curve and 1000 bar injection pressure

(i) With 188 mg total mass and 3 mm nozzle hole diameter, power upgrade is about 19%. NO and Soot emissions increase by 30 and 6%, respectively.
(ii) With 165 mg total mass and 2.8 mm nozzle hole diameter, power upgrade is about 12%. NO and Soot emissions increase by 12 and 16%, respectively.

Rectangular fuel injection discharge curve and 2000 bar injection pressure

(i) With 196 mg total mass and 2.6 mm nozzle hole diameter, upgrading power is about 22%. NO and Soot emissions increase by 36 and 20%, respectively.
(ii) With 168 mg total mass and 2.4 mm nozzle hole diameter, upgrading power is about 13.5%. NO and Soot emissions increase by 14 and 19%, respectively.

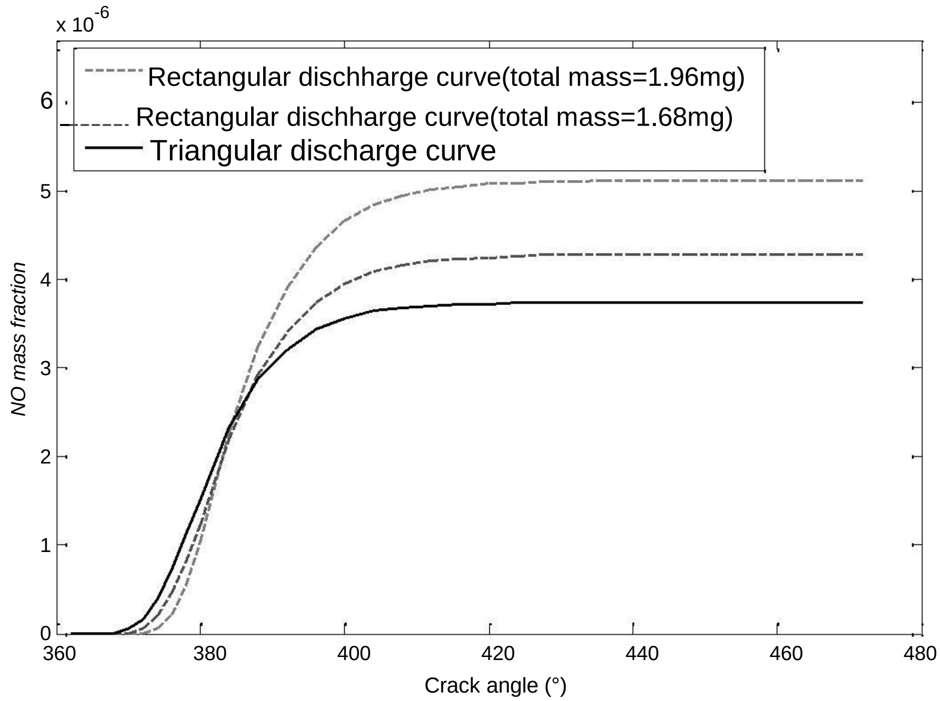

Figure 15. NO mass fraction in 2000 bar fuel injection pressure.

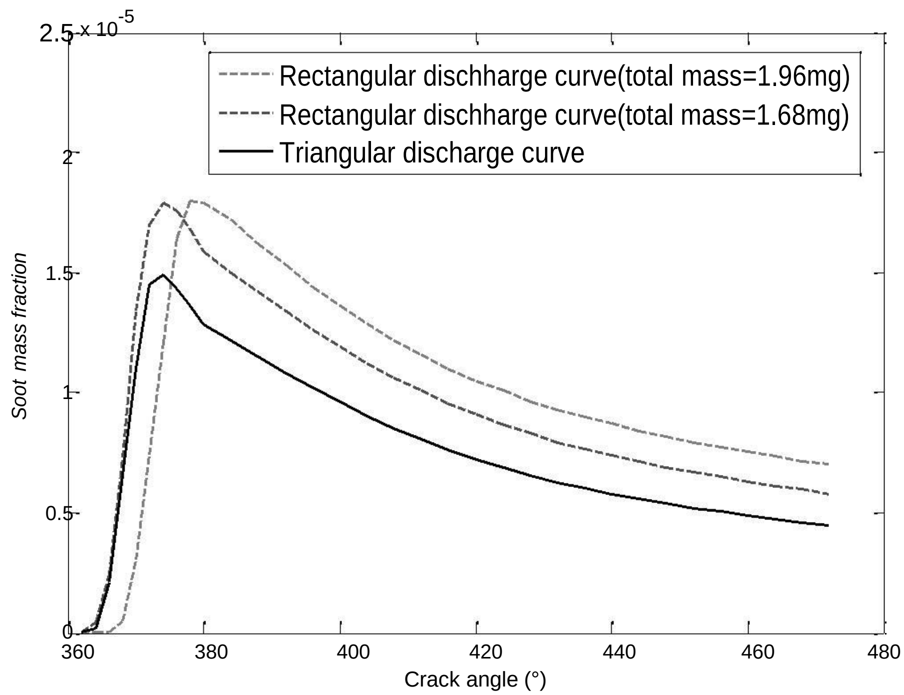

Figure 16. Soot mass fraction in 2000 bar fuel injection pressure.

The noticeable increase in the emissions and the output power has been observed by the change of the fuel injection discharge curve and increase of injection pressure.

The negative effect of increase in Soot emission due to increase of fuel total mass can be compensated by application of filters and/or increasing the amount of intake air. It should be mentioned that the formation of nitrogen monoxide is a chemical reaction of combustion process in the engine. Therefore, the effective parameters on combustion process also have the influence on NO emission. The influencing parameters in producing nitrogen monoxide are: the time of the reaction, air concentration and specially, the peak temperature which is the most influencing factor. In fact, by increasing these parameters the amount of nitrogen monoxide will increase too. So increase of NO emission is related to increase of peak temperature in combustion process. The main goal of this study is to upgrade the output power by fixing the peak pressure in order to control the knock in combustion chamber. However, the increase in NO and Soot emissions is inevitable with this method.

Conflict of Interest

The authors have not declared any conflict of interest.

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