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Effect of fuel injection discharge curve and injection pressure on upgrading power and combustion parameters in HD diesel engine with CFD simulation

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ABSTRACT

In this study, the effects of fuel injection discharge curve and injection pressure on power 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 1000bar injection pressure with 188mg injected fuel and 3mm nozzle hole diameter , power is upgraded about 19% in comparison to original state which is semi-triangular discharge curve with 139mg injected fuel and 3mm 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 2000bar injection pressure, with injected fuel and nozzle diameter, 196mg and 2.6mm respectively, the power is upgraded about 22%, whereas cylinder pressure has been fixed, and the NO_X and the Soot emissions are decreased 36% and 20%, respectively.

1) 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 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 amount of air inlet which three of them are mentioned as follow [1]:

- Geometrical changing on input manifold
- Turbo charging
- Supercharging

In the field of improvement of air inlet conditions to upgrade power, the numerous works have been done [2, 3]. Certainly, computational methods are more common than experimental methods because of lower costs [4-6].

Jemni and et al. 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. [7].

Numerous works have been done for improving the performance of turbocharger in order to increase the output power in heavy duty diesel engines [8-10]. Lee and Choi 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 [11].

Celikten investigated the effect of injection pressure on engine performance and exhaust emission in indirect diesel engines by experimental work. 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 [13].

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 [14]. 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.

2) Engine geometry and requirement 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 post processing of the engine.

Table1: Geometrical and fuel injection data

Engine parameters	Specification
Number of cylinder	16
Bore	0.215m
Stroke	0.275
Connecting rod length	0.502m
Rated speed	1000rpm
Compression ratio	13.5:1
Number of nozzle hole	9
Hole diameter	0.003m
Cone angle1	140^0
Cone angle2	20^0
Start of fuel injection	20^0bTDC
Fuel used	Diesel
Fuel injection quantity	139mg/cycle
Fuel injection duration	30^0
Intake air pressure	4.2bar
Intake air temperature	370 K
Mechanical efficiency	0.905

In preprocessing the entire computational domain and moving mesh for 180° to 540° crack angle is created (Figure 1).

The governing equations introduced in section (4) are used in post-processing to solve the problems with the computational fluid dynamic method.

3) 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 [14]:

$$meip = \left(\frac{0.0426 \times Q \times N}{d^2 \times \theta \times h}\right)^2 \qquad (1)$$

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.

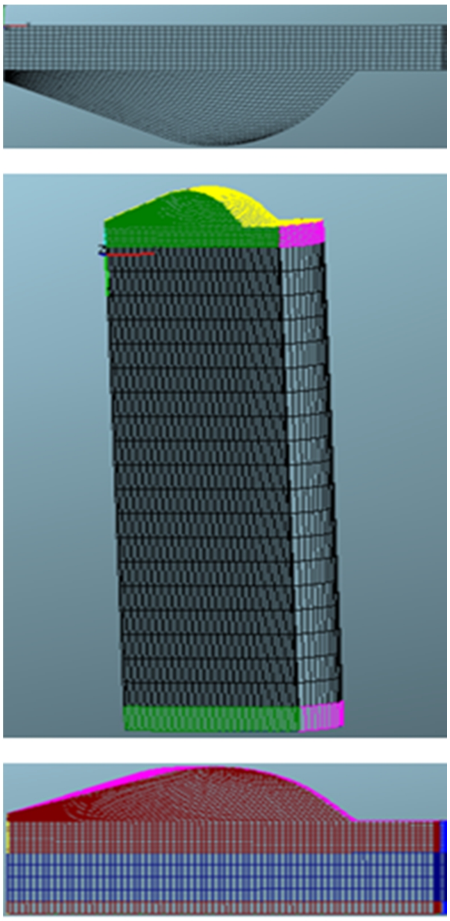

Figure 1: part of pre-processing in AVL-Fire software

4) 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 2.

Table 2: Models for calculation

Models	Specification
Heat transfer	Dukowicz
Break up	Wave
Wall interaction	Wall jet 1
Combustion	Eddy break up
NO	Zeldowich
Soot	Kennedy-Horoyasu-Magnussen

The heat process is described by a model originally derived by Dukowicz [15, 16]. 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.

The standard WAVE model is used for simulating the breakup process [17]. This model has two break-up regimes. One of them is important for high pressure injection systems, which are high velocities Rayleigh used. 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 12 for diesel engine, whereas the value of C_1 should be kept at 0.61[18].

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 and et al [19]. 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 W_{e_c} =80 [20].

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 [18].

At least 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 [16, 21].

5) Experimental setup and validation

For this experiment, a HD diesel engine equipped with a piezotron quartz pressure sensor and coupled to transient AC dynamometer is used. Installed sensor is taken from Kistler Company and with the model of type7613c. These sensors are used for measuring of each cylinder pressure and output power measured with dynamometer. Figure (2) shows the details of experimental setup.

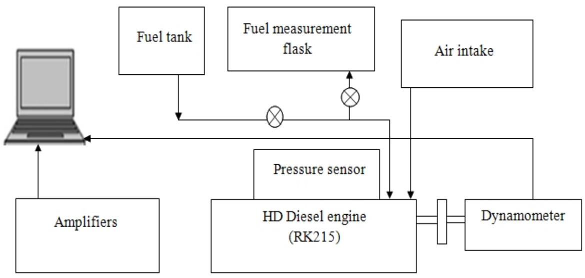

Figure 2: Schematic of test engine

Table 3 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. The output power from the engine test is about 3815hp with considering of the mechanical efficiency the real indicated power is about 4216 hp.

Table 3: Measured cylinder peak pressure

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

Figure 3 shows the cylinder pressure versus volume that is taken from simulating data.

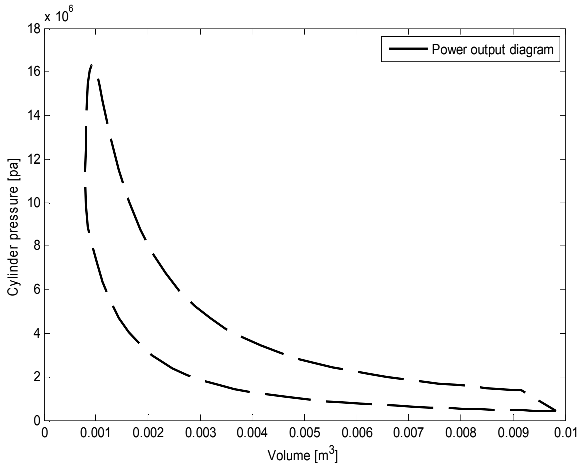

Figure 3: Power output diagram from simulation

The indicated power of simulation resulting from this diagram is 3878 hp. Table 4 Represent all results together from engine test and simulation in one place.

Table 4: Results of engine test and simulation

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

The observed difference of 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.

6) Result 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, upgrading power for two injection pressure (1000 bar and 2000 bar) has been discussed. Fuel injection discharge curve for 1000 bar and 2000 bar injection pressure are shown in figures (4) and (5), respectively. All of these changes for combustion parameters and output power are compared with the original working state of the engine.

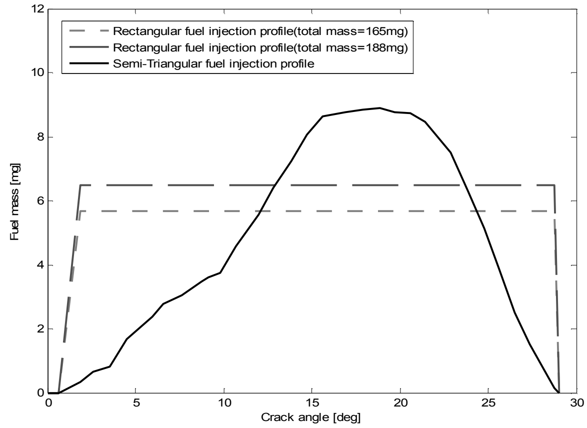

Figure 4: Semi-triangular discharge curve and rectangular with 1000 bar fuel injection pressure

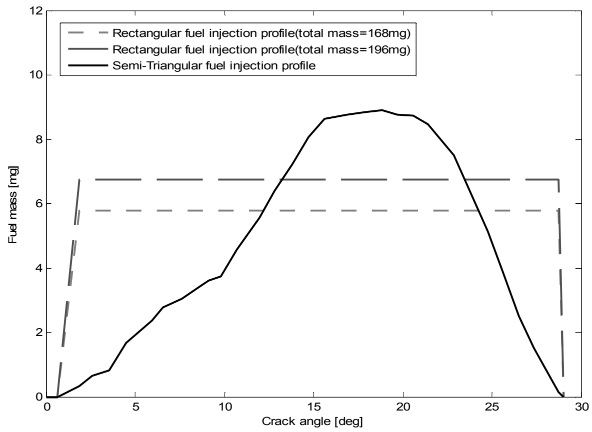

Figure 5: Semi-triangular discharge curve and rectangular with 2000 bar fuel injection pressure

6-1) Effect of fuel injection discharge curve and injection pressure on power upgrade

The combustion process has been simulated due to investigation the effect of fuel injection discharge curve and 1000bar injection pressure on power upgrade. Diagrams which are presented in this section 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 188mg fuel total mass and 3mm nozzle diameter that output power is upgraded about 19%. Figure 6 shows the cylinder pressure versus volume that describes amount of upgrading output power.

The mean cylinder pressure and mean cylinder temperature have been shown in Figures 7 and 8. At the worst condition the peak temperature increases about 150 K whereas peak pressure doesn't have a significant change.

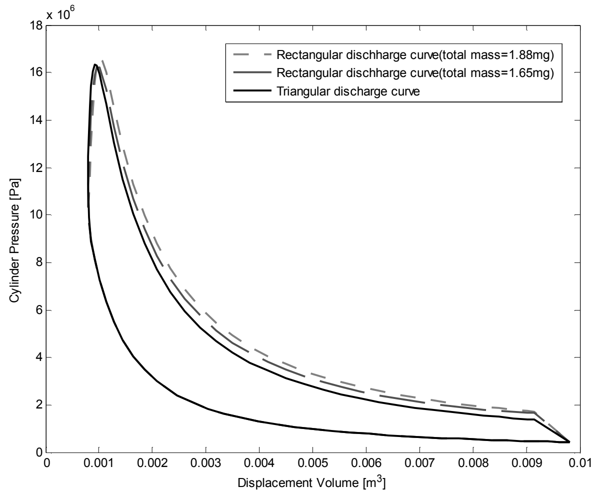

Figure 6: Cylinder pressure versus volume in 1000 bar fuel injection pressure

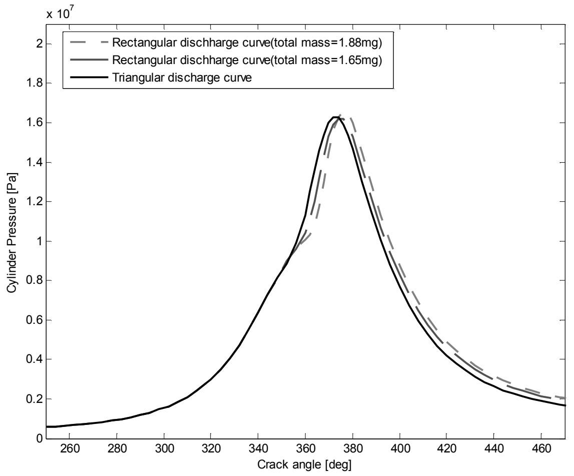

Figure 7: Mean cylinder pressure in 1000bar fuel injection pressure

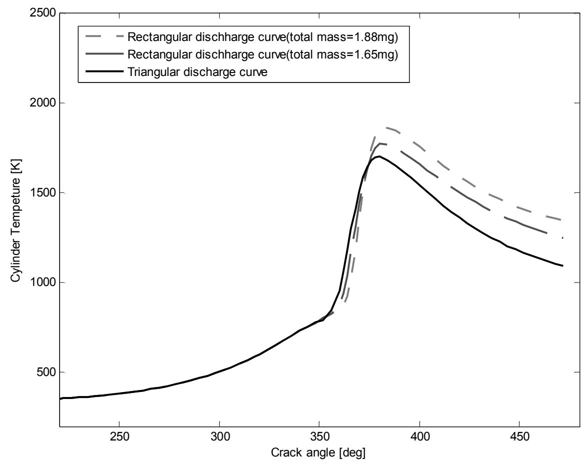

Figure 8: Mean cylinder temperature in 1000bar fuel injection pressure

In Table 5, the amount of peak pressure and temperature for semi-triangular discharge curve and also rectangular discharge curve with 1000bar injection pressure are observed.

Table5. The amount of peak pressure and temperature for 1000bar fuel injection pressure

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

The NO and Soot emission are shown in Figure 9 and 10. The main parameters affecting on NO emission are the cylinder temperature, air concentration and the time required for the reaction [21].

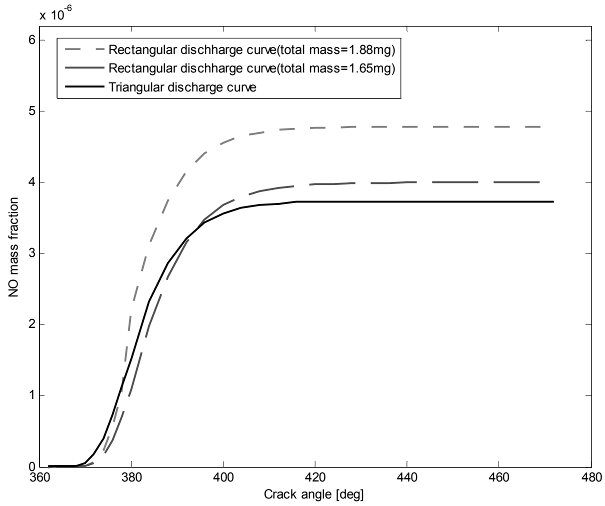

Figure 9: NO mass fraction in 1000bar fuel injection pressure

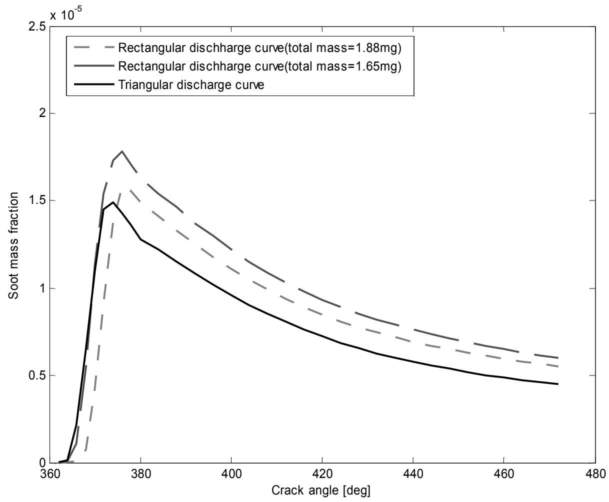

Figure 10: Soot mass fraction in 1000bar fuel injection pressure

Increase of the cylinder temperature about 150^0 K 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%.

6-2) Effect of injection discharge curve and 2000 bar injection pressure on upgrading power

To achieve the best condition of upgrading power, combustion process has been simulated with 2000bar injection pressure, too. Output power of combustion process in this simulation upgrades about 22%. The difference of output power in this section is shown in Figure 11.

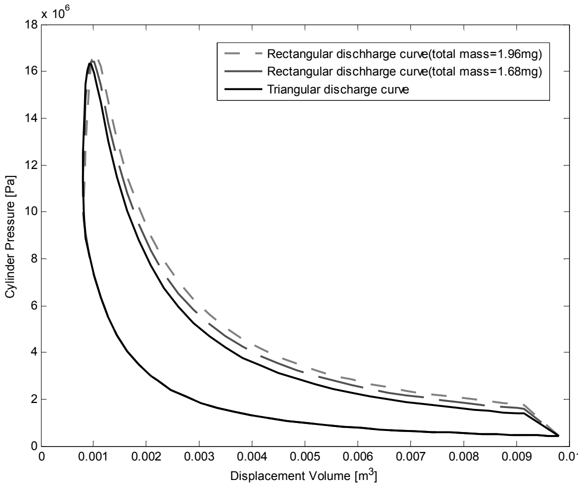

Figure 11: Cylinder pressure versus volume in 2000 bar fuel injection pressure

The cylinder pressure and temperature are shown in Figures 12 and 13, respectively. With increase of injection pressure, peak temperature increases but peak pressure has no special change. The amount of cylinder pressure and temperature for semi-triangular discharge curve and rectangular discharge curve with 2000bar injection pressure are observed in Table 6.

Table 6: The amount of peak pressure and temperature for 2000bar fuel injection pressure

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

The NO and soot mass fraction are shown in Figures 14 and 15, respectively. The total mass of 196mg and with nozzle diameter of 2.6mm which provide the best condition for output power, the NO emission and soot emissions increase by 36% and 20%, respectively

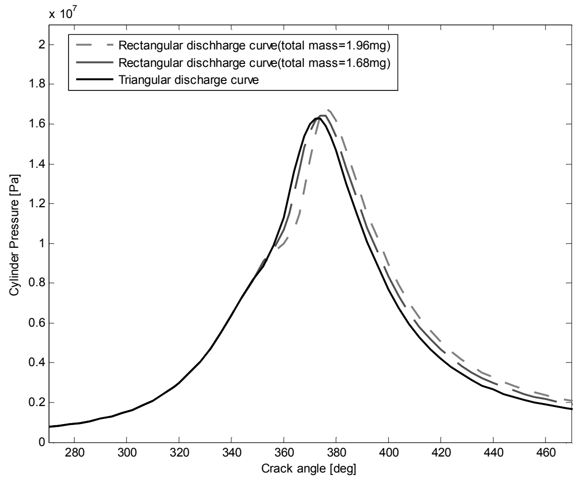

Figure 12: Mean cylinder pressure in 2000 bar fuel injection pressure

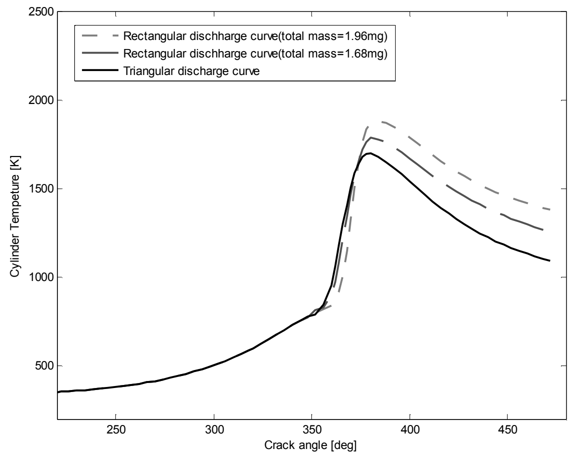

Figure 13: Mean cylinder temperature in 2000bar fuel injection pressure

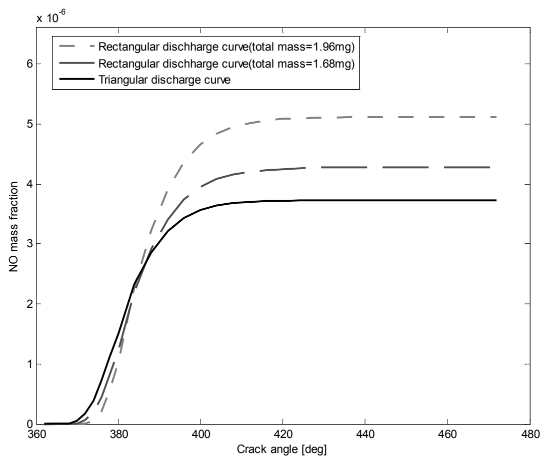

Figure 14: NO mass fraction in 2000bar fuel injection pressure

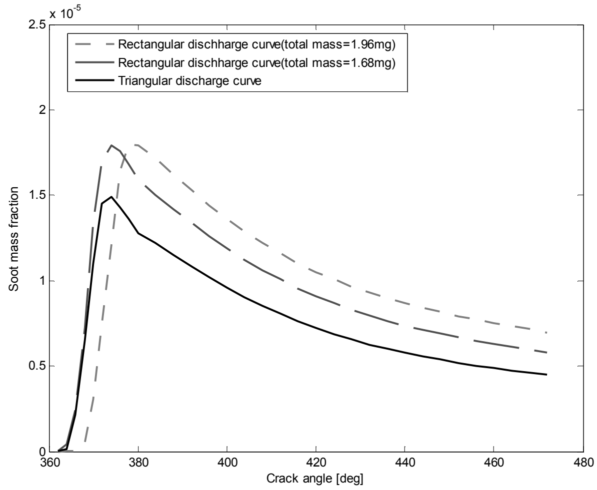

Figure 15: Soot mass fraction in 2000bar fuel injection pressure

7) 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 power are 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.

7.1) Rectangular fuel injection discharge curve and 1000 bar injection pressure

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

7.2) Rectangular fuel injection discharge curve and 2000 bar injection pressure

With 196mg total mass and 2.6 mm nozzle hole diameter, upgrading power is about 22%. NO and Soot emissions increase by 36% and 20%, respectively.
With 168mg 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.
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.

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بررسی تأثیر نیمرخ پاشش و فشار پاشش سوخت بر توان و متغیرهای احتراق موتور دیزل سنگین با شبیه‌سازی دینامیک سیالات محاسباتی

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چکیده

در این مقاله تأثیر نیمرخ پاشش سوخت به همراه فشار پاشش سوخت بر ارتقاء توان موتور دیزل سنگین با شبیه‌سازی فرایند احتراق در نرم افزار فایر مورد بررسی قرار گرفته است. برای این منظور، نیمرخ پاشش سوخت از حالت شبه مثلثی به حالت مستطیلی که در سامانهٔ چند راهه مشترک معمول است، تغییر کرده و فشار پاشش سوخت نیز از طریق یک معادلهٔ تجربی با توجه به میزان سوخت پاشش شده و قطر نازل تغییر داده شده است.

ارتقاء توان برای دو فشار پاشش ۱۰۰۰بار و ۲۰۰۰ بار مورد بررسی قرار گرفته است. برای حالت ۱۰۰۰بار که میزان سوخت پاشش شده ۱۸۸ میلی‌گرم و قطر نازل ۳ میلی متر می‌باشد، نسبت به حالت قبل که نیمرخ پاشش سوخت شبه مستطیلی با میزان پاشش سوخت ۱۳۹ میلی گرم و قطر افشانه ۳ میلی متر بوده، ارتقاء توان مشاهده شده در حدود ۱۹٪ می‌باشد درحالیکه فشار بیشینهٔ تغییر قابل توجهی نداشته است. از طرفی آلایندهٔ نیتروژن منواکسید و دوده به ترتیب ۳۰٪ و ۶٪ افزایش داشته‌اند. برای حالت ۲۰۰۰بار، میزان سوخت پاشش شده ۱۹۶ میلیگرم و قطر نازل ۲٫۶ میلی متر، ارتقاء توان ۲۲٪ بهبود یافته درحالیکه فشار بیشینهٔ ثابت نگه داشته شده است. آلایندهٔ نیتروژن منواکسید و دوده نیز به ترتیب ۳۶٪ و ۲۰٪ افزایش یافته‌اند.

اطلاعات مقاله

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