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ORGINAL ARTICLE

Effects of injection timing, before and after top dead center on the propulsion and power in a diesel engine

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Abstract It is well known that injection strategies including the injection timing and pressure play the most important role in determining engine performance, especially in pollutant emissions. However, the injection timing and pressure quantitatively affect the performance of diesel engine with a turbo charger are not well understood. In this paper, the fire computational fluid dynamics (CFD) code with an improved spray model has been used to simulate the spray and combustion processes of diesel with early and late injection timings and six different injection pressure (from 275 bar to 1000 bar). It has been concluded that the use of early injection provides lower soot and higher NO_x emissions than the late injection. In this study, it has been tried using the change of fuel injection time at these two next steps: before top dead center (BTDC) and after top dead center (ATDC) in order to achieving optimum emission and power in a specific point.

1. Introduction

The NO_x is produced at a great extent, due to the high local temperatures found in Diesel engines which are highly dependent on the initial rise of heat release. In addition, soot production and oxidation are both dependent on the mixing rate and local flame temperatures [1]. The injection velocity is one of the most influent parameters on the factors (which are mentioned before), since it controls both the mixing

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Nomenclature

CI	compression ignition
RHR	rate of heat release
PM	particulate matter
BSFC	brake specific fuel consumption
ISFC	indicated specific fuel consumption
BDC	bottom dead center
CA	crank angle (unit: degree)
DI	direct injection
TDC	top dead center

process and the rate of heat release. This is the reason that injection system parameters and nozzle geometry have been extensively studied due to their direct relation with the fuel injection rate and fuel velocity. To support this, it has been recognized that the characteristics of the injection system are the most important factors in influencing emissions and performance of CI engines [2–4]. Among the different possibilities in a fuel injection system, such as level and control of injection pressure, injection rate, and timing control [5], rate shaping mechanisms which produce boot-type injections have been recognized as the most effective ways to reduce NO_x without a huge detriment in PM and fuel consumption [2]. This method for controlling the rate of fuel injection is based on injecting small quantity of fuel at the beginning of the injection, in order to limit the initial rate of heat release (RHR), and consequently to keep the NO_x production at minimum. At the time of injection, the injection pressure is increased to enhance mixing and reduce soot emissions [6]. Wickman et al, explored the affect of high injection pressure and rate shape, concluded that falling rate-type injections had the potential to reduce soot formation [7]. Sugiyama et al studied numerical high-pressure fuel spraying in two cases, 150 MPa and 50 MPa. The results show that when the spray pressure is increased, maximum pressure inside the combustion chamber grew up too and the time of reaching to the maximum pressure is also faster than before [8]. AbuBakr studied the effects of high pressure diesel fuel spraying in a direct spray diesel engine. In this study the range of spraying pressure changes from 180 kPa to 220 kPa. Based on their results the best performance of spray pressure was obtained 220 bar and the minimum condition on fuel consumption and fixed spraying pressure was 200 bar [9]. Patterson et al. used the KIVA-II to reform after studying spraying schedule, pressure, and multi-level spraying so that the soot was reduced [10]. The common way to control NO_x has been using the turbo-charging with inter-cooling and retarding the fuel injection. Retarding increases BSFC and deteriorates fuel economy [11–13]. The most important engine parameters for NO_x reduction are ignition delay, injection timing, inter-cooling, combustion chamber design, injection rate, and compression ratio [14].

In recent years, computer codes have been used for simulating three-dimensional combustion in internal combustion engines [15–23]. This paper studies the theoretical effects of fuel injection timing on performance and pollutants in the four-cylinder direct injection diesel engine equipped with a turbocharger in the form of a numerical simulation by CFD code.

2. Governing equations

Governing equations including continuity, momentum and energy are modified based on Reynolds average and according to Reynolds-averaged Navier-Stokes (RANS) equations based on semi-implicit method for pressure-linked equations (SIMPLE) algorithm and k-ε standard turbulence model for numerical simulation of flow, inside the combustion chamber is used [15,23].

2.1. Spray and combustion models

The standard WAVE model, described in [24] was used for the primary and secondary atomization modeling of the resulting droplets. In this model the growth of an initial perturbation on a liquid surface is linked to its wave length and other physical and dynamic parameters of the injected fuel and the domain fluid. Drop parcels are injected with characteristic size equal to the nozzle exit diameter (blob injection). The injection rate profile is rectangular type and consists of four injection schemes, i.e. single injection and three split injection cases. The Dukowicz model was applied for treating the heat-up and evaporation of the droplets, which is described in [25,26]. This model assumes a uniform droplet temperature. In addition, the rate of droplet temperature change is determined by the heat balance, which states that the heat convection from the gas to the droplet either heats up the droplet or supplies heat for vaporization. A stochastic dispersion model was employed to take the effect of interaction between the particles and the turbulent eddies into account by adding a fluctuating velocity to the mean gas velocity [24]. This model assumes that the fluctuating velocity has a randomly Gaussian distribution. The spray wall interaction model used in the simulations was based on the spray-wall impingement model described in [27]. This model assumes that a droplet, which hits the wall is affected by rebound or reflection based on the Weber number. The Shell auto-ignition model was used for modeling of the auto ignition [25,28,29]. In this generic mechanism, 6 generic species for hydrocarbon fuel, oxidizer, total radical pool, branching agent, intermediate species and products were involved. In addition the important stages of auto ignition such as initiation, propagation, branching and termination were presented by generalized reactions, described in [25,28,29]. The eddy break-up model (EBU) based on the turbulent mixing was used for modeling of the combustion in the bowl [25]. This model assumes that in premixed turbulent flames, the reactants (fuel and oxygen) are contained in the same eddies and are separated from eddies containing hot combustion products. The rate of dissipation of these eddies determines the rate of combustion.

In other words, chemistry occurs fast and the combustion is mixing controlled.

2.2. Emission models

The Zeldovitch mechanism was used for prediction of NO_x formation [25]. The soot formation rate is described as a model which is based on the difference between soot formation and soot oxidation [30].

2.3. Profile engine

The engine under study is a commercial DI, water cooled four cylinders, in-line, turbocharged aspirated with inter-cooler diesel engine whose major specifications are shown in Table 1 [28,29]. Figure 1 shows details of the computational grid and how the spray injection.

Table 1 MT4.244 engine specification.

Bore	100
Stroke	127.0 mm
Displacement	3.99 liters
Combustion chamber	Reentrant
Compression ratio	17.5
Number of valves/cylinder	2/4
Injection type	Direct injection
Fuel injection pump	DPA
Injection pressure (typical)	450 bar
Fuel injection nozzle	5 hole
Maximum power output	61.5 kW @ 2000 rpm
Maximum torque output	340 N · m @ 1400 rpm
Kind of aspirated	Turbocharged & intercooled

2.4. Boundary conditions and initial conditions

The pistons were considered as a moving wall. Cylinder head temperature, cylinder wall and piston surface are based on experimental data. Cylinder head temperature was 550 K, cylinder wall temperature 400 K and pistons surface 590 K.

3. Results and discussion

Injection pressure is changed from 275 to 1000 bar. Comparison of cylinder pressure changes according to crank angle at 14308, 18604, and 12862 cells is represented in Figure 2. As shown, the differences between three diagrams are small.

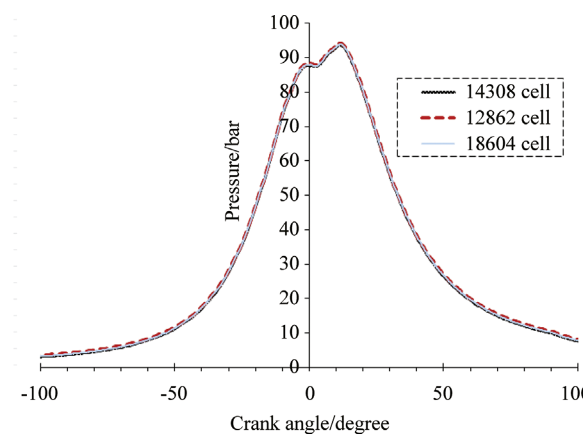

Figure 2 Grid independence compared to the pressure curves according to crankshaft angle.

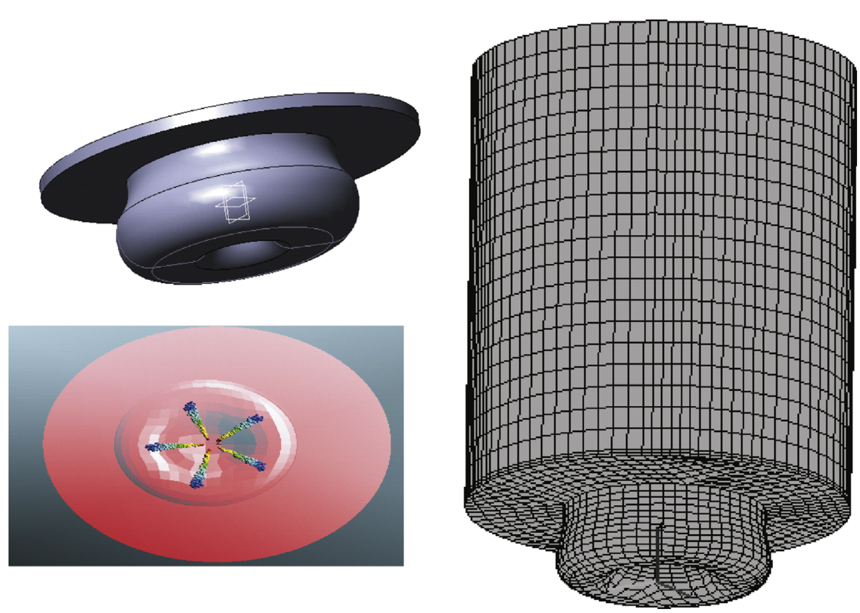

Figure 1 Details of the computational grid and how to spray.

3.1. Simulation results of motor function and credit rating results

Figure 3 represents that the experimental data and numerical solutions are matching well [15,23].

The above diagram shows about 3% error between numerical and experimental pressure. The value of indicatory power becomes equal to 72.87 kW which has 1% difference with obtained 71.6 kW experimental, and Figure 4 shows the diagram of producing nitrogen oxide and diagram of producing soot based on the crankshaft angle for diesel engine [15,23].

In Table 2, the amounts of pollutants are compared with the numerical results in which a good match is observed,

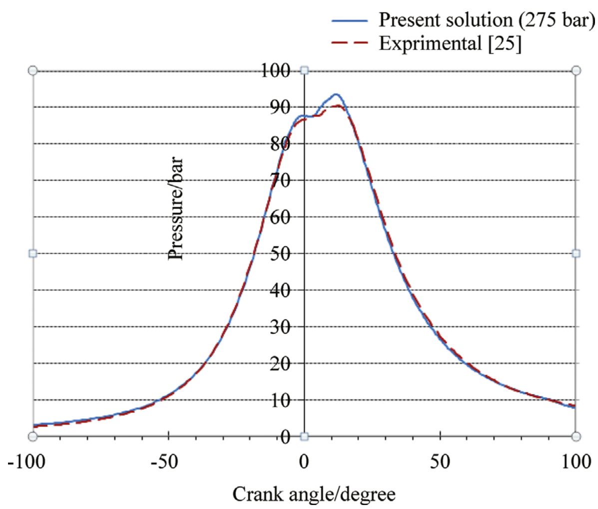

Figure 3 Comparison of pressure versus crank angle inside combustion chamber, experimental and numerical cases.

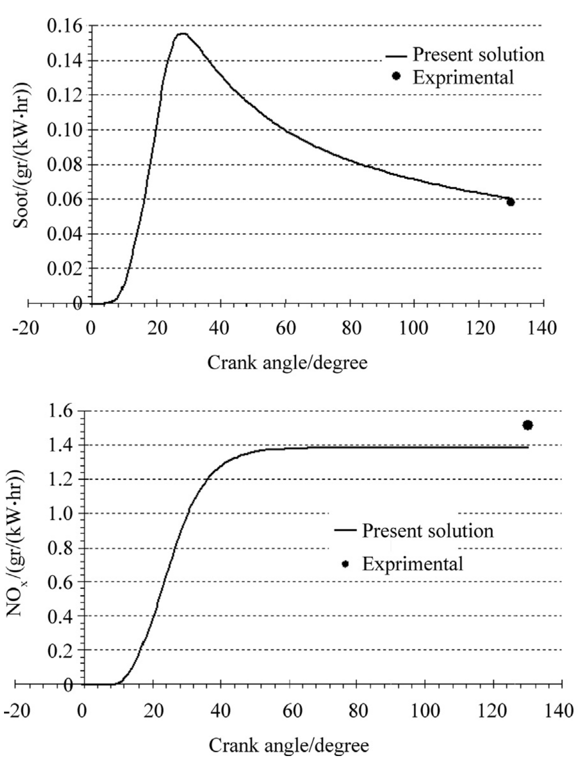

Figure 4 Comparisons of formation of NO_x and soot according to the diesel engine versus crank angle.

3.2. Effect of injection pressure on performance and engine pollutants output

In this paper the results of injection pressure impact on engine performance parameters and engine pollutants output with constant amount of consumable fuel per cycle are checked. In Figure 5, with increasing injection pressure, maximum value of pressure is increased inside the cylinder. Increasing of spraying pressure raises the fuel speed and air mixture formation in delay phase of combustion. This provides more mixture for pre-mixing phase.

Figure 6 shows energy release in each crankshaft degree for different spraying pressures. As can be seen, by increasing the injection pressure, ignition delay reduces. This means that by increasing the injection pressure, speed of droplets increases. Due to the constant fuel consumption and increase of spraying pressure, sprayed fuel will sprinkled in shorter time and that will cause shorter length of forced combustion, hence is increased the pre-mixed combustion zone. Figure 7 is shown indicated power of engine and indicatory special fuel consumption.

Table 2 Comparison of experimental and simulation results for the base engine.

	Experimental value	Numeric value	Percentage of lapse /%
NO_x/(gr/(kW · hr))	1.15	1.38	8
Soot/(gr/(kW · hr))	0.057	0.01	4.3
Power output per cycle	72.87	71.6	1

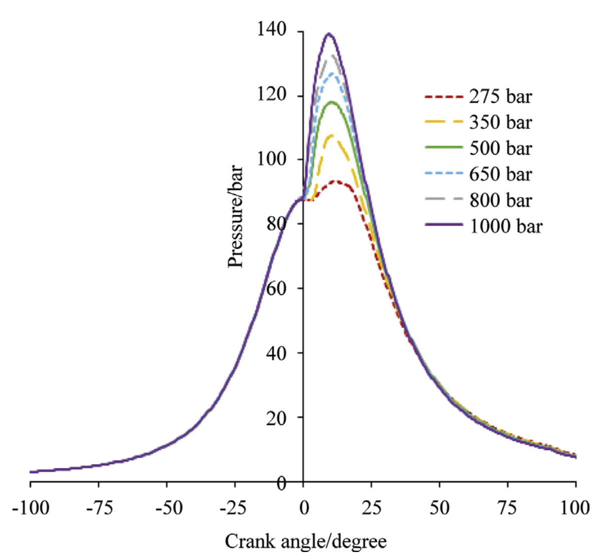

Figure 5 Pressure variations versus crankshaft angle at different spraying pressures.

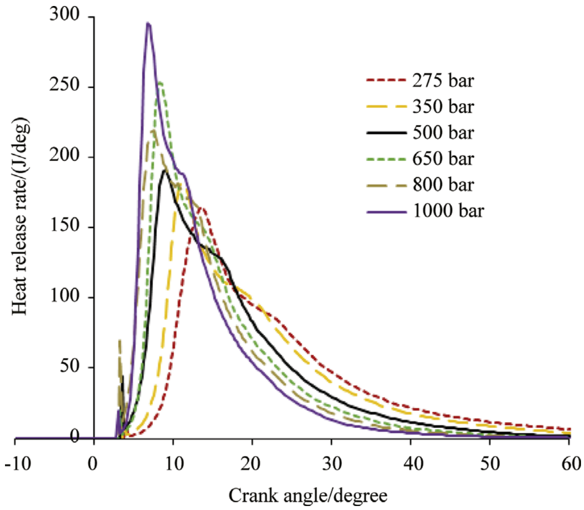

Figure 6 Energy release versus crankshaft angle at various spraying pressures.

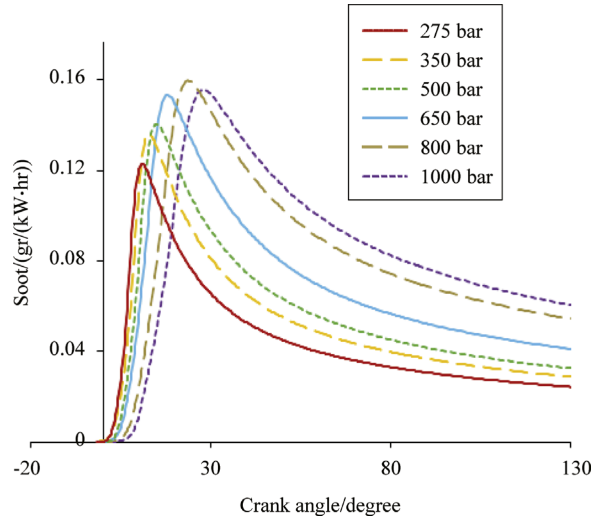

Figure 9 Effect of pressure increase on soot.

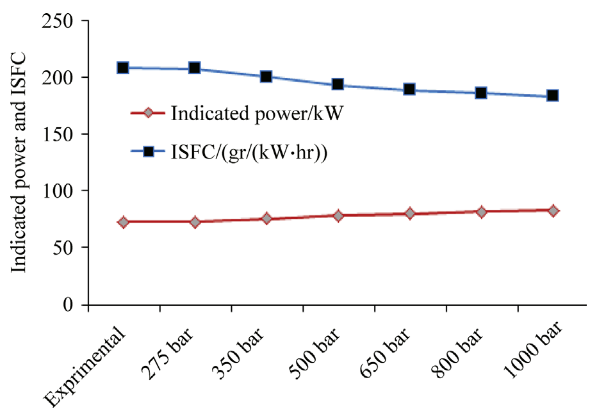

Figure 7 Diagrams of indicated power and ISFC.

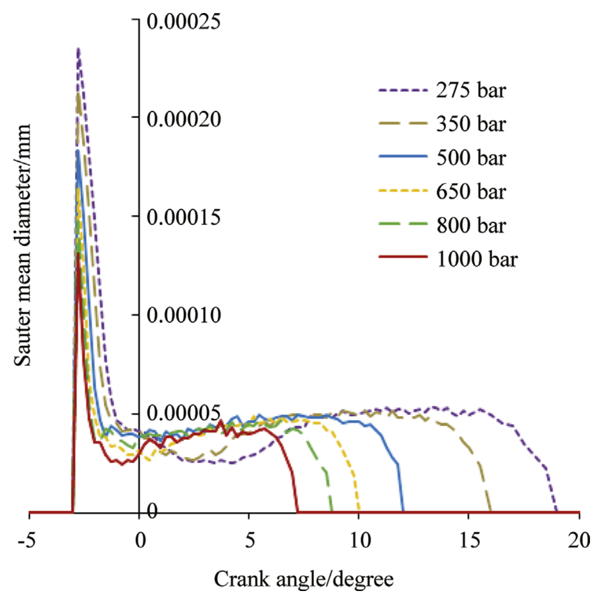

Figure 10 Impact of the speed of fuel injection on average fuel drop diameter.

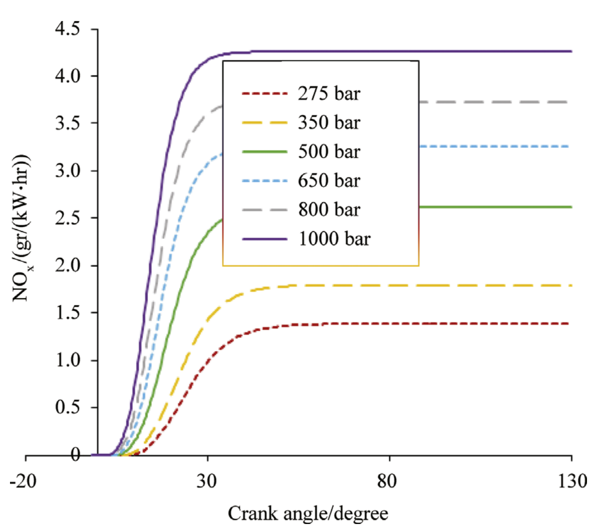

Figure 8 Effect of increased injection pressure on NO_x.

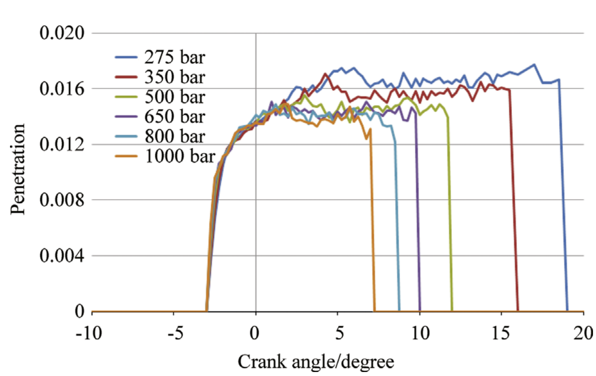

Figure 11 Impact of different injection pressure on jet penetration.

With increasing injection pressure, indicated power of engine is increased about 12% and indicated specific fuel consumption (ISFC) of engine is decreased. This means that in return, for the same fuel consumption, engine power output is increased that cause combustion process better. Figure 8 shows effect of fuel injection pressure on NO_x pollution per each crankshaft angle. From Figure 12 can be seen that with increasing injection pressure, NO_x increases too. By increasing injection pressure, fuel particles became smaller and in fact atomization of fuel will get better and area of pre-mixing is caused by faster. Figure 9 shows that by increasing the injection pressure from 275 to 1000 bar, soot decreases about 58%.

Figure 10 represents the effect of fuel injection pressure on the average diameter of droplets and Figure 11 shows jet penetration.

By increasing the injection pressure, due to the constant fuel consumption per cycle, length of spraying had also reduced. For each injection pressure, the process of average

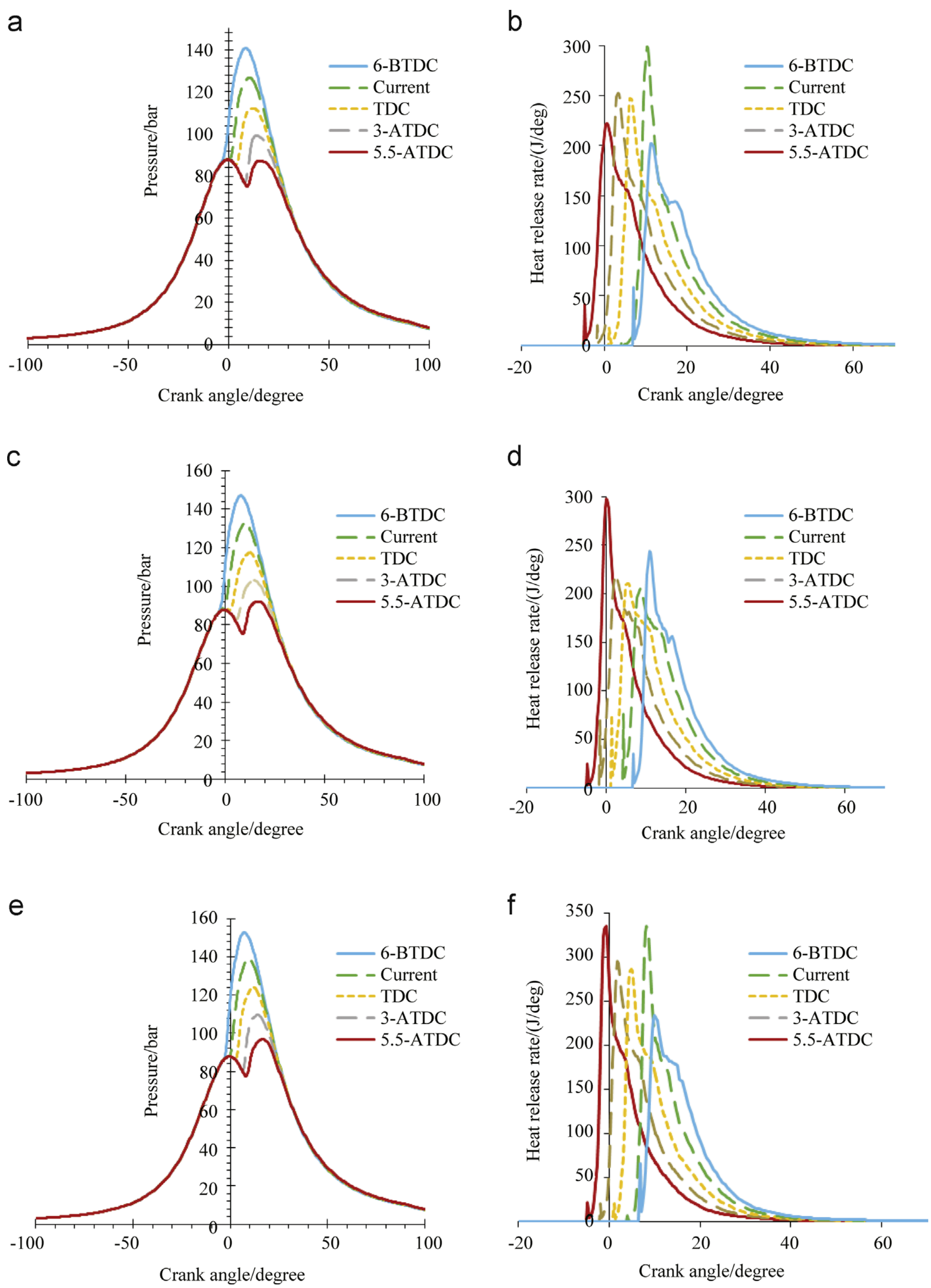

Figure 12 Pressure and heat release versus crank angle. (a) Pressure cylinder at injection pressure of 650 bar, (b) heat release rate at injection pressure of 650 bar, (c) pressure cylinder at injection pressure of 800 bar, (d) heat release rate at injection pressure of 800 bar, (e) pressure cylinder at injection pressure of 1000 bar, and (f) heat release rate at injection pressure of 1000 bar.

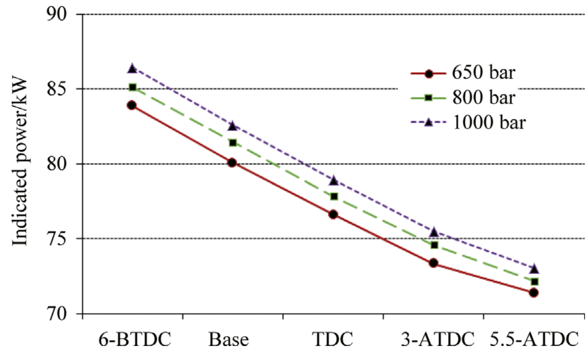

Figure 13 Effect of times and pressures injections on the engine power.

diameter changing is as follows: in primary degrees during the expansion course due to the high pressure of pre-mixed combustion the diameter of droplets decreases and with the reduction of pressure in cylinder chamber due to the increased mass, the average of droplets diameter increases.

3.3. Effect of time injection on performance and pollutants

Since the fuel spray is 3°BTDC to determine the pressure and spraying conditions compared with the case with current position each spraying pressures studied in five

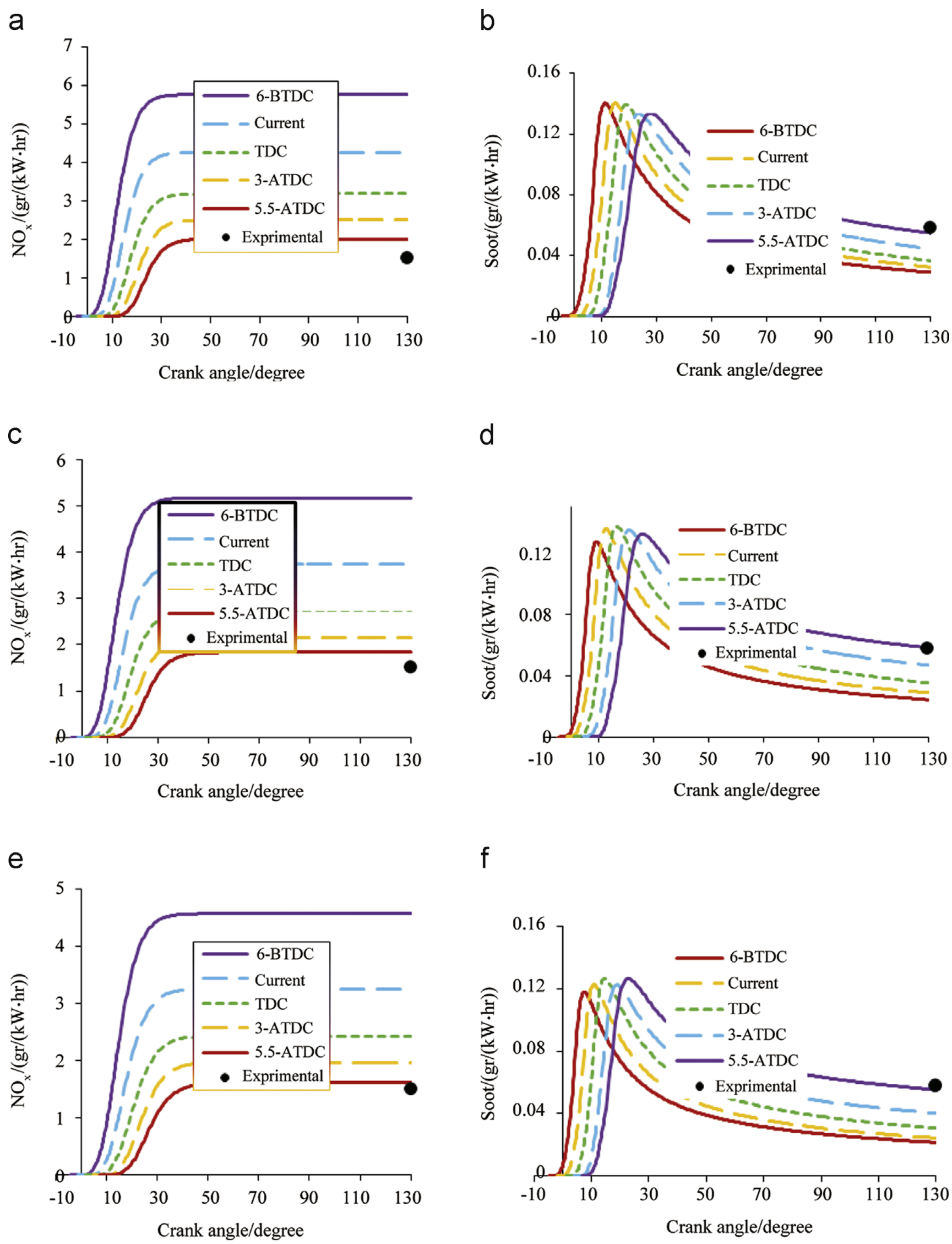

Figure 14 NO_x and Soot output for different pressures and times injection versus crank angle. (a) The NO_x at 650 bar, (b) the soot at 650 bar, (c) the NO_x at 800 bar, (d) the soot at 800 bar, (e) the NO_x at 1000 bar, and (f) the soot at 1000 bar.

Table 3 Percentage difference between the numerical results with experimental data.

	NO_x/%			Soot/%		
	650 bar	800 bar	1000 bar	650 bar	800 bar	1000 bar
6-BTDC	+202	+241	+280	−50	−58	−63
Current	+115	+146	+180	−44	−50	−58
TDC	+60	+81	+110	−36	−39	−47
3-ATDC	+30	+41	+65	−23	−27	−30
5.5-ATDC	+6	+21	+33	−5	0.0	−4

state fuel spraying time. The NO_x shows this matter that all sprayings will be put forward to 3°CA. The goal is keeping the fuel consumption at the high spray pressures and reducing the pollution. Figure 12 shows the performance of spraying at different crank angles. The cylinder pressure increases and this can cause the fuel and air to be mixed quickly. The first peek pressure shows the state of motoring (with no combustion) and the second peak is related to the combustion. Energy release grows more quickly because the fast formation of mixture. All spraying time before ignition effect has more influence than combustion.

Figure 13 shows effect of fuel injection pressure at the time of spraying various indicators on the engine power. As shown dropping the preset time fuel injection engine indicators on average about 4.6 percent compared with the current mode will increase the this story is set in the cylinder pressure diagrams also observed because the area under graph cylinder pressure also indicates this. It also can be seen that in all cases, increasing spray pressure indicators engine power has increased.

Figure 14 shows NO_x and soot for pressure spraying variations. Before spraying started, the maximum cylinder pressure had increased, so the temperature inside the cylinder increases (no NO_x).

Compression of numerical results of different spraying pressures in different spraying times of the primary mode engine experimental result in Table 3, the percentage is shown.

At 3°CA before starting spray, soot is around 50%, but instead NO_x increased. Postponing the injection time at the time of spraying in the areas of TDC and 3°ATDC is acceptable.

4. Conclusions

At this present work, the effects of time injection on combustion and pollution of a DI diesel engine have been investigated with using multi-dimensional CFD code AVL-FIRE. The calculations were based on the described conditions. It was concluded that:

- A good conformity is between predicted in cylinder pressure and exhaust NO_x and Soot emissions with the experimental data that can be observed.
- The calculated results of the cylinder pressure and heat release rate for the optimum time and pressure injection cases show very good similarity with the numerical results that obtained by phenomenological combustion models.
- Increasing the fuel injection pressure in the combustion chamber due to better mixing of fuel and air so that the rate of heatrelease energy and peak pressure increases.
- The injection pressure variation is reviewed and the change of droplet diameter is studied. By increasing the spray pressure, diameter of fuel droplets has been decrease, so small diameters cause faster atomization and evaporation.
- Increasing the spraying pressure generate the output work per cycle. Growing the spraying pressure will produce faster mixture formation, hence combustion delay time was reduced and the maximum energy release was also gone up. By Increasing the spraying pressure from 275 bar to 1000 bar, the soot pollution will experience up to 58% and power up to12% grow.
- With this change, efficiency of the engine will increase around 4% but for every drop 3°CA before spraying time the increase of efficiency will be 4% compared with main spraying time of the engine. If starting spray temperature is around 5.5°ATDC, temperature inside the combustion chamber will increase therefore the soot was reduced while indicated power and NO_x were increased small amounts.

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