Investigation of intake pressure and fuel injection timing effect on performance characteristics of diesel engine

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Abstract. The low fuel consumption and efficient power usage in diesel engine has made diesel engine as the more appealing option in comparison with other type of engines. To raise the performance of DI (direct injection) diesel engine further, injection timing and intake pressure variation were studied under 2000 RPM (rotation per minute) engine rotation. This experiment used DI diesel engine with single cylinder. The length of stroke was set to 96.9 mm, the diameter of bore was set to 85 mm and the compression ratio of the engine was 16.3. The variations of main injection timing were set for 1 ° after TDC (top dead centre) as advanced injection timing, and 3 ° after TDC as retarded injection timing. Boost pressures for intake pressure were varied with 20 in increments and started from 0 KPa to 60 KPa. In-cylinder pressure characteristics and heat release rate were used to evaluate the engine performance. The experiment indicated, as the boost pressure raises, the heat release rate and in-cylinder pressure are increased. The main injection timing advancement from 3 ° after TDC to 1 ° after TDC causes increase to the peak of in-cylinder pressure after TDC in DI diesel engine. This phenomenon is due to the slower combustion in retarded injection timing. For heat release rate, the advancement of injection timing causes the differences between various intake pressures to be more apparent.

1. Introduction

Compression ignition (CI) engine, due to their excellent fuel efficiency and durability, has become the popular power plant for automotive application. This is globally the most accepted type of internal combustion engine used for powering agricultural implements, industrial applications, and construction equipment along with marine propulsion. However, emissions from diesel engines have been focused in increasingly stringent emission regimes because of their adverse health impact on humans. Diesel particulates are classified as ‘probable carcinogen’. Under tremendous pressure to comply with increasingly stringent emission norms adopted worldwide, mass emissions of particulate matter (PM) from diesel engines have been significantly reduced by automotive OEMs (original equipment manufacturers) by employing improved exhaust gas after-treatment technologies [1]. In diesel engines, it is rather difficult to lower NOx and PM emissions simultaneously due to soot-NOx trade-off. High NOx and PM emissions are still the main obstacle in the development of next generation conventional diesel engines.

Combustion, performance and emission characteristics of diesel engines depend on several factors including fuel injection timing (FIP), start of injection (SOI), fuel quantity injected, number of injections (post- and pilot-), design of combustion chamber and nozzle spray patterns. High-pressure direct injection (HPDI) seems to be one of the most efficient ways to comply with the stringent global emission norms. FIP for different generation of diesel engines varies from 200 to 2000 bars. Kato et al. [2] demonstrated using high fuel injection pressures as a means to reduce PM emissions without increasing NOx emissions. High FIP seem to induce a very different spray structure than low FIP sprays used earlier [3]. This is mainly due to cavitation created in the nozzles at high FIP, which results in significantly faster atomization [4].

Other studies [5,6] showed that higher FIP improve fuel–air mixing, followed by faster combustion, which directly influences pollutant formation. Diesel spray characterization is usually done for parameters such as spray tip penetration, spray angle, droplet velocities, droplet sizes and distributions, and global spray structure. A good understanding of these characteristics is essential for increasing the combustion efficiency and reducing the environmental impact. High pressure difference across the injector nozzle is necessary to atomize the liquid fuel into small droplets in order to enable rapid vaporization as well for high jet penetration in the combustion chamber [7,8]. Droplets size distribution of a spray fundamentally affects CI engine combustion. Smaller fuel droplets vaporize rather quickly compared to larger droplets however their penetration is shorter therefore the size distribution needs to be optimized. Chen et al. reported that small droplets and high penetration depth of fuel jet enhances the fuel–air mixture quality, which provides shorter ignition delays and more complete combustion [8,9]. Lower FIP gives larger droplet diameters, and thus increasing ignition delay during combustion [9]. This also leads to higher cylinder pressures, which ultimately results in higher NOx emissions. When FIP increases, spray droplet diameter distribution reduces. This leads to improved fuel–air mixture formation because of superior mixing during ignition delay, therefore smoke and CO emission are reduced [10]. However, if FIP is too high, ignition delay period becomes too short. Hence, possibility of homogeneous mixing decreases and as a result, combustion efficiency is reduced [11]. Bruneaux [12] investigated spray characteristics of common rail direct injection (CRDI) system in a high pressure, high temperature cell, which created conditions existing in a typical diesel engine. An increase of FIP was found to enhance the fuel atomization at the nozzle outlet, resulting in more distributed vapor phase, which improves mixing. Hence the fuel injection strategy is an important parameter in diesel engines to optimize the combustion, performance and tailpipe emissions.

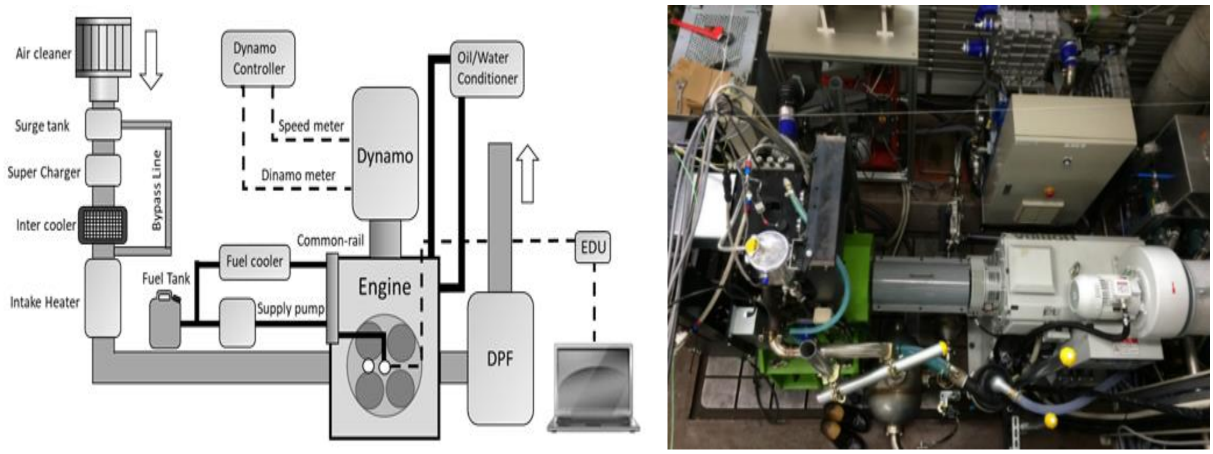
These injection parameters also affect the particulate emission from diesel engines. High compression ratios, along with relatively high oxygen concentration in diesel combustion chamber deliver excellent thermal efficiency, and low CO and HC emissions in contrast with a comparable gasoline engine [13]. However, mass of particulates emitted from diesel engines are generally 10–100 times higher than SI engines [14–16]. Particulates are of concern from engine performance, durability and harmful environmental impact. Higher particulate emissions result in reduced fuel economy because of fuel loss due to the incomplete combustion. Interaction of these particulates results in increased wear of the engine components. Agarwal et al. [16] carried out experiments to investigate the characteristics of particulates and concluded that lubricating oil contaminated by diesel soot is a key factor responsible for higher engine wear [17].

Particulates have adverse environment impacts such as they affect human and live-stock health, lead to poor visibility, and soil the buildings. While attempting engine optimization, it is required to consider particulate numbers as well with the particulate mass. Methods used for reducing the particulate mass such as increasing FIP, use of variable geometry turbochargers (VGT) and diesel particulate filters (DPF) tend to increase particulate numbers by reducing their size, which is likely to be more harmful for human health [18,19]. A serious study on the diesel particulate characterization is important because a significant proportion of diesel particulates have aerodynamic diameters less than 1 μm. Diesel particulates in this size range have a high probability of being inhaled and deposited in the respiratory tract, and potentially cause respiratory diseases and consequently damage the lungs [20,21]. Particles emitted from diesel engines can be completely characterized by gravimetric measurements, particle number size distribution, particle surface area-size distribution, particle volume-size

distribution, soluble organic fraction (SOF), elemental trace metals, elemental carbon (EC), organic carbon (OC), total carbon, and polycyclic aromatic hydrocarbons (PAHs) [22–26]. Agarwal et al. [16] reported an increase in particle number concentration at lower engine loads and particle number concentration reduction at higher engine loads with addition of 20% biodiesel to diesel.
In the present investigation, a flexible single cylinder research engine was used to experimentally evaluate the effect of fuel injection timings and FIP on combustion, emissions and performance. Mineral diesel was used as test fuel. This engine is capable of precisely controlling fuel injection parameters such as FIP, SOI and injected fuel quantity. The effect of variations in these parameters on engine combustion, performance and emission characteristics is evaluated. For particulate size and number distribution, engine exhaust particle sizer (EEPS) was used.

2. Experimental setup and procedure

In this experiment, diameter of the engine bore was set to 85 mm, the length of the stroke was set to 96.9 mm and the compression ratio of the engine was set to 16.3. The piston and head of the cylinder were manufactured using aluminum alloy while the dry liner in the cylinder is manufactured with cast iron as material. Diagram showing the schematic of the experiment is displayed in figure 1. The complete specification of the DI diesel engine with single cylinder utilized in this experiment is displayed in table 1. Kistler Japan Type 6052 pressure sensor was used to measure in-cylinder pressure in this experiment.



(a) (b)

Figure 1. Schematic of (a) the experimental system and (b) experimental system construction.

Signals that originate from coaxial heat flux meter with high-speed response and coaxial thermocouple were amplified and documented inside data logger simultaneously to obtain the heat release rate (HRR) data. The heat release rate data were obtained along with the signal from in-cylinder pressure sensor in every crank angle (CA) position.

Table 1. Specifications of the engine used in the experiment.

Parameter	Value / Description
Type of engine	Single cylinder engine
Length of connecting rod [mm]	150.46
Stroke [mm]	96.9
Bore [mm]	85
Capacity of the cylinder [cc]	550
Ratio of compression [-]	16.3
Duration of Intake valve opening [deg.]	From 347 to −120 (Comp. TDC is 0 deg.)
Duration of Exhaust valve opening [deg.]	From 122 to −330 (Comp. TDC is 0 deg.)
Offset of the cylinder [mm]	6.5
Offset of the piston-pin [mm]	0.8

Table 2. Experiment condition at 2000 rotation speed with boost pressure of 0 KPa, 20 KPa, 40 KPa, and 60 KPa.

Rotation speed [RPM]	Injection step	Injection timing [deg.]	Injection time[ms]	Injection quantity [g/st]	Total injection quantity[g/st]
2000	pilot	-25	0.16	0.0014	0.0398
	pre	-15	0.18	0.0024	
	main	3	0.83	0.0360	
	pilot	-25	0.16	0.0014	0.0398
	pre	-15	0.18	0.0024	
	main	1	0.83	0.0360	

The intake pressures in this experiment were varied from naturally aspirated (0 KPa) to 60 KPa with 20 KPa increment. The boost pressures originated from utilizing supercharger. Naturally aspirated condition was examined to understand further the outcome of supercharging in DI diesel engine. As for the main injection timing, the timings were varied into two, advanced injection timing located 1° after TDC and retarded injection timing located 3° after TDC. Details regarding injection timing, quantity, and time is provided in table 2. The evaluation of engine performance was conducted through examining both heat release rate and in-cylinder pressure of the engine.

3. Results and discussion

The results of injection timing and intake pressure effect on in-cylinder pressure are shown in figure 2. Figure 2 shows main injection step of 3° and 1° after TDC, with 2000 RPM engine speed and various intake pressures. This figure indicates the rise of in-cylinder pressure with the rise of intake pressure. The results are similar with investigation done by Lee et al. that propose the rise of maximum in-cylinder pressure under supercharged condition with engine speed below 3000 RPM [27].

Noticeable difference between main injection of 3° and 1° after TDC, as seen in figure 3, is found in the timing of maximum in-cylinder pressure. The maximum in-cylinder pressure during main injection step of 3° after TDC is located during TDC while the maximum in-cylinder pressure during main injection of 1° after TDC is located few degrees after TDC. This phenomenon occurs due to the slower combustion reaction with main injection timing of 3°. The slower combustion reaction reduces the rise of in-cylinder pressure value because of the fuel mainly burns after TDC. The lower value of maximum in-cylinder pressure in main injection timing of 3° after TDC causes the in-cylinder pressure during TDC to be higher than the aftermath in-cylinder pressure rises [28-30].

The results of injection timing and intake pressure effect on heat release rate are shown in figure 4. Figure 4 shows the main injection step of 1° and 3° after TDC with 2000 RPM engine speed and various intake pressures. This figure indicates the rise of heat release rate is proportional to the rise of intake pressure. The results are similar with investigation done by Lee et al. that propose the rise of maximum in-cylinder pressure under supercharged condition with engine speed below 3000 rpm [27].

Noticeable difference between main injection of 1° and 3° after TDC is found in the rise of heat release rate before TDC. Heat release rate raises before TDC in all level of intake pressures. In main injection timing of 3° after TDC, the differences of heat release rate value between various intake pressures are miniscule. Contrary to this, with main injection timing of 1° after TDC, the heat release rate value and timing differences are more apparent. Both injection timings have similar pattern with higher intake pressure having lower heat release rate before TDC.

In figure 5, the effect of injection step advancement from 3° and 1° to the heat release rate is displayed with respond to certain level of intake pressure. The maximum heat release rate between 3° and 1° are quite similar with small degree of differences. It is also found that the maximum heat release rate between 3° and 1° occurred in different time. Maximum heat release rate occurs earlier in 1° injection timing. Another point of interest is seen in the trend of heat release rate of both 3° and 1° injection timing. It is observed that the heat release rate of 3° overcomes the heat release rate of 1° after certain degree from TDC. These phenomena are caused by the slower combustion rate of 3° which delays the enormous increase of heat release rate to higher crank angle timing.

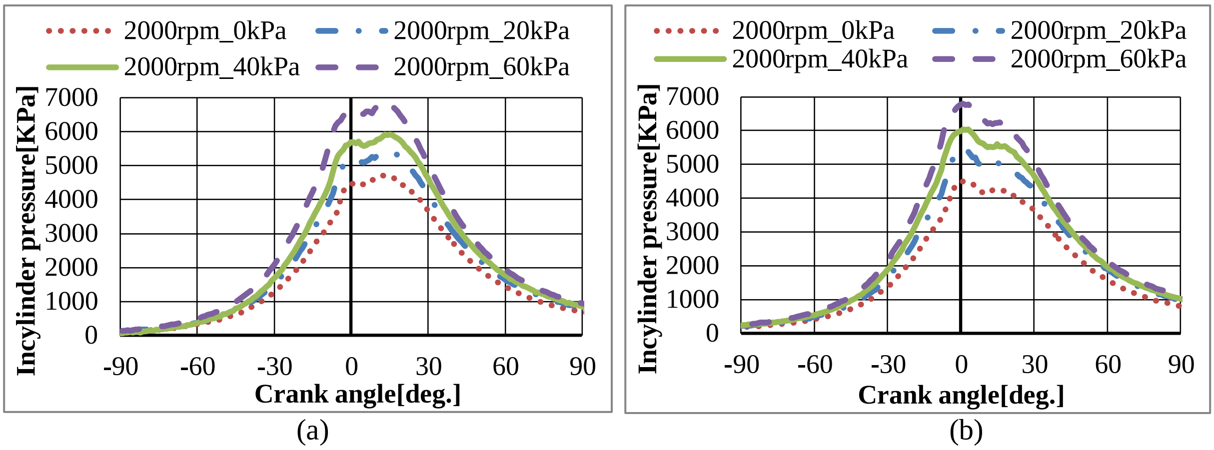


Figure 2. The in-cylinder pressure of DI diesel engine with injection timing of (a) 1° and (b) 3° after TDC under 2000 RPM and various intake pressures.

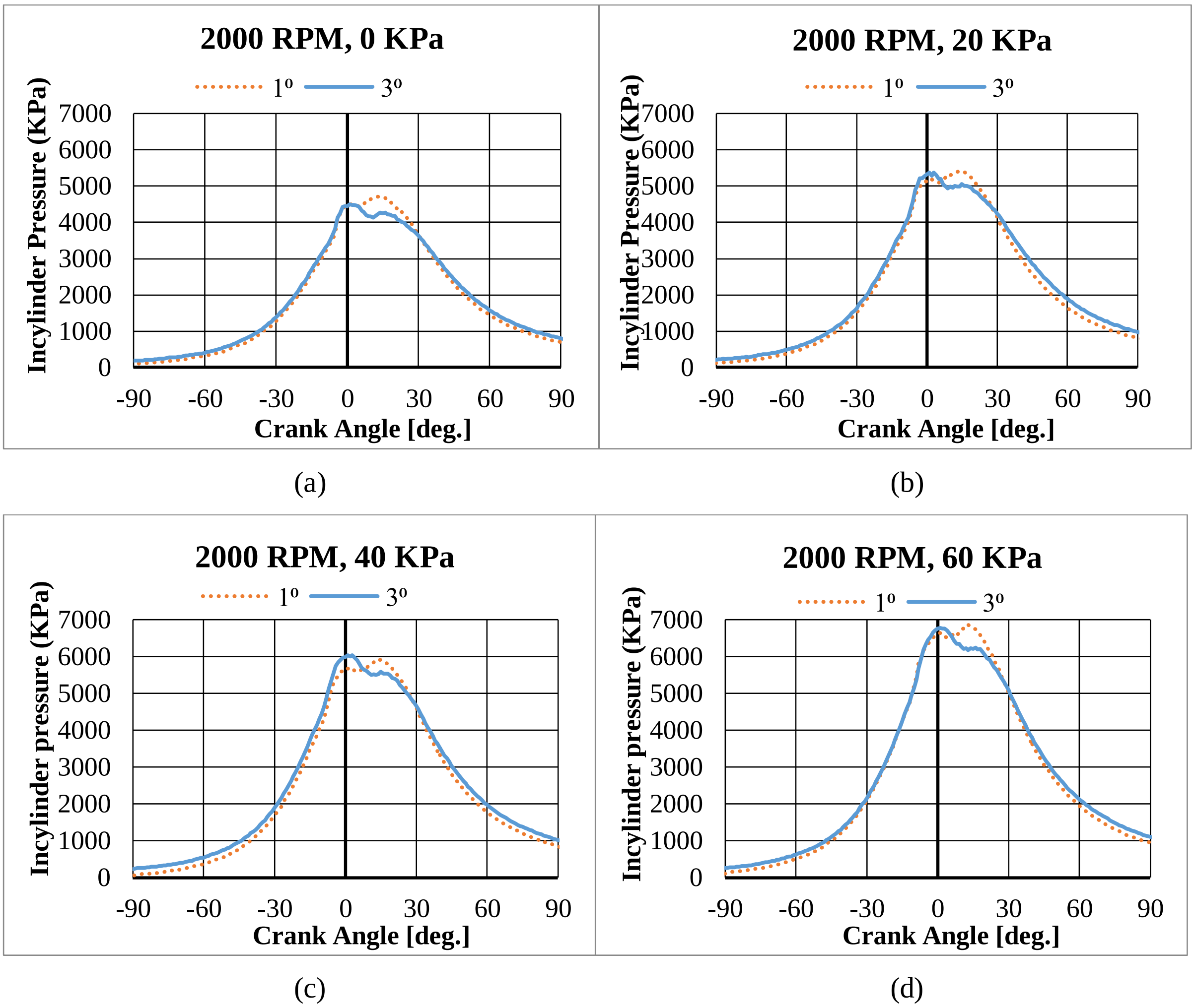


Figure 3. The in-cylinder pressure comparison of DI diesel engine with injection timing of 1° and 3° after TDC under 2000 RPM and various intake pressures (a) 0 KPa (b) 20 KPa (c) 40 KPa (d) 60 KPa.

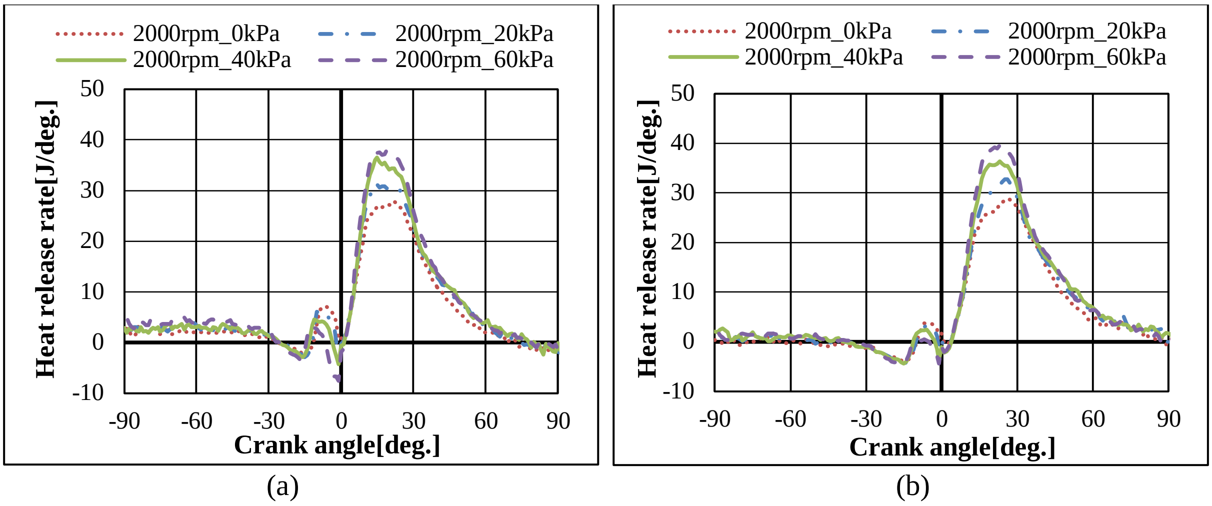


Figure 4. The heat release rate of DI diesel engine with injection timing of (a) 1° and (b) 3° after TDC under 2000 RPM and various intake pressures.

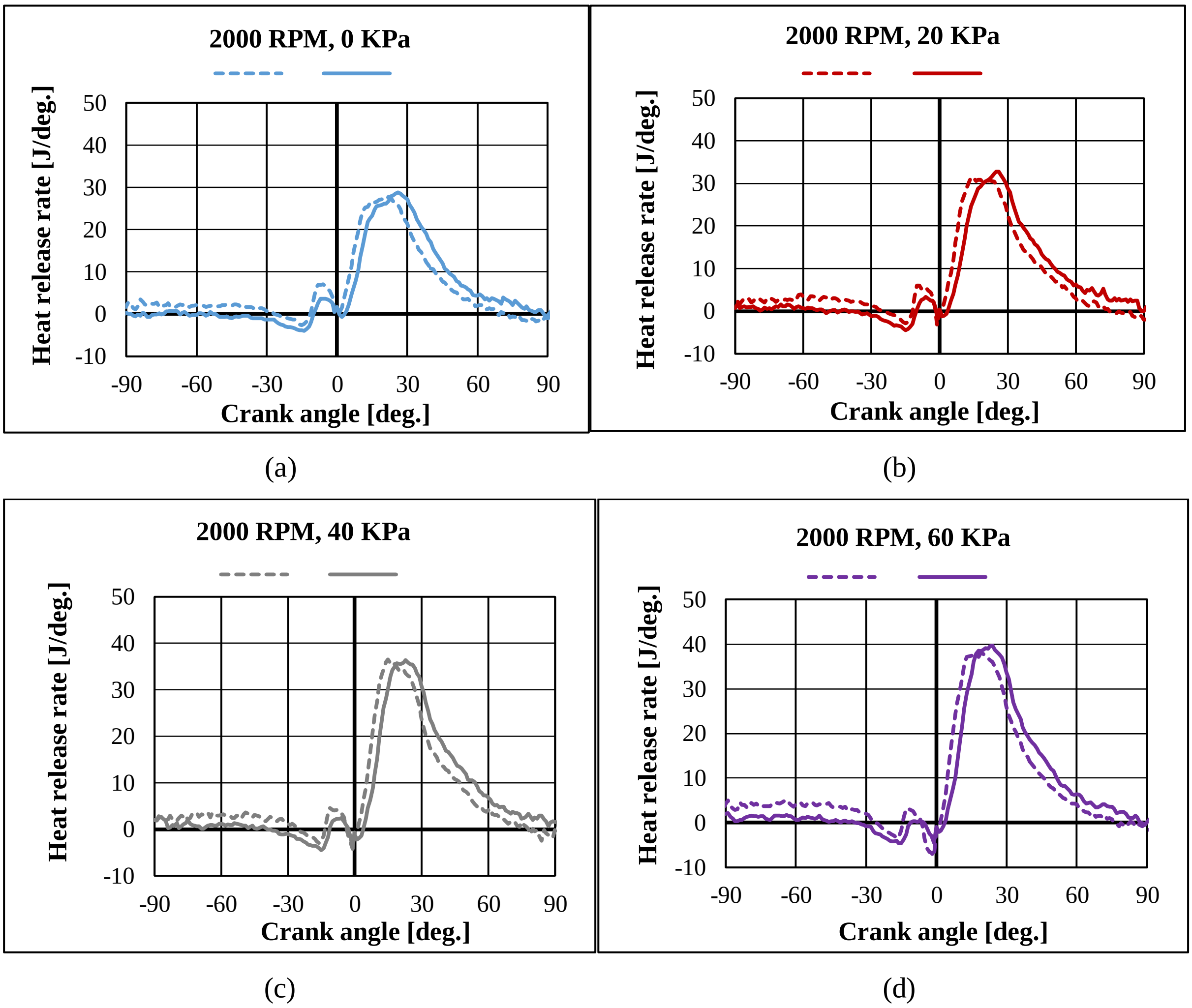


Figure 5. The heat release rate of DI diesel engine with injection timing of 1° and 3° after TDC under 2000 RPM and various intake pressures (a) 0 KPa (b) 20 KPa (c) 40 KPa (d) 60 KPa.

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4. Conclusion

From these investigations, it can be concluded that the injection timing and intake pressure highly affect the in-cylinder pressure and heat release rate performance characteristics in DI diesel engine. The rise of intake pressure from naturally aspirated to 60 KPa boosts the in-cylinder pressure and heat release rate of DI diesel engine. The main injection timing advancement from 3° to 1° after TDC causes increase to the rise of in-cylinder pressure after TDC in DI diesel engine. In retarded injection timing, the rise of in-cylinder pressure during TDC is higher compared to the rise of in-cylinder pressure after TDC. In the advanced injection timing, the rise of in-cylinder pressure during TDC is lower compared to the rise of in-cylinder pressure after TDC. For heat release rate, the occurrence of maximum heat release rate appears earlier in advanced injection timing compared to retarded injection timing. Both phenomena are due to the slower combustion in the retarded injection timing condition.

5. References

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