

EFFECT OF BOOST PRESSURE AND INJECTION STRATEGY TO THE IN-CYLINDER PRESSURE AND HEAT RELEASE RATE OF DIRECT INJECTION DIESEL ENGINE

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Abstract

An optimum diesel engine helps to solve the increasing energy demand, the depletion of fossil fuels, and the environmental problems from the utilization of combustion engines. To optimise the operation of a direct injection diesel engine, the effects of various boost pressures under different rotations and main injection timings were studied experimentally and numerically. The boost pressure was set between 0 kPa to 60 kPa with increment of 20 kPa using a supercharger. The engine rotation was set between 800 RPM to 2000 RPM with an increment of 400 RPM. The main injection timing was varied with 2° increment from 1° BTDC to 3° ATDC. The results indicated the increase of in-cylinder pressure and heat release rate with increased boost pressure. Higher engine rotation led to the decrease of the maximum heat release rate, maximum in-cylinder pressure, and the difference between the magnitude of the first and second onsets of the in-cylinder pressure raise. It also shifted the timing for the peak of the heat release rate to occur further away from TDC. The change of the main injection timing from 1° BTDC to 3° ATDC decreased the maximum in-cylinder pressure and moved the location of the maximum in-cylinder pressure away from TDC. The delay of the main injection timing brought larger in-cylinder pressure raise for the first onset but lower cumulative heat release rate. The difference between experimental and numerical measurements of the in-cylinder pressure was found to be less than 4%. The results of the study suggested that boost pressure of 60 kPa and main injection timing of the 1° BTDC provide higher in-cylinder pressure and cumulative heat release rate and consequently better engine performance.

Keywords: Boost pressure, Direct injection, Diesel engine, Engine performance, Heat release rate, Injection strategy, In-cylinder pressure.

1. Introduction

The world is confronted with issues regarding energy availability due to the increase of the energy needs at a global scale, lingering dependence with non-renewable source of energy, and overpopulation [1, 2]. Excessive utilization of non-renewable fuels depletes natural assets and leads to an increase in carbon dioxide emissions, which contributes heavily to global warming [3]. According to the report from the International Energy Agency, 26.5% of the CO₂ emissions are related to the transportations sector [4]. With the advancements of technology, the demand for energy in society raises; coupled with the depletion of fossil fuel and its high dependency, it is reasonable to assume that the demand for energy will eventually be higher than the available supply of energy. The awareness and solutions for this issue are of high importance and have become more relevant in order to maintain a functioning society in the future [5].

Many solutions have been devised to solve those issues. Some methods used include the use of renewable fuel like biogas, biodiesel from edible and non-edible sources, briquette which is cheaper and beneficial in the economic aspect, and the production of efficient engines that generate more power and less hazardous emissions [5-11]. Another known method includes the development of the engine technology [12]. Advancements of the engine technology are motivated by rigorous standards of emissions and demand for better efficiency. It is also required to minimize the excessive use of fossil fuels. All these methods aim to decrease the deterioration of nature and human well-being [13].

Compression ignition (CI) engine is one of many engines used to convert fossil fuel into usable energy. With its great endurance and efficiency [14], CI engine has become popular in the automotive industry. Diesel engine stands out as the most universal combustion engine utilized for generating energy in many sectors [15]. The use of a diesel engine is not limited to transportation; power generation is a good example of a sector that utilizes the diesel engine. Unfortunately, the universal usage of diesel engines leads to the overutilization of them which causes a rapid decline of diesel fuel's availability and increases its price. This, in turn, leads to negatively affect the importing countries' economy. To avoid such an outcome, management of the non-renewable fuels is required. The management for non-renewable fuel is highly related to the manufacturer of engines, consumers, and scientists associated with combustion studies [15]. Another alternative is to increase the availability of diesel fuel through the development of alternative fuels derived from renewable sources such as microalgae [16].

Another important aspect of the utilization of the diesel engine comes from its emissions. Health concern due to the exposure of the engine emissions began in the mid-1990s and motivated the advancement of emission regulations and studies to enhance the technology and identifying possible health threats. For recent decades, particulate matters (PM) derived from a diesel engine have been strictly regulated [17]. Strict regulations for emission control were adopted worldwide, and therefore automotive OEMs minimized the emissions of PM tremendously through the development of technologies that treat the exhaust gas [18, 19]; such as the port fuel injection method [19]. Unfortunately, difficulties were found in decreasing the NO_x and PM residue together as a result of the soot-NO_x exchange. Large NO_x and PM emissions are currently the main problem in the advancement of regular diesel engine.

Despite the challenges, many methods have been discovered to enhance the performance of a diesel engine. Some proven methods include maintaining an optimal air-fuel ratio, oxygen availability, a proper level of injected fuel, and coupling the engine with supercharger [20, 21]. According to Ichiyanagi and Suzuki, the thermal efficiency and engine torque are increased through combining long-stroke engine with supercharger [21]. Several studies have also investigated the engine power, brake power, in-cylinder pressure, and specific power output in a diesel engine under supercharged conditions [22, 23]. Two parameters that are commonly used to measure the engine performance are in-cylinder pressure and heat release rate. In-cylinder pressure gives the idea of the processes that are occurring within the combustion chamber [24]. The heat release rate, on the other hand, varies the temperature and pressure of the gas inside the chamber [25]. The heat release rate and in-cylinder pressure are good indicators of the fuel economy, power output, gas exchange, emissions, and IC engine operating processes [26-29].

The nature of combustion, performance, and discharge of negative emissions of a diesel engine rely on a range of aspects, including Start of Injection (SOI), Fuel Injection Parameter (FIP), number of fuel insertions, construction of the ignition chamber, and the arrangement of the nozzle spray. High-pressure direct injection (HPDi) shows promise as a highly effective approach to adhere to the strict international emission regulations. FIP for diverse types of diesel engines has varying degrees between 200 to 2000 bar. Kato et al. [30] indicated the utilization of high insertion pressures of fuel in order to decrease the emission of PM while maintaining similar level of NO_x. However, High FIPs are likely to require an unusual spray design in comparison with low FIPs sprays [31]. This situation is a result of considering the generation of cavitation around the nozzles at high FIPs that leads to large atomization speed [32]. Several investigations [33, 34] indicated that larger FIPs increase the quality of air-fuel mixing and cause quicker combustion that precisely affects the formation of negative emissions. The identification of diesel spray is commonly conducted by focusing on parameters including the penetration of spray edge, the angle of spray, speed of droplet, the proportion of droplet and its distribution, and overall spray construction.

A good understanding of these characteristics is crucial in developing a more efficient diesel engine while reducing the environmental damage of using it. Large pressure differences among the nozzles of the injectors are important to atomize the fuel into tiny droplets that allow quick vaporization and large jet penetration towards the ignition chamber [35, 36]. The proportion of droplets allocation for a spray typically influences the burning process inside CI engine. Smaller fuel droplets vaporize faster in comparison with bigger droplets. Nonetheless, the penetrative attribute for small droplets is less, and accordingly, the proportion of distribution requires optimization. Chen et al [36] suggested that little droplets and huge penetrating distance for the jet fuel increase the nature of the air-fuel blend that generates less delay of ignition and a more thorough combustion. Less FIPs provide bigger droplet and therefore raise the delay of ignition time at the time of combustion [37]. It increases the cylinder pressure and leads to a larger NO_x expulsion. As the FIPs are increased, the size and distribution of the spray droplet decrease, and the air-fuel blending is improved as it has more preferable mixing attributes during ignition delay, and thus carbon monoxide and smoke are decreased [38]. Unfortunately, when FIPs reach a large value, the time of ignition becomes too short. Therefore, the prospect of consistent blending reduces which decreases the efficiency of the combustion process [39]. Bruneaux [40] studied the spray characteristics with relation to the common rail direct injection (CRDi) system

in elevated temperature cell and high pressure that simulate the state of a common diesel engine. Raise of FIPs was discovered to improve the breakdown of fuel at the exit, leading to a better proportion of gas state and enhance the blending process. Hence, the strategy for fuel insertion is an essential method in a diesel engine to improve the performance, ignition, and residual expulsions as a result of the combustion process.

For the current study, an adjustable one-cylinder engine was utilized to figure out the influence of timings in the insertion of fuel and FIP on emissions, ignition, and operational behaviours. The engine was able to accurately regulate the variables that exist in the fuel injection process. To perform the numerical study, simulation program called GT-Power was used. GT-Power is an internal combustion engine simulation program that is widely applicable in the automotive industry. Through using the one-dimensional analysis method, it decreases the time needed to develop a product and the cost involved with it [41]. It is based on a one-dimensional flow simulation and provides several combustion and in-cylinder heat transfer models. Therefore, the fundamental aim in this investigation is to study the behaviour of the DI diesel engine using in-cylinder pressure as the parameter and to support the results using GT-Power simulation with predetermined parameters. Additionally, heat release rate investigation was performed experimentally to give a more in-depth understanding of the engine performance.

An identical method of investigation has been performed by Shah et al. [42] who studied the advancement of a turbocharged DI engine simulation with GT-Power and investigate the use of B20 fuel by running it on an alternating current powered dynamometer. They discovered that the numerical results from GT-Power and the experimental results were in-line with one another as the largest difference between them was found to be 4.2% for engine brake power (BP), 3% for NO_x, 5.1% for brake specific fuel consumption (BSFC), and 3.4% for maximum combustion pressure [42].

2. Experimental and Numerical Setup

This study was conducted by initially selecting the engine. Afterward, the modelling of the selected engine in the GT-Power program was performed. After modelling of the engine, the experiments and simulations were conducted and the results from both experiments and simulations were compared. In-cylinder pressure, an important parameter in the engine performance, was measured experimentally and numerically. Additional measurement of the heat release rate was performed experimentally as well to give a more in-depth understanding of the engine performance.

In this study, an adjustable one-cylinder DI diesel engine was selected to be used in the experimental investigation. The engine bore diameter was 85 mm, the stroke's length was 97 mm, and the compression ratio was 16.3. The cylinder head and piston were made of aluminium alloy, whereas the dry liner was made of cast iron. The cylinder offset was 6.5 mm, while the piston-pin offset was 0.8 mm. The length of the connecting rod was 150.5 mm. The 4-strokes engine had 720° per engine cycle. The configuration of the experimental setup is given in Fig. 1 and the photograph of the experimental setup is given in Fig. 2. The specifications of the DI diesel engine used in this study is shown in Table 1. To experimentally measure the in-cylinder pressure, Kistler Japan 6052 pressure probe was utilized. As for the heat release rate, the measurement was conducted by measuring the heat flux inside the engine with high-response coaxial thermocouple.

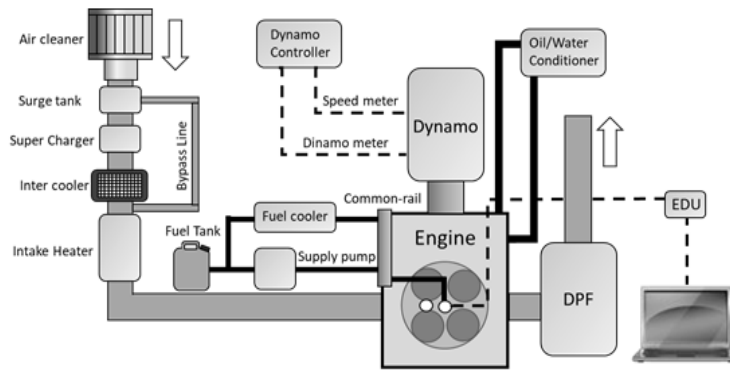


Fig. 1. Configuration of the experimental setup.



Fig. 2. Photograph of the experimental setup.

Table 1. Engine specifications.

Engine	Single cylinder engine
Cylinder Displacement (cc)	550
Bore (mm)	85
Stroke (mm)	97
Cylinder offset (mm)	6.5
Piston-pin offset (mm)	0.8
Connecting rod (mm)	150.5
Compression ratio	16.3
Intake valve opening period (deg.)	347 to -120
Exhaust valve opening period (deg.)	122 to 330

Modelling of the engine to numerically measure the in-cylinder pressure in GT-Power was firstly done on the inlet system at the pipe parts of air inductions process. The intake system was composed of a few components such as inlet, intake manifold, and intake port and intake valve. In this diesel engine, intake port was composed of intake manifold tangential port and intake manifold swirl port. After modelling of the inlet system, the exhaust system was modelled as well. The diameter of the inlet pipe was 52 mm, while the diameter of the outlet pipe was 54 mm. The intake valve was composed of intake valve tangential and intake valve

swirl. After modelling of the exhaust system, the main engine component was modelled. The modelling of the engine included the engine head and cylinder. After the modelling of the engine, the supercharger was modelled. The modelling for the numerical study was made to be as accurate as possible by representing it with a one-cylinder diesel engine and supercharger.

For this study, two experiments with a different set of parameters were performed. Both experiments varied the boost pressure from 0 kPa (naturally aspirated) to 60 kPa with an increment of 20 kPa. The first experiment was conducted with fixed main injection timing of 1° ATDC (After Top Dead Centre) and varying engine rotations of 800 RPM to 2000 RPM with 400 RPM increments. The first experiment was performed to understand the impact of varying boost pressures under various engine rotations in a DI diesel engine under a constant main injection timing. In conducting the first experiment, the lubricating oil and coolant were set at a controlled temperature of 80 °C. Afterward, the engine rotation was set to the desired speed and kept constant with the help of a dynamo controller. Fuel was injected into the combustion chamber using a common rail injection system under 80 mPa pressure. When the ignition started, the rotation of the engine was automatically kept constant at a certain value under the controlled load of the dynamometer. The measurement started from 0 kPa boost pressure to 60 kPa boost pressure. Initially, at 0 kPa boost pressure, the engine load was set at full load in all levels of engine rotations and the excess air ratio was set to 1. Afterward, the boost pressure was increased with an increment of 20 kPa and up to 60 kPa. Details about the first experiment are further given in Table 2.

Table 2. Experimental conditions at constant main injection timing of 1° ATDC for various engine rotations and powers.

Engine rotation [RPM]	Step	Main injection timing [deg.]	Time of insertion [ms]	Fuel injected [g/st]	Total fuel insertion [g/st]	Power [watt]
800	Pilot	-25	0.16	0.0014	0.0398	2700
	Pre	-15	0.18	0.0024		
	Main	1	0.83	0.036		
1200	Pilot	-25	0.16	0.0014	0.0398	4500
	Pre	-15	0.18	0.0024		
	Main	1	0.83	0.036		
1600	Pilot	-25	0.16	0.0014	0.0398	6200
	Pre	-15	0.18	0.0024		
	Main	1	0.83	0.036		
2000	Pilot	-25	0.16	0.0014	0.0398	7800
	Pre	-15	0.18	0.0024		
	Main	1	0.83	0.036		

The second experiment was performed under a fixed engine rotation of 1200 RPM with various main injection timings of 1° BTDC, 1° ATDC, and 3° ATDC. The second experiment was conducted to find out the effect of boost pressure and sequence of main fuel injection towards the behaviour of a DI diesel engine at a constant engine rotation. The steps performed to conduct the second experiment were similar to the first experiment. The differences between them are in the constant engine rotation and the change of main injection timing for the second experiment. The engine load was also set at full load from the aspect of air excess

ratio for engine rotation of 1200 RPM. The history of both in-cylinder pressure and heat release rate were adjusted following the difference of the main injection timing. Details regarding the second experiment are provided in Table 3.

Many previous studies compared the main injection timing only between different BTDC timings or different ATDC timings. This study investigated and compared the influence of main injection timing for both BTDC and ATDC timings on the performance of a DI diesel engine. This holistic investigation was conducted as Anggono et al. [25] suggested that the engine rotation and the combustion speed in the engine will be balanced by advancing the fuel main injection timing from ATDC to BTDC. With the balance between engine rotation and combustion speed, the maximum in-cylinder pressure will occur near TDC.

Table 3. Experimental conditions at a constant engine rotation of 1200 RPM for various fuel insertion timings and powers

Engine rotation [RPM]	Step	Main injection timing [deg.]	Time of insertion [ms]	Fuel injected [g/st]	Total fuel insertion [g/st]	Power [watt]
1200	Pilot	-25	0.16	0.0014	0.0398	4600
	Pre	-15	0.18	0.0024		
	Main	-1	0.83	0.036		
	Pilot	-25	0.16	0.0014	0.0398	4300
	Pre	-15	0.18	0.0024		
	Main	1	0.83	0.036		
	Pilot	-25	0.16	0.0014	0.0398	4300
	Pre	-15	0.18	0.0024		
	Main	3	0.83	0.036		

3. Results and Discussion

The outcomes of the first experiment under various engine rotations and boost pressures towards in-cylinder pressure are displayed in Figs. 3 to 6. These figures display the result of the in-cylinder pressure from the experiment (solid line) and GT-Power simulation (dashed line). The results indicate that in-cylinder pressure increased gradually as the boost pressure was raised from 0 kPa to 60 kPa. In contrast, the in-cylinder pressure decreased with increased engine rotation caused by an extended period of combustion on greater engine rotation.

It was also discovered from the results that the in-cylinder pressure increased in two onsets. The first onset occurred in the negative crank angle near TDC and the second onset occurred in the positive crank angle. The increase of the engine rotation reduced the difference between the magnitude of the in-cylinder pressure raise of the first and second onsets. This phenomenon was caused by the increase of combustion duration at higher engine rotation.

In order to obtain the simulated results, the measured pipes data, compressor map, burn rate, amount of fuel burned for each case, and swirl data were required [43]. Unfortunately, some of the required data obtained were not accurate in this study; the inaccuracy of these data may explain the deviation that occurred in the case of 1200 RPM. The accuracy of the results may be improved if a compressor map was available from the manufacturer and the burn rate data were obtained from the experiments. The burn rate is a crucial parameter as it precisely manages the period of efficiency, the

vibration and noise due to engine operation, and the maximum pressure that was exposed to the engine [44]. However, it was impossible to acquire the burn rate due to the lack of the appropriate measuring tool.

To analyse the difference between experimental and numerical results, both results were compared. Table 4 shows the value and location of the maximum in-cylinder pressure and the difference between experimental and numerical results for 1° ATDC with various boost pressures and engine rotations. It was observed that the location deviation did not exceed beyond 3° and the deviation of value between experimental and numerical results did not exceed beyond 184 kPa (4%).

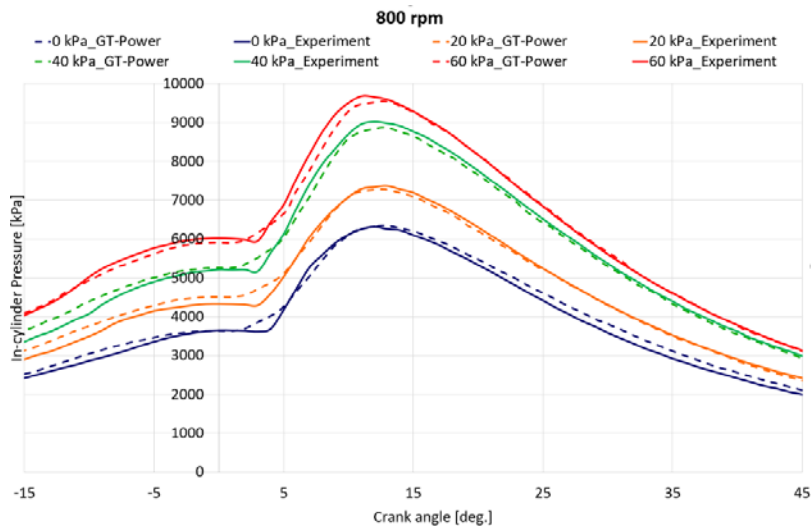


Fig. 3. In-cylinder pressure with main injection timing of 1° ATDC for various boost pressures and engine rotation of 800 RPM.

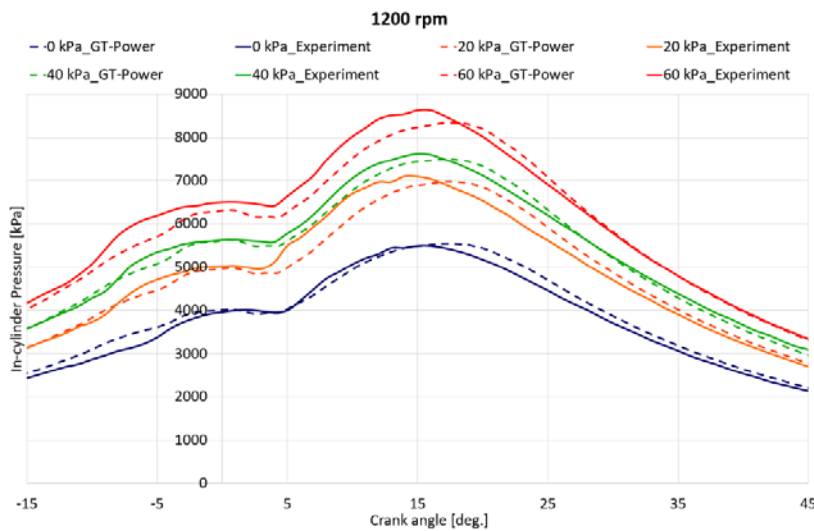


Fig. 4. In-cylinder pressure with main injection timing of 1° ATDC for various boost pressures and engine rotation of 1200 RPM.

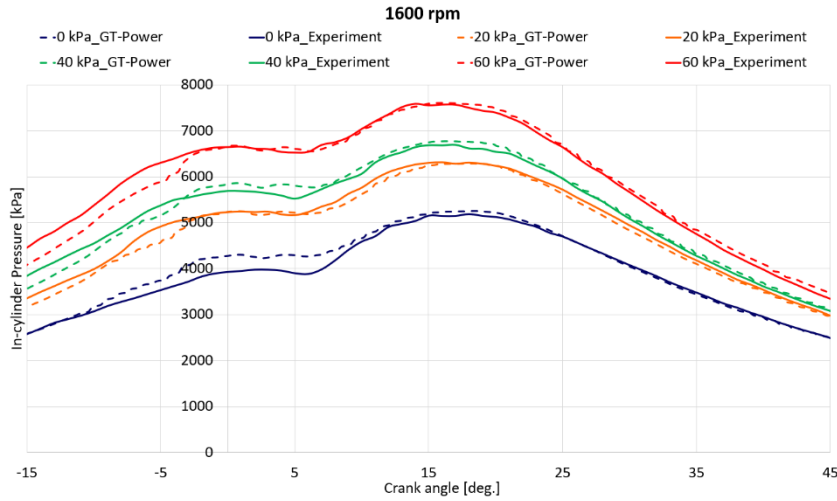


Fig. 5. In-cylinder pressure with main injection timing of 1° ATDC for various boost pressures and engine rotation of 1600 RPM.

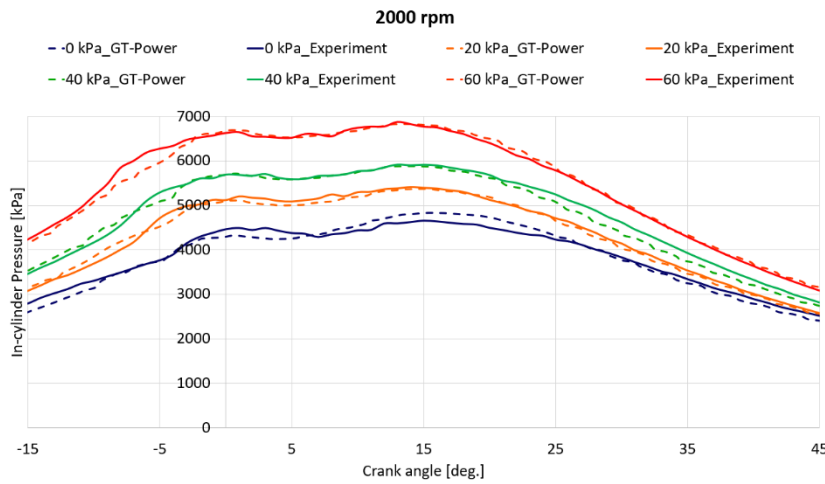


Fig. 6. In-cylinder pressure with main injection timing of 1° ATDC for various boost pressures and engine rotation of 2000 RPM.

Table 4. Deviation among experiment and GT-Power results for the in-cylinder pressure comparison with main injection timing of 1° ATDC under various boost pressures and engine rotations.

Engine rotation [RPM]	Boost pressure [kPa]	GT-Power		Experiment		Deviation		
		Location [deg.]	Value [kPa]	Location [deg.]	Value [kPa]	Location [deg.]	Value [kPa]	[%]
800	0	16.42	5630.26	15.00	5499.77	1.42	130.49	0.31
	20	16.42	7177.25	14.00	7096.19	2.42	81.06	1.21
	40	16.21	7716.32	15.00	7623.65	1.21	92.67	1.56
	60	16.25	8687.32	16.00	8633.97	0.25	53.35	1.33
1200	0	12.97	6342.81	12.00	6323.25	0.97	19.56	0.91
	20	12.36	7276.73	13.00	7366.01	0.64	89.28	1.72

	40	12.46	8879.45	12.00	9020.45	0.46	141.00	2.05
	60	12.47	9554.50	11.00	9683.28	1.47	127.78	3.35
	0	18.14	5249.90	18.00	5192.84	0.14	57.06	1.10
1600	20	18.32	6295.21	16.00	6309.52	2.32	14.31	0.23
	40	16.34	6173.35	17.00	6691.63	0.66	82.32	1.23
	60	16.01	7603.25	14.00	7579.99	2.01	23.56	0.31
	0	15.36	4834.12	15.00	4654.46	0.36	183.66	3.95
2000	20	15.09	5372.05	14.00	5421.23	1.09	49.18	0.91
	40	14.04	5885.65	13.00	5914.38	1.04	28.73	0.49
	60	13.32	6830.96	13.00	6872.61	0.32	41.65	0.61

The results of the heat release rate measurements for the first experiment are shown in Figs. 7 to 10. The measurements suggested that the maximum heat release rate increased with increased boost pressure. The results are in line with studies conducted by Chen et al. and Chala et al. [45, 46]. In this study, an exception was found in the 800 RPM condition where the heat release rate for 40 kPa was slightly higher than 60 kPa boost pressure. This could be the result of the shorter combustion duration in lower engine rotation which caused the mass-burn rate to be constant even with increased boost pressure. For engine rotation of 1200 to 2000 RPM, the difference between boost pressures was apparent with 60 kPa having the highest maximum heat release rate.

On the other hand, the increase of the engine rotation reduced the maximum heat release rate for all levels of boost pressures. It was also discovered from the results that the timing of the maximum heat release rate shifted towards higher crank angle (slower) with increased engine rotation. Longer combustion duration in higher engine rotation reduced the burn rate. It caused the heat release rate to be distributed along with a large range of crank angle and therefore shifting the peak away from TDC.

Figures 11 to 14 show the results of the second experiment with various main injection timings and boost pressures. The results showed an increase of in-cylinder pressure with increased boost pressure for all main injection steps. The results are identical to the evaluation performed by Lee et al. [47]. They suggested that the maximum in-cylinder pressure increased due to supercharging the engine with engine rotation below 3000 RPM. The differences between various main injection timings studied were found in the value and the timing of the maximum in-cylinder pressure.

The maximum in-cylinder pressure in the main injection timing of 1° BTDC was higher than that of the 1° ATDC and 3° ATDC in all boost pressure variations. Advancing the main injection timing also led to the maximum in-cylinder pressure to occur closer to TDC [48]. These conditions occurred due to the lagging reaction of the combustion caused by the retardation of the main injection timing. It shifted the location of the maximum in-cylinder pressure towards a larger crank angle (slower). The lagging reaction of the combustion also decreased the maximum in-cylinder pressure as the fuel was mostly combusted after TDC. It is also discovered that in delayed main injection timing, a lower value of the maximum in-cylinder pressure made the raise of the in-cylinder pressure in the first onset to be larger [5].

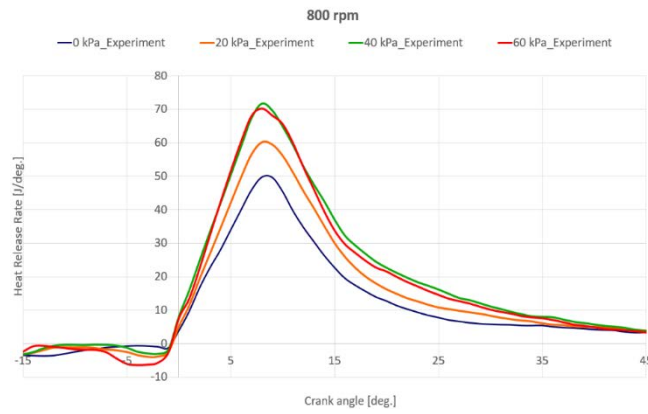


Fig. 7. Heat release rate with main injection timing of 1° ATDC for various boost pressures and engine rotation of 800 RPM.

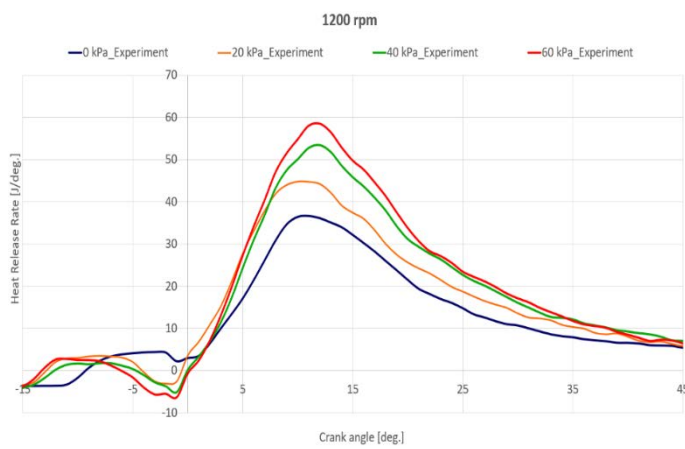


Fig. 8. Heat release rate with main injection timing of 1° ATDC for various boost pressures and engine rotation of 1200 RPM.

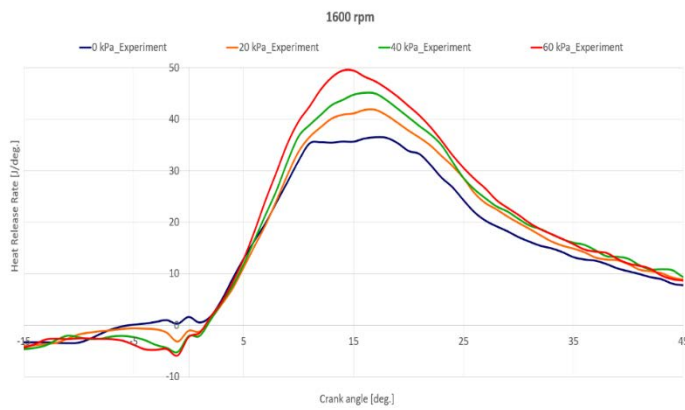


Fig. 9. Heat release rate with main injection timing of 1° ATDC for various boost pressures and engine rotation of 1600 RPM.

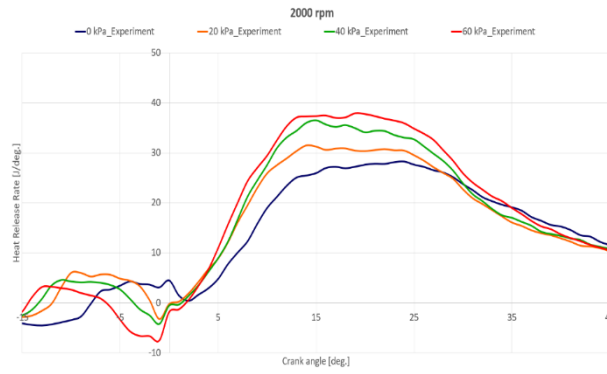


Fig. 10. Heat release rate with main injection timing of 1° ATDC for various boost pressures and engine rotation of 2000 RPM.

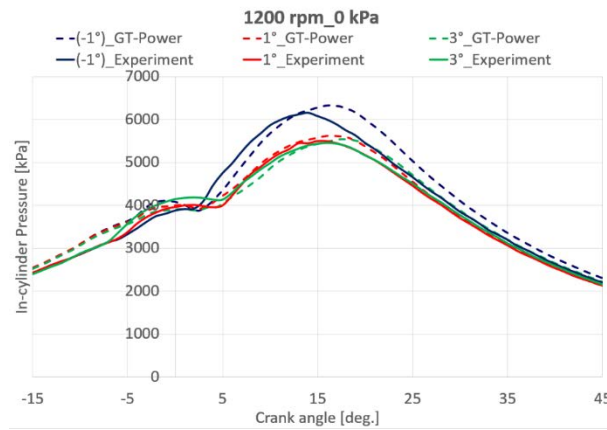


Fig. 11. In-cylinder pressure of various main injection timings with 1200 RPM engine rotation and 0 kPa boost pressure.

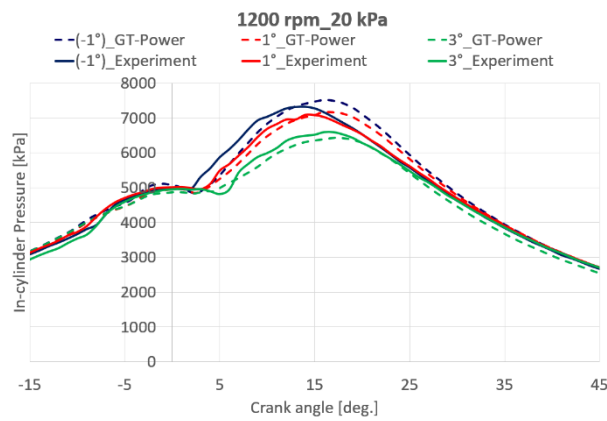


Fig. 12. In-cylinder pressure of various main injection timings with 1200 RPM engine rotation and 20 kPa boost pressure.

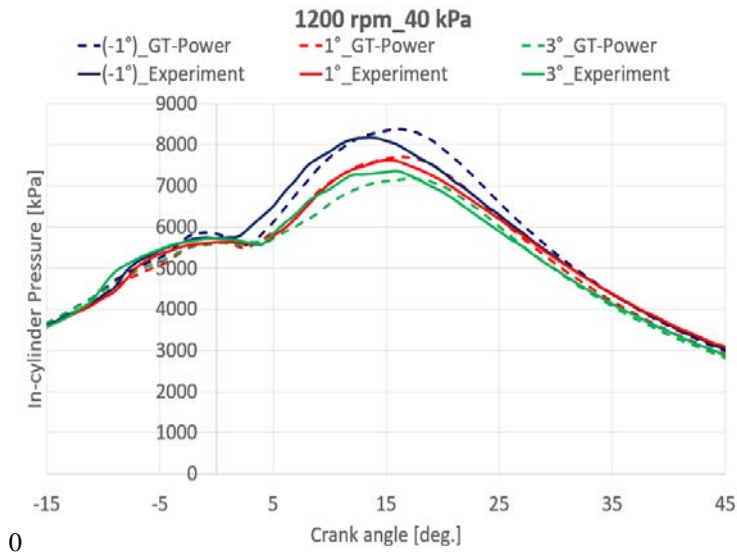


Fig. 13. In-cylinder pressure of various main injection timings with 1200 RPM engine rotation and 40 kPa boost pressure.

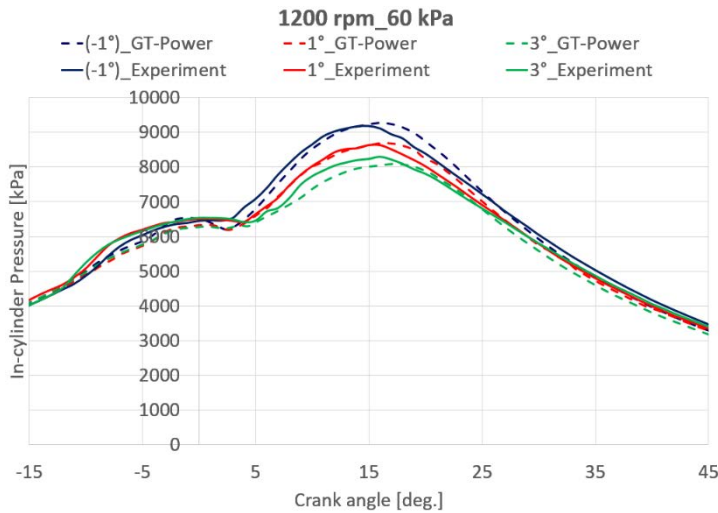


Fig. 14. In-cylinder pressure of various main injection timings with 1200 RPM engine rotation and 60 kPa boost pressure.

Table 5 shows the value and location of the maximum in-cylinder pressure and the difference among experiment and GT-Power results for main injection timing of 1° BTDC, 1° ATDC, and 3° ATDC under 1200 RPM and various boost pressures. The largest location deviation was 2.38 degrees, and the largest deviation of value between experimental and numerical results was 222.93 kPa (4%). The small deviations of location and value show a good agreement between experimental results and the numerical results from GT-Power.

Table 5. Deviation among experiment and GT-Power results for in-cylinder pressure comparison with main injection timing of 1° BTDC, 1° ATDC, and 3° ATDC under 1200 RPM and various boost pressures.

Boost pressure [kPa]	Main injection timing [deg.]	GT-Power		Experiment		Deviation		
		Location	Value	Location	Value	Location	Value	
		[deg.]	[kPa]	[deg.]	[kPa]	[deg.]	[kPa]	[%]
0	-1°	16.38	6330.72	14.00	6155.43	2.38	175.29	0.28
	1°	12.97	6342.81	12.00	6323.25	0.97	19.56	0.91
	3°	17.84	5534.43	16.00	5351.98	1.84	182.45	1.40
20	-1°	16.20	7516.16	14.00	7319.84	2.20	196.32	0.61
	1°	12.36	7276.73	13.00	7366.01	0.64	89.28	1.72
	3°	17.17	6439.82	16.00	6496.43	1.17	56.61	1.66
40	-1°	16.26	8389.33	14.00	8166.40	2.26	222.93	0.62
	1°	12.46	8879.45	12.00	9020.45	0.46	141.00	2.05
	3°	17.22	7184.84	16.00	7262.82	1.22	77.98	2.07
60	-1°	16.28	9281.22	15.00	9177.34	1.28	103.88	2.12
	1°	12.47	9554.50	11.00	9683.28	1.47	127.78	3.35
	3°	17.22	8100.36	16.00	8198.28	1.22	97.92	2.07

Figures 15 to 18 show the results of the heat release rate measurements for the second experiment under various main injection timings and boost pressures. Increasing the boost pressure led to a higher heat release rate for all main injection timings. The results also showed that the retardation of the main injection timing from 1° BTDC to 1° ATDC shifted the peak of the heat release rate away from TDC. These results are in line with the study conducted by How et al. [49]. However, retardation of the main injection timing from 1° ATDC to 3° ATDC did not show a significant shift of timing for the peak to occur.

These results also showed that the retardation of main injection timing caused the peak of heat release rate to be narrower and therefore having a faster burning rate [49]. Only slight differences in maximum heat release rate were observed between different main injections timing at all levels of boost pressures. However, the main injection timing of 1° BTDC generated the highest overall heat release rate as it had a less narrow peak and therefore released more heat.

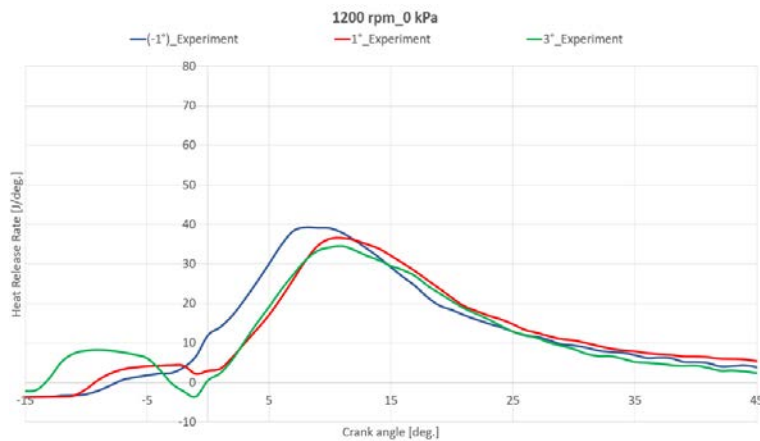


Fig. 15. Heat release rate of various main injection timings with 1200 RPM engine rotation and 0 kPa boost pressure.

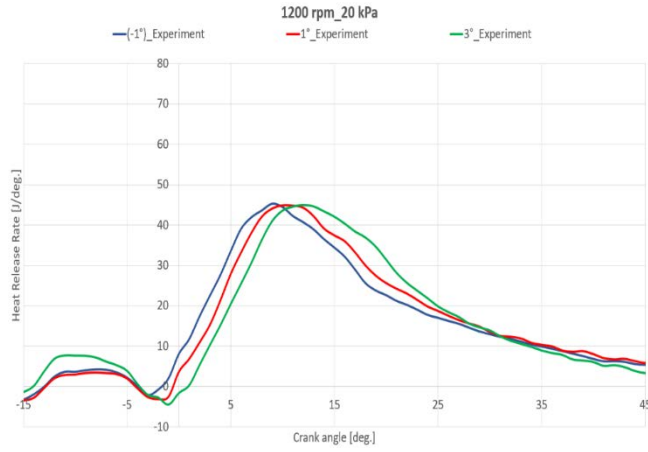


Fig. 16. Heat release rate of various main injection timings with 1200 RPM engine rotation and 20 kPa boost pressure.

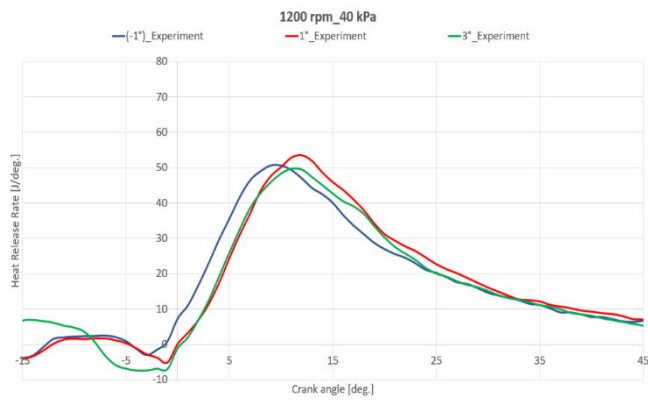


Fig. 17. Heat release rate of various main injection timings with 1200 RPM engine rotation and 40 kPa boost pressure.

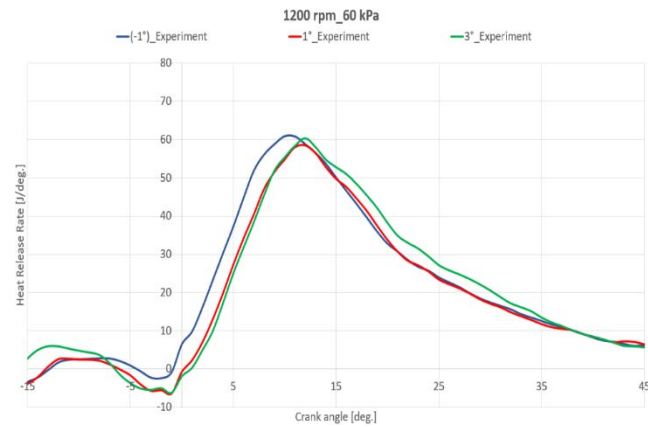


Fig. 18. Heat release rate of various main injection timings with 1200 RPM engine rotation and 60 kPa boost pressure.

4. Conclusions

The effect of varying boost pressures from 0 kPa to 60 kPa and main injection timings from 1° BTDC to 3° ATDC in a direct injection diesel engine has been investigated in terms of the in-cylinder pressure and heat release rate. From the investigation, several points may be raised as follows:

- The in-cylinder pressure results from both GT-Power simulation and experiment were in good agreement with the largest deviation not exceeding beyond 4%.
- Addition of boost pressure to the DI diesel engine increased the in-cylinder pressure and heat release rate in all main injection timings investigated. On the other hand, the increase of the engine rotation from 800 to 2000 RPM reduced the in-cylinder pressure and heat release rate due to the longer combustion process at higher engine rotation.
- Two onsets of in-cylinder pressure raise were found in this study. The first onset occurred in the negative crank angle and the second onset occurred in the positive crank angle. The increase of the engine rotation reduced the contrast between the maximum in-cylinder pressure of the first and second onsets. Furthermore, the timing of the maximum heat release rate moved further away from TDC with higher engine rotation. The contrast between onsets and shifts of the timing for the maximum heat release rate were caused by the delay in the combustion with increased engine rotation as it raised the overall duration of the combustion process and therefore providing longer duration for heat to release.
- The retardation of the main injection timing caused a decrease in the maximum in-cylinder pressure and shifted the location to occur further away from TDC. Those were due to the lagging reaction of the combustion caused by the retardation of the main injection timing. The in-cylinder pressure raise on the first onset was found to be larger on the retarded main injection timing. Advancing the main injection timing to 1° BTDC led to earlier raise of heat release rate and higher cumulative heat release rate due to the less narrow peak.
- The outcomes of this study suggested that increasing the boost pressure with supercharger up to 60 kPa and advancing the main injection timing to 1° BTDC resulted in higher boost pressure and cumulative heat release rate and therefore better engine performance.

Acknowledgment

This study was conducted through collaborative supports from Sophia University, Tokyo, Japan, under Sophia Lecturing - Research Grant Scheme 2018 and Petra Christian University, Surabaya, Indonesia (Project code: 553/YPTK/XII/2017). This study was also supported by Japan Society for the Promotion of Science, Grants-in-Aid for Scientific Research (No.19K04244)

Abbreviations

1-D	One-Dimension
AC	Alternative Current
ATDC	After Top Dead Centre
BP	Brake Power
BSFC	Brake Specific Fuel Consumption

BTDC	Before Top Dead Centre
CI	Compression Ignition
CRDi	Common Rail Direct Injection
DI	Direct Injection
FIP	Fuel Injection Parameter
HPDi	High-Pressure Direct Injection
IC	Internal Combustion
NOx	Nitrogen Oxides
OEMs	Original Equipment Manufacturers
PM	Particulate Matters
SOI	Start of Injection
TDC	Top Dead Centre

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