EXPERIMENTAL INVESTIGATIONS OF A SIGNLE CYLINDER GENSET ENGINE WITH COMMON RAIL FUEL INJECTION SYSTEM

by

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Performance and emissions characteristics of compression ignition engines are strongly dependent on quality of fuel injection. In an attempt to improve engine combustion, engine performance, and reduce the exhaust emissions from a single cylinder constant speed genset engine, a common rail direct injection fuel injection system was deployed and its injection timings were optimized. Results showed that 34° CA bTDC start of injection timings result in lowest brake specific fuel consumption and smoke opacity. Advanced injection timings showed higher cylinder peak pressure, pressure rise rate, and heat release rate due to relatively longer ignition delay experienced.

Key words: *common rail direct injection, emission, engine performance, combustion, rate of heat release*

Introduction

Diesel engines are high thermal efficiency machines, which are reliable, require lower maintenance and have lower specific fuel consumption. Simultaneous reduction in NO_x and particulate emissions from compression ignition (CI) engines is a major research challenge being faced today due to enormous pressure exerted by stringent emission norms being adopted worldwide. Fuel injection at higher pressure and precise control of injection timing are helpful in reducing the engine out emissions. In the electronic fuel injection system, the fuel injection parameters such as fuel injection pressure, rate of fuel injection, multiple injections, and the start of injection (SOI) timings are controlled precisely by an electronic control unit (ECU) for different engine operating conditions. Various types of electronic fuel injection systems used in CI engines today include unit pump systems, unit injector systems and common rail direct injection systems. Common rail direct injection system is suitable for comparatively lower fuel injection pressure than the other two systems mentioned above and it offers unparallel flexibility in injection strategy. For the reduction of combustion noise, NO_x and soot emissions, different combination of fuel injection strategies such as pilot-main, main-post, pilot-main-post, pilot-pilot-main-post, *etc*. can be devised and applied in production grade engine to meet desirable performance and emission goals [1].

Cheng *et al*. [2] carried out research on single cylinder engine with fuel injection pressure of 1800 bar. Such a high pressure provided superior charge mixing and reduced the emissions. Kong and Karra [3] further increased the fuel injection pressures up to 2000 bar. This higher pressure significantly reduced the soot emissions. Researchers have demonstrated that by

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pre-injection of fuel, direct injection (DI) diesel engines can simultaneously reduce NO_x and particulate emissions, while reducing the combustion noise also [4, 5].

Mallamo *et al.* [6] explored different injection strategies such as main, pilot-main, pilot-pilot-main, and pilot-pilot-main-post fuel injection strategies on a small non-road diesel engine prototype. Different fuel injection strategies with a fixed number of multiple injections (up to 4) were tested for different load-speed operating points. It was found that optimized injection timings with multiple injection strategies are effective in reduction of particulate matter (PM), NO_x, and noise levels without increasing the specific fuel consumption [6, 7]. Takeda *et al*. [8] advanced the fuel injection timing greatly to promote fuel-air mixing. Nehmer and Reitz [9] carried out research on split injection. Split injection leads to improved utilization of the charge and allows combustion to continue later into the power stroke compared to a single injection case. This reduces the NO_x level without significant increase in soot production level. Lee and Reitz [10] extended the work on split injection in combination with exhaust gas re-circulation (EGR) and other parameters on a high speed direct injection (HSDI) diesel engine using response surface method (RSM) optimization method. The benefits of combined effect of EGR with supercharging were realized by RSM optimization. This study concluded that there were simultaneous reductions in both NO_x and PM emissions while it also improved brake specific fuel consumption (BSFC).

By post injection, soot particles are re-burned and this reduces the soot by 20-70% [11]. Tsurushima *et al*. [12] introduced the concept of post-injection in an optical engine to investigate the cause for unburnt hydrocarbons. This study revealed that post-injection reduces the HC, CO, as well as PM emissions. This was primarily due to the oxidation of unburned fuel which remained after the main injection. Desantes *et al*. [13] carried out research on a DI diesel engine to show the effect of post injection timings on the exhaust emissions. They concluded that if the post-injection is placed close enough to the main injection, the end of combustion can take place earlier compared to single injection strategy. In such conditions, NO_x emissions increase due to higher temperature levels prevailing during the last stage of combustion but soot and specific fuel consumption decrease due to faster combustion in the last phase. In an engine, friction is affected by liner surface properties [14] and the wear debris affect the lubricating oil significantly [15] therefore it is essential to investigate the effect of change in injection strategies on the friction, engine durability and lubricating oil performance.

In the present study, above mentioned advancements in common rail direct injection (CRDI) system were utilized to improve the performance, emissions and combustion characteristics of a genset engine by deploying and optimizing CRDI fuel injection system. Detailed investigations of variations in engine performance, emissions and combustion characteristics with varying fuel injection timings at different engine loads were carried out.

Experimental set-up

A single cylinder, four-stroke, water-cooled, DI diesel engine, coupled with an alternative current (AC) genset was modified to change its fuel injection system from mechanical fuel injection system to CRDI fuel injection system. Compression ratio of engine was 17.5 with 102 mm bore and 115 mm stroke length. Schematic of this experimental set-up is shown in fig. 1. Modified fuel injection system includes a high pressure fuel pump, a common rail, a solenoid valve fuel injector, a fuel filter and high pressure fuel lines (fig. 2). The purpose of this system is to supply required fuel quantity at desired pressure at appropriate timing in an engine cycle to the engine. High pressure pump (DELPHI; 15316) is used for generating the high fuel pressure. The pump is driven separately by an electric motor, thus ensuring that the high pressure fuel sup-

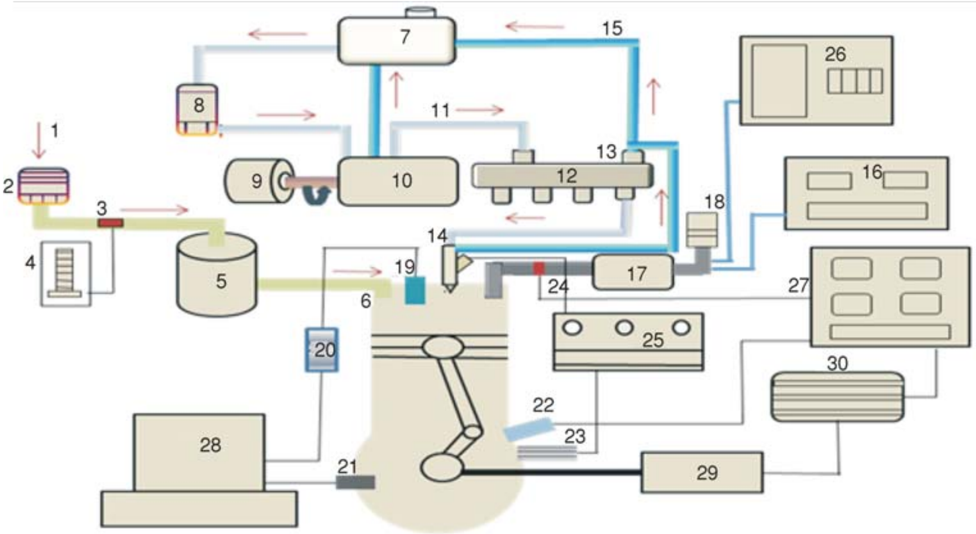

Figure1. Schematic of the experimental set-up
1 – air intake, 2 – intake aif filter, 3 – u-tube manometer, 4 – orifice plate, 5 – air box, 6 – intake manifold, 7 – fuel tank, 8 – fuel filter, 9 – electrical pump drive, 10 – high pressure pump, 11 – high pressure line, 12 – common rail, 13 – fuel pressure regulator, 14 – solenoid fuel injector, 15 – fuel return line, 16 – AVL 444, 17 – exhaust muffler, 18 – exhaust outlet, 19 – pressure transducer, 20 – charge amplifier, 21 – shaft encoder, 22 – optical speed sensor, 23 – TDC sensor, 24 – exhaust temperature sensor, 25 – injection driver circuit, 26 – smoke opacimeter, 27 – control panel, 28 –high speed DAQ, 29 – AC alternator, 30 – load bank

ply is available at all times irrespective of the engine speed. The electric motor was coupled to fuel pump through a gear drive arrangement.

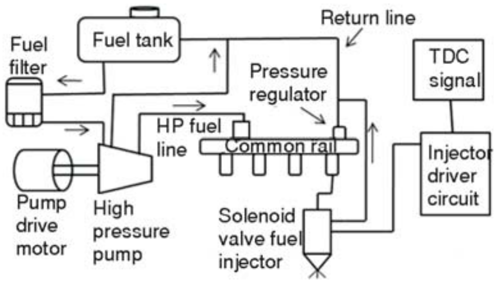

Figure 2. Schematic of the CRDI fuel injection system deployed

Common rail (DELPHI TVS; 001) accumulates the fuel at high pressure, and reduce the pressure oscillations caused by the opening and closing of the injector. For regulating the rail pressure, one spring loaded valve is connected to one of the outlet ports of the common rail thus maintaining the pressure in the rail at 280 bar; therefore the fuel injection pressure in this version of CRDI was limited to 280 bar. The common rail fuel injector is a vital component in the fuel injection system, regulating the exact fuel quantity that is required to be delivered to the combustion chamber at an appropriate timing in the engine cycle. Therefore a solenoid valve injector was used in this experiment. An analog injection driver circuit was designed, fabricated and tested to control the "rate of fuel injection" and SOI timing of the solenoid valve injector by generating the required TTL output signal pulses. The signal for SOI was derived from the inductive proximity sensor used to detect the top dead centre (TDC), which is connected to the engine camshaft and the "rate of injection" is control by changing the pulse width of the TTL signal to the injector. For controlling the SOI timing, output pulse was delayed with respect to trigger pulse using a potentiometer.

For carrying out the engine performance and emission experiments, fuel and air flow rates were measured and engine load was varied by changing load on the generator. Exhaust gas composition was measured by exhaust gas analyzer (AVL India; 444). This equipment measures NO, HC, CO, O_2, and CO_2 in the raw exhaust gas. Smoke opacity of the exhaust was measured by smoke opacimeter (AVL, India; 437).

For combustion analysis, an inductive type proximity sensor (Transducers and Allied Products, India; GLP18APS) was used as a TDC sensor. A high precision shaft encoder (Encoder India Limited, India, ENC58/6-720ABZ/5-24V) was mounted on the crankshaft for piston position measurements with respect to (w. r. t.) rotation of the crank shaft. A piezoelectric pressure transducer (Kistler Instruments, Switzerland; 6613CQ09-01) was installed in the engine cylinder head to measure instantaneous cylinder pressure w. r. t. crank angle position. Pressure and crank angle data was acquired and analyzed simultaneously by using high speed combustion data acquisition system (Hi-Techniques, USA; Synergy). For acquisition of cylinder pressure, output pulse of shaft encoder was used as an external clock. Shaft encoder has resolution of two pulses per crank angle degree rotation of the crank shaft thereby 720 pressure measurements were taken in one revolution of the engine crank shaft.

Test engine was operated at a constant engine speed of 1500 rpm, 280 bar fuel injection pressure at different engine loads. Engine performance, emission and combustion parameters were acquired and analyzed for varying SOI timings of 25º, 34º, 40º CA bTDC, respectively, at different loads at 1500 rpm engine speed.

Results and discussion

Common rail fuel injection system was optimized at 1500 rpm by varying fuel injection timings from 25º bTDC to 40º bTDC for minimum BSFC and lowest emissions of NO_x and particulates (smoke opacity).

Performance and emissions

Figure 3 shows the variation of BSFC for different fuel injection timings. The graph indicates that when the engine load increases, BSFC initially decreases to a minima and then increase and this trend was seen for all SOI timings. Fuel consumption increased with retarded (25º bTDC) as well as advanced (40º bTDC) injection timings. With retarded injection timings, there is significant amount of fuel available for late combustion, which extended well into the expansion stroke. This reduced the effective gas pressure (IMEP) used to generate engine power output. With advanced SOI timing, after the ignition delay, combustion occurs earlier and opposes the piston motion during the compression stroke, which reduces the effective power output therefore increases the BSFC.

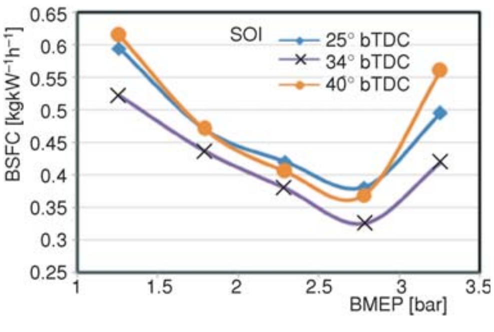

Figure 3. Variation of BSFC with engine load for varying SOI timings

Hydrocarbons mainly form as a result of incomplete and inferior combustion. Richer and leaner fuel-air mixtures, both lead to higher hydrocarbon emissions. Variation in emission of hydrocarbons for varying engine load *i. e.* varying brake mean effective pressure (BMEP) with different SOI timings are shown in fig. 4. Figure 4 shows that for lower BMEP, brake specific hydrocarbon (BSHC) emissions are relatively higher and they first decrease and then again increase with further increase in BMEP.

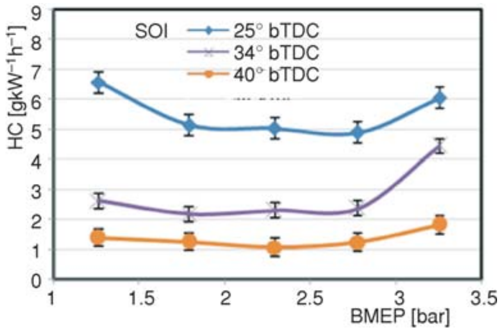

Figure 4.Variation of BSHC emissions with engine load for varying SOI timings

At higher engine loads, more fuel quantity is injected in constant mass of intake air therefore the fuel-air mixture becomes too rich to burn completely, resulting in higher HC emissions. However BSHC emissions either remains constant or decreases with increasing engine load because increase in HC emissions is proportionally lower compared to the increase in engine power output. It starts increasing again, when the engine load increases to maxima. Figure 4 also shows that with retarded fuel injection timings, emission of BSHC is higher in comparison to advanced SOI timings. This is due to delayed start of combustion (SOC) and relatively slower burning rate of air-fuel mixture for retarded fuel injection timings. At lowest engine load, BSHC emissions appear to be high because power produced by engine is disproportionally low in comparison to raw HC emissions. This trend is clearer at 25° bTDC SOI timing, when BSHC emissions at lower engine load are very high because this is the most retarded SOI timing.

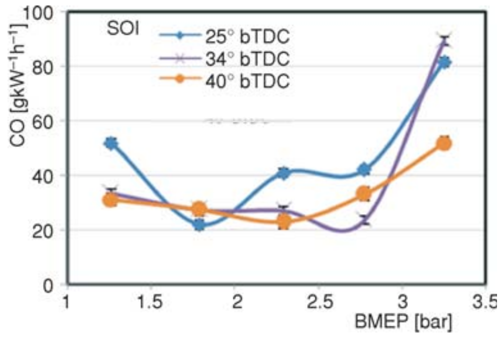

Figure 5.Variation of BSCO emissions with engine load for varying SOI timings

CO is an intermediate combustion product of incomplete fuel combustion. It is largely affected by the mixture strength in various zones of the combustion chamber. Variation of CO emissions with varying engine loads and SOI timings is shown in fig. 5.

The figure shows that at lower engine loads, brake specific carbon monoxide (BSCO) emissions are relatively higher and first they decrease and then increase with increasing engine load. At lowest engine load, BSCO emissions are high due to high CO formation in colder regions of the combustion chamber and relatively lower power produced by the engine. As the engine load increases, fuel-air mixture strength increases, resulting in combustion at higher temperature, which leads to lower CO formation. At very high loads again, lower amount of air is available and more fuel is injected therefore higher BSCO emissions are observed in such engine operating conditions. Generally, BSCO emissions were higher at retarded SOI timings because retarded fuel injection results in relatively lower combustion temperatures in the combustion chamber, which increases CO formation in the cylinder.

The formation of NO_x in CI engine is largely dependent on the oxygen concentration, peak in-cylinder temperatures, and the residence time of the species at this peak temperature for the NO_x formation reactions to occur. The variation of $BSNO_x$ emissions for different SOI timings is shown in fig. 6. The figure shows that there is overall reduction in the mass emission of NO_x with increasing engine load. This is due to decrease in oxygen concentration and increase in the turbulence levels at higher engine loads, resulting in lower residence time for NO_x specific reactions to occur. Lower combustion temperature are seen during combustion of richer mixtures, therefore reduction in $BSNO_x$ emissions is seen at high load. Advance fuel injection tim-

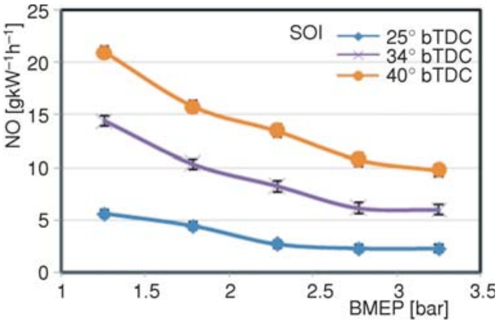

Figure 6. Variation of BSNO$_x$ emissions with engine load for varying SOI timings

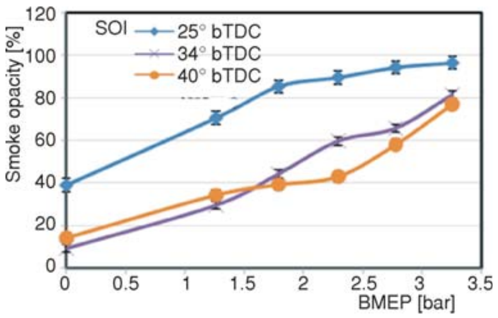

Figure 7.Variation of smoke opacity with engine load for varying SOI timings

ings result in relatively higher BSNO$_x$ emissions due to higher fuel burning rate in premixed combustion phase which leads to higher peak cylinder pressures, and higher peak combustion temperatures, therefore higher NO$_x$ formation takes place as compared to retarded fuel injection timings [16]. NO$_x$ emissions can be controlled after combustion by several techniques [17].

The variation of smoke opacity for different engine loads for varying SOI timings is shown in fig. 7. There was an overall increase in smoke emission with increasing engine loads. Increasing engine loads leads to increase in fuel-air equivalence ratio and longer diffusion combustion phase, resulting in lower oxygen concentration in the combustion chamber therefore the smoke opacity increases due to higher particulate formation. Fuel injection timings also play a vital role in the formation of soot. Retarded fuel injection timings increase the smoke opacity due to lowering of combustion temperatures and reduction in the time available for oxidation of soot during the expansion stroke, whereas advanced SOI timings lead to more complete combustion at higher temperatures, resulting in lower smoke opacity.

Combustion characteristics

Combustion analysis is important characterization tool for the analysis of engine behavior because it directly governs the engine performance and emission characteristics [18]. For combustion characteristics, high speed combustion data acquisition was carried out using high speed DAQ system for 100 consecutive engine cycles and combustion analysis was carried out on the average data set obtained from these 100 consecutive engine cycles.

Cylinder pressure variation

Measurement of cylinder pressure is important for understanding the combustion taking place in the engine combustion chamber. Measurement of cylinder pressure data is helpful in finding other combustion parameters such as heat release rate, cumulative heat release, pressure rise rate, *etc*. Cylinder pressure *vs.* crank angle diagram for different SOI timings is shown in fig. 8. From these graphs, it may be noted that advancing the SOI timing leads to higher peak cylinder pressure for all engine loads. Advancing SOI timing in CI engine provides more time for formation of premixed charge therefore relatively larger amount of fuel is burnt in the premixed combustion phase. Earlier SOI also leads to longer ignition delay therefore more fuel quantity is injected before the SOC. Upon ignition, higher cylinder temperature is attained in the combustion chamber leading to higher cylinder pressures. The peak cylinder pressure largely depends on the burnt fuel fraction in the premixed combustion phase.

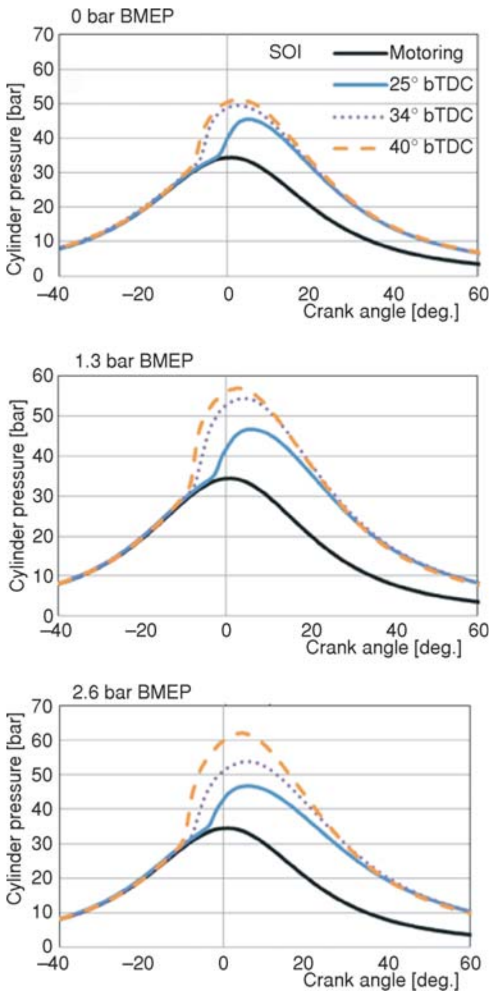

Figure 8. In-cylinder pressure variation with engine load for varying SOI timings

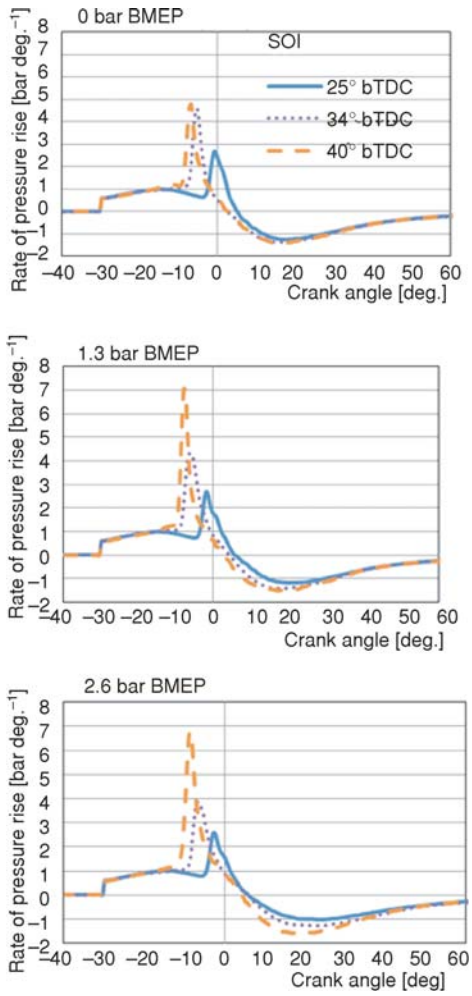

Figure 9. ROPR variation with engine load for varying SOI timings

Peak cylinder pressure for advanced SOI timings is higher in comparison with retarded SOI timings. Peak cylinder pressure increases from 45.6 bar at no load to 46.9 bar at 3 kW engine load (2.6 bar BMEP) for 25º bTDC SOI timing. It increases to 51.3 bar at no load to 62.3 bar at 3 kW load for 40º bTDC SOI timing. As the engine load increases, the peak cylinder pressure increases and its peak shifts away from the TDC due to higher fuel quantity being burnt in premixed combustion phase.

Rate of pressure rise variation

Rate of pressure rise (ROPR) was calculated by differentiating the cylinder pressure w. r. t. crank angle degrees. ROPR is an important parameter, which gives information about the force transfer rate because of combustion gas pressure to the mechanical linkages such as piston, camshaft, *etc*. and has a direct bearing on engine's structural safety and integrity. Figure 9 shows the ROPR for different SOI timings.

ROPR reaches its maxima during the premixed combustion phase due to larger heat release occurring in very short time interval. After reaching its maxima, it drops continuously to a lower value in the expansion stroke in the mixing controlled combustion phase. For advanced SOI timings, ROPR is much higher than that for retarded SOI timings. The reason is longer ignition delay for advanced SOI timings. Due to this, higher fuel quantity gets accumulated in the combustion chamber during ignition delay period. Thus there is formation of more premixed charge quantity, which is burnt in the premixed combustion phase and as a result, higher ROPR is experienced. The peak of maximum ROPR shifts away from TDC also (into the compression stroke) because of relatively slower burning in predominantly mixing controlled combustion phase at higher engine loads.

Increased cylinder temperature reduces ignition delay. This leads to relative earlier SOC of the air-fuel mixture. Because of earlier combustion of premixed charge, there will be lesser fuel quantity accumulated in the combustion chamber, which will result in lower ROPR with increasing engine load. The peak of maximum ROPR shifts away from TDC also (into the compression stroke) because of relatively slower burning in predominantly mixing controlled combustion phase at higher engine loads.

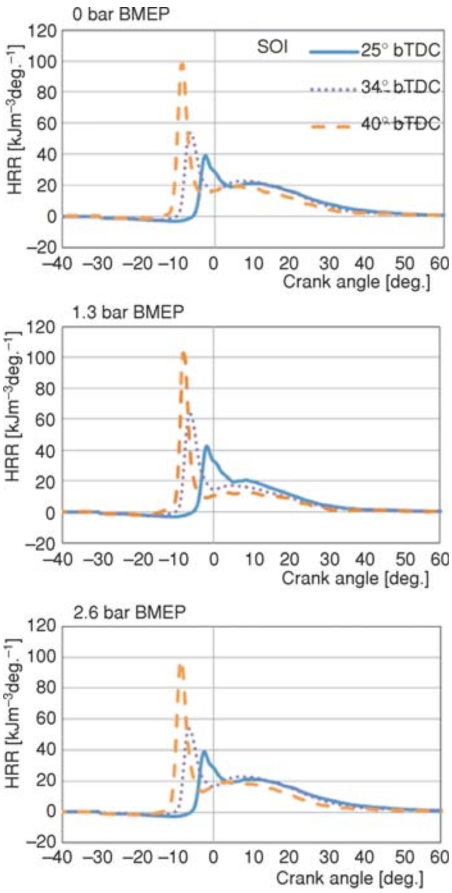

Figure 10. HRR variation with engine load for varying SOI timings

Heat release rate variation

From the cylinder pressure data, heat release rate (HRR) ($\mathrm{d}Q(\theta)/\mathrm{d}\theta$), was calculated by the first law of thermodynamics according to eq. (1):

$$\frac{\mathrm{d}Q(\theta)}{\mathrm{d}\theta} = \frac{1}{\gamma - 1}\left(V(\theta)\frac{\mathrm{d}P(\theta)}{\mathrm{d}\theta} + \gamma P(\theta)\frac{\mathrm{d}V(\theta)}{\mathrm{d}\theta} \right) \quad (1)$$

where $P(\theta)$ and $V(\theta)$ are cylinder pressure and cylinder volume at θ rotation of crank shaft. γ is the polytrophic coefficient for combustion products.

Figure 10 shows HRR for various SOI timings. Figure 10 indicates two different stages of heat release. The first phase is from the SOC to the point where the HRR sharply drops. This is due to ignition of premixed charge combustion in ”premixed combustion phase”.

The second phase starts from the end of the first phase to the end of combustion called “mixing controlled phase”. This phase is dominated by the ignition of air-fuel mixture in the mixing zone and the rate of combustion is controlled by rate of mixing of fuel and air to form combustible mixture. The HRR peak is higher for advanced SOI timings as compared to retarded SOI timings because of higher fuel fraction being combusted in premixed combustion phase for advanced SOI timings. This also explains the cylinder pressure and ROPR curve trends observed earlier.

As engine load increases, maximum HRR decreases due to increase in heat release during mixing controlled combustion phase because the ignition delay reduces due to rise in cylinder temperature. The crank angle position of the maximum HRR shifts towards TDC because as the engine load increases, fuel quantity injected also increases, hence the magnitude of premixed combustion phase increases relatively.

Mass fraction burn rate variation

Figure 11 shows the crank angle position for 5 and 95% mass fractions burnt (MBF) for varying SOI timings. Advanced SOI timings show earlier MBF compared to retarded injection. As the engine load increases, the crank angle position shifts away from the TDC due to increase in BSFC.

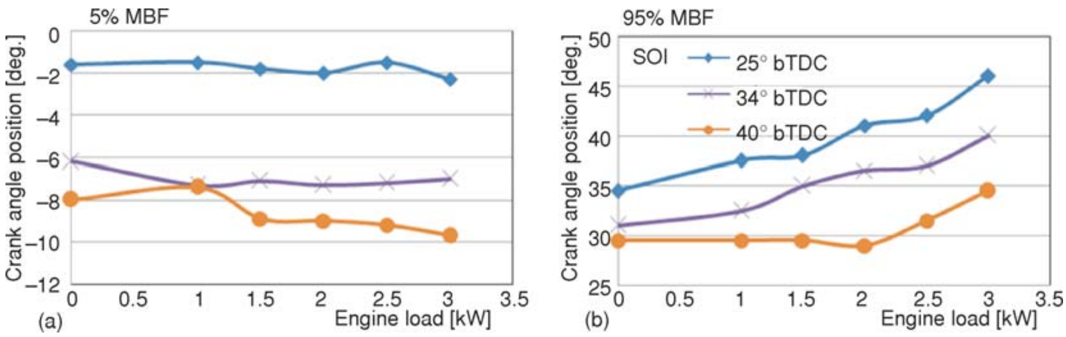

Figure 11. (a) Start and (b) end of combustion (5% and 95% MBF) with engine load for varying SOI timings

Timing for 5% MBF can be taken as SOC and 95% MBF can be taken as end of combustion (EOC) therefore the difference between these two timings is regarded as combustion duration. Combustion duration increases as the engine load increases due to increase in the fuel quantity injected.

Conclusions

A common rail fuel injection system was deployed for a genset engine and start of injection (SOI) timings were optimized at constant engine speed of 1500 rpm. From the experiments, it was observed that the fuel injection timings have a significant influence on the engine performance, combustion, and emissions characteristics. Results of performance characteristics show that SOI timing of 34º bTDC results in lowest BSFC among the three investigated SOI timings at all engine loads. Advanced SOI timings also lead to an earlier SOC. Cylinder charge is compressed as piston moves towards the TDC and attains relatively higher temperature and pressure for advanced SOI timings thus lower BSHC and higher $BSNO_x$ emissions are observed. The smoke opacity increases with retarded SOI timings due to reduction in the peak combustion chamber temperatures. BSCO emissions decrease with advanced SOI timings.

Injection timings have a great influence on the engine combustion characteristics also. Advanced injection timings showed higher peak cylinder pressure, rate of pressure rise (ROPR), and heat release rate (HRR) due to relatively longer ignition delay periods under these conditions. As engine load increases, the premixed combustion phase shortens due to reduction in ignition delay, consequently mixing controlled combustion phase starts dominating. For retarded

SOI timings, due to late SOC, peak cylinder pressures occur later in the expansion stroke. In summary, this low cost CRDI system can be deployed for optimizing the engine performance and emissions from genset engines.

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