Experimental Investigations on the Effects of Low Compression Ratio in a Direct Injection Diesel Engine

M. Vivegananth, K. Ashwin Kanna, and A. Ramesh

1 Introduction

Diesel engines are based on the concept of compression ignition. They rely mainly on the high temperature achieved during the compression stroke for autoignition of the injected diesel. So, higher the compression ratio, better is the cold starting ability. However, high compression ratios lead to many disadvantages like bulky and heavy engine components, high friction, low rated engine speed, and high NO_x and soot emission levels. On account of these conflicting requirements, optimization of the compression ratio is one of the major challenges often faced by designers, Gardner and Henein (1988). In order to achieve maximum benefit at all engine operating conditions, variable compression ratio operation in diesel engines has been explored. However, the complexity of this method makes it impractical for production engines.

At present, considerable research is being carried out in order to achieve very low NO_x and soot emissions. In diesel engines, reducing the compression ratio is one of the promising ways to meet these stringent demands. While reducing the compression ratio, the in-cylinder gas temperature decreases, which in turn reduces the thermal nitric oxide (NO) formation. It also increases the ignition delay, hence leading to better fuel-air mixing and thus more fuel burns in the premixed phase of combustion which leads to low smoke. Beatrice et al. (2008) and MacMillan et al. (2012) showed that soot/NO_x trade-off was improved in low compression ratio diesel engines. On the other hand, due to incomplete combustion, a significant increase in CO, HC emission, and fuel consumption was also observed. The injection timing can be advanced in a low compression ratio engine, because of

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its low peak cylinder pressure when compared to the conventional engines. Hence, the drop in thermal efficiency in a low compression ratio engine can be addressed. Cursente et al. (2008) reported a 12% increase in brake power at 4000 rpm by changing the compression ratio from 18:1 to 14:1, because of advanced fuel injection timing in which the combustion center occurred close to TDC.

Another major issue in operating a low compression ratio diesel engine at high loads is combustion noise. At high loads the drop in cylinder air temperature during fuel evaporation was observed to be considerable, and it increased the ignition delay period which led to high combustion rates. Suh (2011) showed that for the same peak pressure, the heat release rate could be reduced by about 47% with two pilot injections, when compared to the single injection at a compression ratio of 15.3:1.

Though researchers are trying to lower compression ratios to about 14:1, poor cold starting ability and warm-up stability are issues that are to be solved. Diesel engine cold start problems include long cranking time, combustion instabilities, and high emissions. Figure 1 shows the various parameters that can affect the starting of a diesel engine. One of the main challenges in cold starting is to understand the various thermodynamic processes during engine cranking. Liu et al. (2003) used a thermodynamic simulation model to study the key parameters that affect the cranking time and combustion instability during idling. They found that accumulated fuel in the combustion chamber during misfiring cycles has a major impact on engine cold starting. Further, cranking speed should be at an optimum level for effective cold starting, because lower speed causes high heat transfer and blowby losses, whereas time available for evaporation is reduced in case of higher cranking speeds.

Henein et al. (1992) and Han et al. (2001) investigated combustion instabilities during cold starting and found that misfiring is not random but is repeatable. The engine may often skip one or two cycles during starting because the vaporized fuel quantity is not sufficient, due to the slow evaporation rate and the net energy produced from combustion in one cycle not being capable of overcoming the frictional/inertial losses.

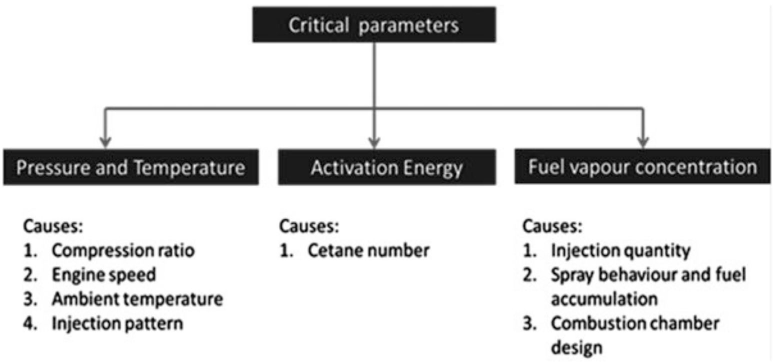


Fig. 1 List of critical parameters affecting the engine starting

Zahdeh (1990) found that the peak compression temperature was decreased by 254 °C, when the ambient temperature was reduced from +20 to −20 °C. So, cold starting becomes much more challenging in a low compression ratio diesel engine at very low ambient conditions. Pacaud et al. (2008), improved the cold starting ability of a low compression ratio diesel engine with the aid of a glow plug and by the use of multiple pulse fuel injection techniques. They found that the pilot injection of diesel promoted the cold flame combustion reaction which in turn reduces the ignition delay. However, the pre-glowing duration (time required for the glow plug tip to reach 800 °C) will increase drastically at low ambient temperatures and, hence, leads to long cranking time. In order to improve warm-up stability, additional methods to trap hot exhaust gases were needed. Peng et al. (2008) found that recirculating fuel-rich exhaust gases back into the cylinder through the intake manifold reduced ignition delay and improved combustion stability. Added advantage of recirculating exhaust gases is the reduction of HC emission (white smoke) during engine warm-up.

In order to implement suitable cold starting strategies, a proper understanding of the transient behavior of an engine during starting is essential. Thus, this work is aimed at understanding the advantages in performance and emissions and also the challenges in cold starting a low compression ratio diesel engine.

Nomenclature

CR	Compression ratio (–)
NO	Nitric oxide (ppm)
HC	Hydrocarbon (ppm)
CO	Carbon monoxide (% volume)
BMEP	Brake mean effective pressure (bar)
IMEP	Indicated mean effective pressure (bar)
TDC	Top dead center (TDC)
degCA	Degree crank angle (degree)
P	Cylinder pressure (bar)
V	Displacement volume (m^3)
n	Polytropic constant (–)
Q	Heat release rate (J/degCA)
SOI	Start of injection

2 Experimental Facility

The schematic arrangement of the experimental setup is shown in Fig. 2. A 0.55 L single cylinder diesel engine was used for this experimental study. Detailed specifications of the engine are provided in Table 1. The geometric compression ratio was modified from 16.5:1 (as in the production engine) to 15:1 and 14:1 progressively by increasing the volume of the piston bowl and also maintaining its shape similar to the original as shown in Fig. 3. Hereafter, these compression ratios will be referred to as CR16.5, CR15, and CR14, respectively. The engine was coupled to an

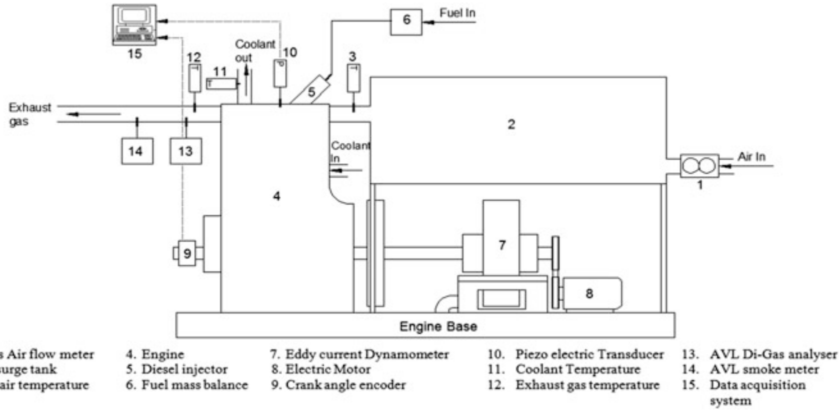
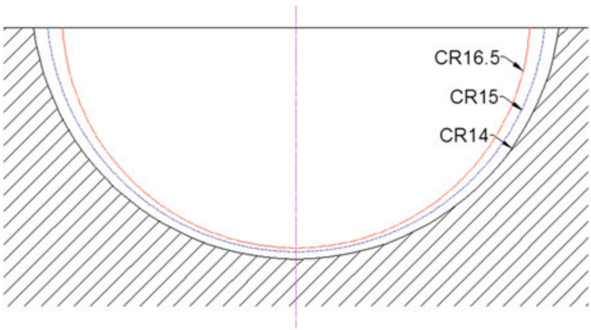


Fig. 2 Schematic layout for the single cylinder diesel engine experimental setup

Table 1 Engine specifications

Engine type	Single cylinder, water cooled, diesel engine
Bore × stroke	80 × 110 mm
Displacement volume	553 CC
Compression ratio	16.5, 15 and 14
Connecting rod length	231 mm
Max. torque and power	23.5 Nm at 1500 rpm 3.7 kW at 1500 rpm
Fuel injection system	Mechanical (NOP = 220 bar)

Fig. 3 Piston bowl profile for three compression ratio configuration



eddy current dynamometer for loading and to maintain its speed. The airflow rate to the engine was measured using a positive displacement-type airflow meter (make, Dresser, model, Roots Series B3). Fuel consumed by the engine was measured directly on the mass basis. Exhaust gas emissions (HC, CO, and NO) were measured using a NDIR-based (AVL Di-gas 444) portable analyzer, while an AVL 415S smoke meter was used for smoke measurements. K-type thermocouples were used to measure the intake air and exhaust gas temperatures. A resistance temperature detector was used to measure the outlet temperature of the coolant.

In-cylinder pressure was measured using a flush-mounted piezoelectric pressure transducer (make, Kistler, model, 6043A60) along with a charge amplifier. An optical encoder was used to determine the position of the crankshaft. A high-speed data acquisition system (NI data acquisition card 6070E) with in-house developed software was used to record the in-cylinder pressure on the crank angle basis. An average of 100 consecutive cycles of cylinder pressure data was used for the calculation of heat release rate. The heat release rate was determined through a first law analysis of the cylinder pressure data as given below:

$$\frac{dQ_n}{d\theta} = \frac{n}{n-1} P \frac{dV}{d\theta} + \frac{1}{n-1} V \frac{dP}{d\theta} \tag{1}$$

Engine was maintained in the required ambient temperature through cold air and coolant conditioning systems. Engine cooling was achieved by circulating chilled water in to the engine coolant jacket. Cold air was supplied through a refrigeration system mounted on the engine intake. Intake air temperature was controlled by adjusting the refrigerant temperature in the evaporator coil of the air conditioner. The engine was also equipped with a starter motor and a 12 V cranker (rectifier which converts 220 V AC to 12 V DC) for starting. To maintain constant cranking condition and to ensure repeatability during starting, the cranker was used instead of a battery. Another in-house developed data acquisition software was used for acquiring the instantaneous cylinder pressure and engine speed data of first 100 cycles at 1° intervals during engine starting. Starting was considered to be successful if the engine fired and accelerated to the idle speed. As mentioned earlier, engine cranking time will vary for different compression ratios and different ambient temperatures, so it was necessary to control the starter motor in order to disengage it once the engine operation became stable. A control system was developed using a microcontroller for this application. This system measures the engine speed and disengages the starter motor when it crosses the set threshold; at this condition it was considered that the engine attains stability and has started accelerating steadily toward the idling speed. The start ability experiments were performed in the following order: First, the data acquisition was started; then it triggers the starter motor control system to crank the engine; this control system disengages the starter motor if the engine crosses the threshold speed of 700 rpm.

3 Results and Discussions

Experiments were initilally conducted at different compression ratios under different constant injection timings, while the load (brake mean effective pressure – BMEP) was varied. Parameters like brake thermal efficiency, cylinder pressure, heat release rate, and emissions were obtained under steady operating conditions. These are reported and discussed to evaluate the influence of compression ratio.

Subsequently experiments were conducted to evaluate the startability under different ambient temperatures that were simulated as explained earlier.

3.1 NO and Smoke Emissions

Figure 4 indicates the variation of NO emissions under different BMEPs at three compression ratios with a static injection timing of 23°bTDC. The nitric oxide (NO) emission decreased with a reduction in the compression ratio. This was mainly due to the low peak in-cylinder temperatures reached after combustion which were influenced by the lower charge temperatures at the end of compression stroke with reduced compression ratios. At BMEPs lower than 3 bar, reduction in NO levels was significant with respect to reduction in compression ratios. However, at higher BMEPs, NO levels were higher in case of CR14 than CR15 due to the higher rate of combustion. This is explained in detail later with heat release rate data. Figure 5 shows that smoke levels get reduced with a reduction in the

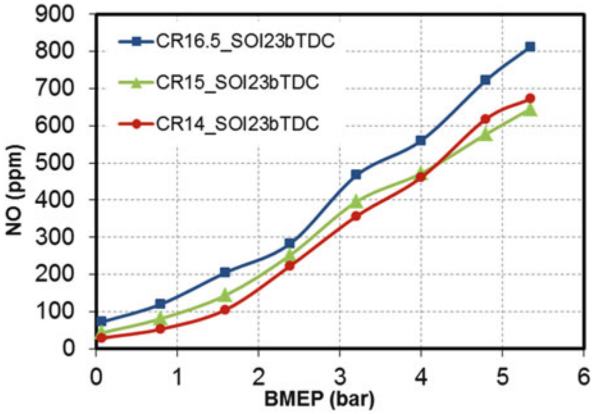


Fig. 4 Variation of nitric oxide emissions with load

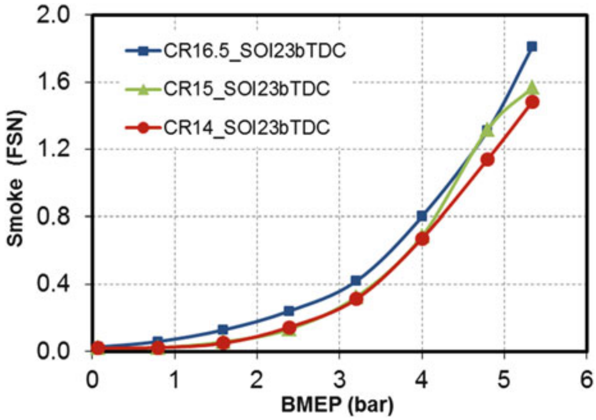


Fig. 5 Variation of smoke emissions with load

compression ratio. This is due to increase in the premixed phase of diesel combustion, i.e., with a reduction in the compression ratio, the ignition delay is longer and majority of injected diesel is burnt in the premixed combustion phase. The higher ignition delay in the case of low compression ratios thus leads to improved fuel-air mixing. However, the lower charge temperatures at low compression ratios will also affect fuel vaporization and mixing under these conditions.

3.2 Combustion Characteristics

Brake thermal efficiencies at the three compression ratios were almost similar as shown in Fig. 6. At higher BMEPs (85% and 100% load), a marginal drop in efficiency of about 1% was observed with CR14. Figure 7 shows the variation in peak cylinder pressure with load for various compression ratios. It was observed that for same BMEP the peak cylinder pressure in low compression ratio engines is

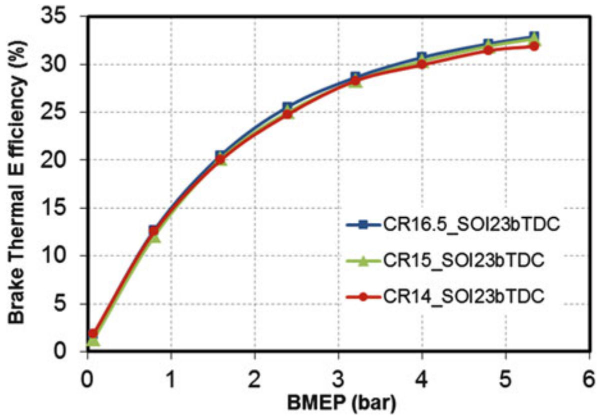


Fig. 6 Variation of brake thermal efficiency with load

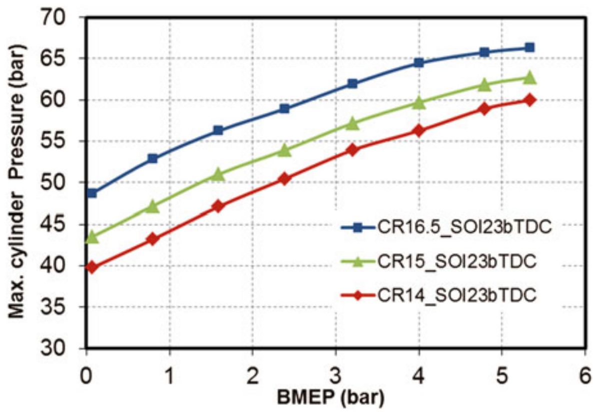


Fig. 7 Variation of peak cylinder pressure with load

reduced. This reduced in-cylinder pressure leaves scope for improving the brake thermal efficiency by advancing the fuel injection timing and also by reducing engine friction through reduction in the size and weight of other engine components.

The variation of heat release rate at fixed BMEPs of 5.3 bar and 2.4 bar under a constant static injection timing of 23°bTDC is depicted in Figs. 8 and 9. It is observed that combustion gets retarded as the compression ratio is reduced. This is because of the low cylinder pressure and temperature during the compression stroke. At high BMEPs (>4 bar), the combustion rate is high in CR15 and CR14 because of the high ignition delay caused due to the drop in cylinder temperature during fuel evaporation. This leads to high local temperatures inside the cylinder during combustion which in turn results in high NO emissions and high combustion noise. At low BMEPs (refer to Fig. 9), the start of combustion gets retarded

Fig. 8 Effect of reducing compression ratio on heat release rate (HRR) at 5.3 bar BMEP

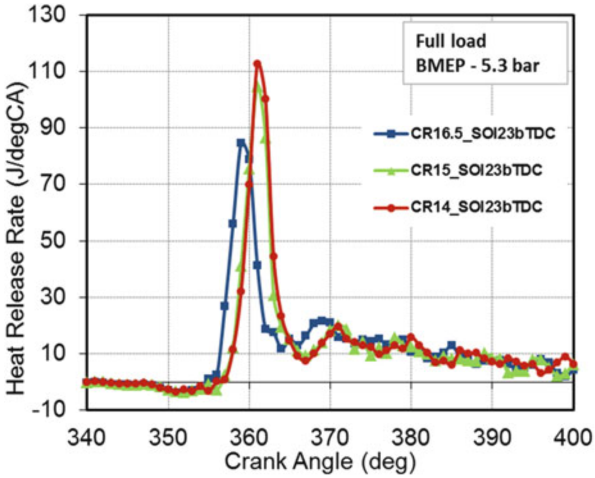
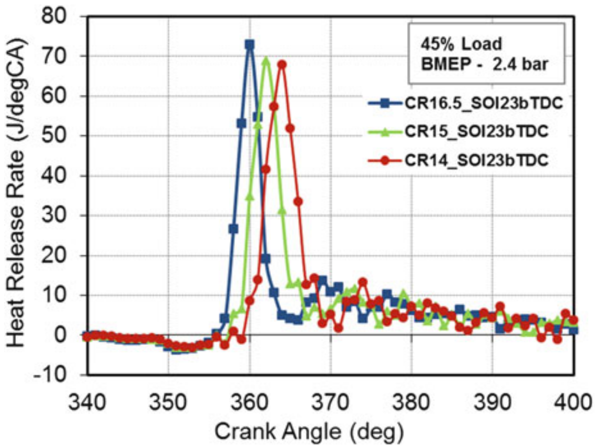


Fig. 9 Effect of reducing compression ratio on heat release rate at 2.4 bar BMEP



(ignition delay is increased) in CR14 and CR15 because of low in-cylinder pressure and temperature. However, the peak heat release rate is lower at lower compression ratios because of the low temperatures in the cylinder that affect mixture preparation during the ignition delay period. This leads to low NO and smoke but affects effective expansion of the combusted gases.

3.3 Cold Starting

The effect of lowering the ambient temperature on engine startability was evaluated in subsequent experiments. Startability of the engine at two compression ratios, namely, CR14 and CR16.5, was evaluated at 10 °C, 15 °C, 20 °C, and 28 °C. Figures 10 and 11 show the instantaneous engine speed and cylinder pressure plots for the first 30 cycles during engine cranking and starting. At the highest intake temperature of 28 °C, there was no difference in startability between the two compression ratios as seen in Fig. 10. Combustion occurred right from the first cycle which is inferred from the cylinder pressure plot. The engine started accelerating right from the first cycle and reached the governor-controlled idle speed after 25 cycles. The initial peak pressures were higher with CR16.5 as compared to CR14. Cylinder pressure plot shown in Fig. 10 is at the starting condition.During starting, the fuel injection pump of this engine injects more fuel than at full load. Hence, this leads to high cylinder pressures (Pmax) during starting.

At the intake temperature of 10 °C (Fig. 11), the engine with compression ratio of 16.5 (CR16.5) fired in all cycles from the beginning, and this shows that the engine startability did not deteriorate while reducing the ambient temperature from 28 to 10 °C. However, in the case of CR14, the engine misfired in the first 6 cycles, and the first firing occurred only at the seventh cycle because of the fuel accumulated during the previous cycles.

Combustion instabilities were there till the first 20 cycles. The engine fired in every other cycle or every third cycle till it attained stability. After the twentieth cycle, the engine was stable and accelerated steadily because of the reduced heat transfer and blowby losses at high engine speeds and slightly warm engine walls. For CR14 at ambient temperature of 10 °C, the engine took 36 cycles to reach the governor-controlled idle speed of 1500 rpm.

In order to determine whether the misfiring cycles that occurred in between the firing cycles are due to irregularities in injection or actual lack of combustion, heat release rates were obtained for the fired and misfired cycles and also for a cycle where the fuel injection was completely absent. This is seen in Fig. 12. We see that in the fired cycle, the heat release rate (HRR) shows a sharp positive peak. It also shows a negative portion after fuel injection as indicated in Fig. 12. This is due to vaporization of the fuel accumulated before ignition. This negative portion is not seen in the cycle where fuel injection is not present. In the misfiring cycle, we see that the negative HRR portion is present, but the positive HRR portion is absent indicating that fuel was injected but combustion was not initiated in the misfired

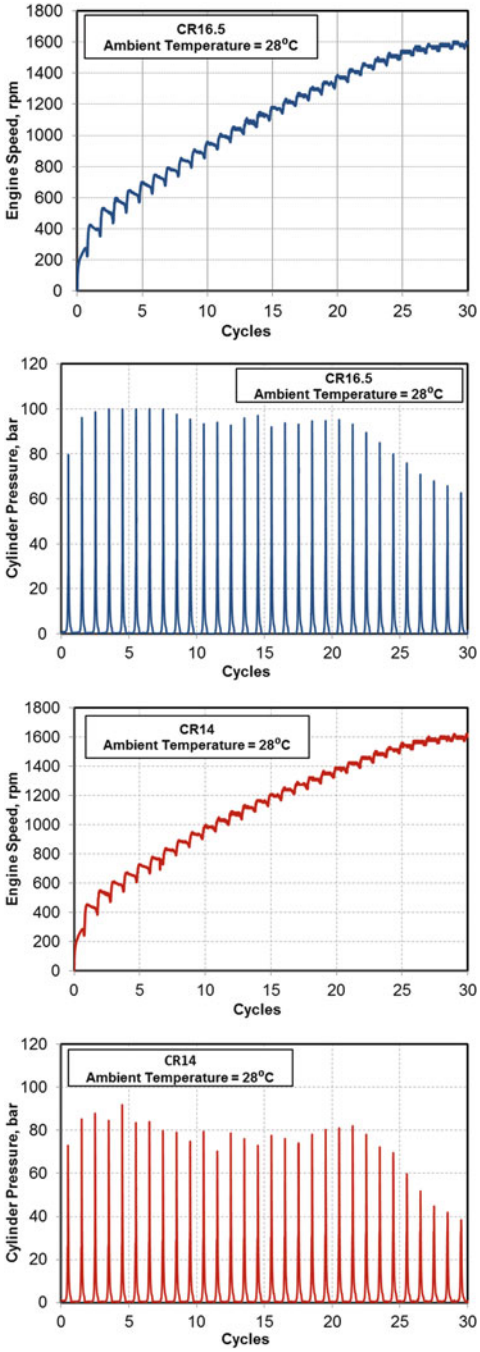


Fig. 10 Engine startability trials at 28 °C ambient temperature

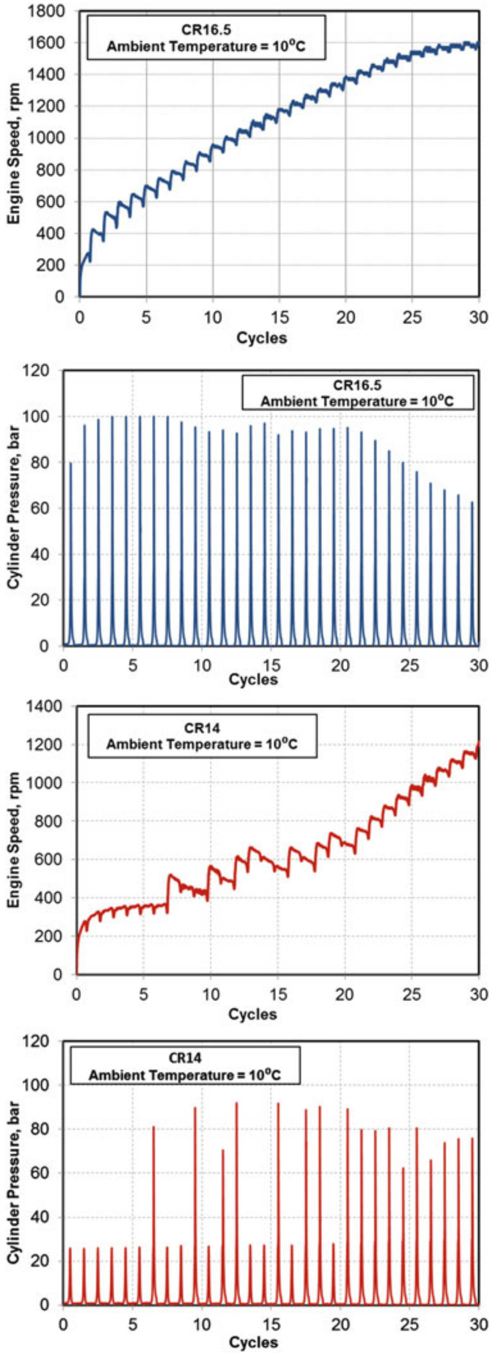


Fig. 11 Engine startability trials at 10 °C ambient temperature

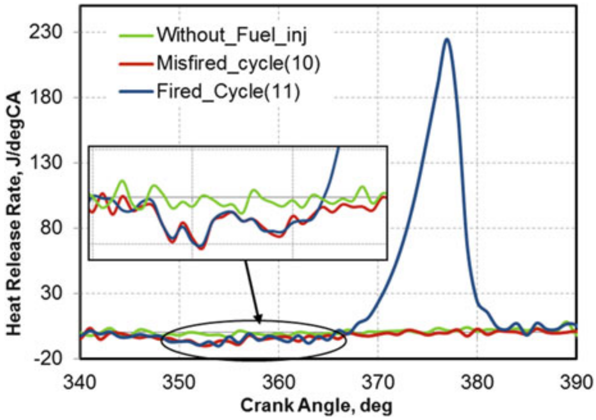


Fig. 12 Rate of heat release plot during cranking with CR14 at 10 °C

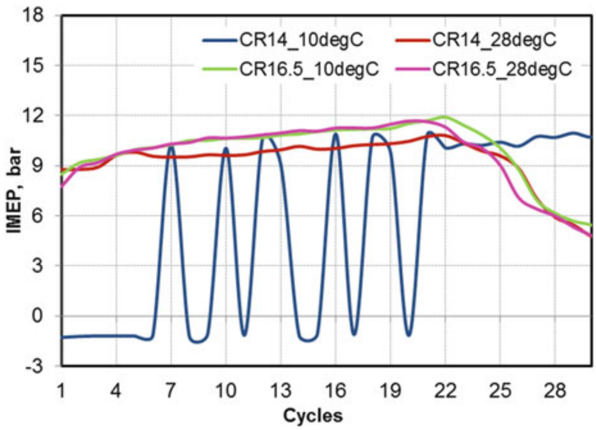


Fig. 13 Indicated mean effective pressure (IMEP) plot during cranking at different ambient temperatures

cycle. Hence, some of the cycles even after a firing cycle misfire under cold start conditions in the case of CR 14. This could be because during cold start the concentration of fuel vapor is insufficient for ignition. Hence, repeated injection of fuel in consecutive cycles that misfire raises the vapor concentration and aids ignition in a following cycle. It is seen that the sequence of firing and misfiring cycles is not entirely random. Such observations have also been reported in literature [Henein (1992)]. It may also be noted that the heat release shows only premixed combustion. This also indicates that only the fuel that is vaporized participates in combustion during the initial cycles of cold starting.

The severe IMEP fluctuations seen at starting in CR14 at 10 °C are indicated in Fig. 13.

Figure 14 shows how the starting delay (number of cycles to reach idling speed) of the engine increases with reduced ambient temperatures. It is evident that even below ambient temperatures of 15 °C, reduction in the compression ratio significantly affects startability. This will also have a significant influence on emissions during starting.

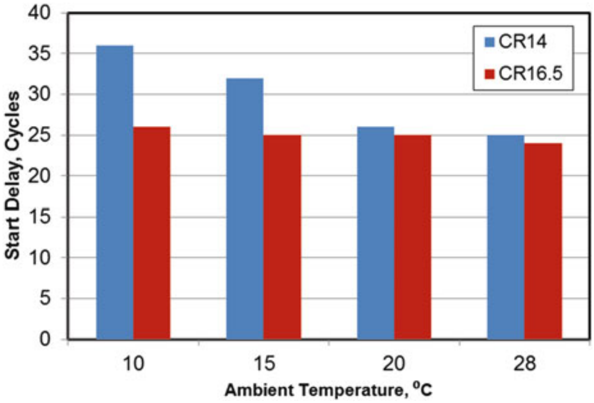


Fig. 14 Start delay for CR16.5 and CR14 at different ambient temperatures

4 Conclusions

Based on the experimental investigations on a direct injection diesel engine with three different compression ratios (CR14, CR15, and CR16.5), the following conclusions were made.

(a) Reducing the compression ratio in the diesel engine reduces the nitric oxide (NO) emissions at all load conditions by reducing the peak in-cylinder temperature. Particularly significant reductions are seen at lower loads (BMEP $<$ 3 bar) where the combustion rate is minimum. At a BMEP of 1.6 bar, the reduction in NO with CR14 and CR15 was 49% and 30%, respectively, as compared to CR16.5. Smoke at all load conditions is also reduced at low compression ratios due to longer ignition delay. At a BMEP of 1.6 bar, the reduction in smoke with CR14 and CR15 was 62% and 55%, respectively, as compared to CR16.5.
(b) Brake thermal efficiency was not significantly affected with reduction in compression ratio.
(c) Starting delay of the engine with CR14 was increased at reduced ambient temperatures, because of misfiring (combustion failure) during the intial cycles when the engine is cold.
(d) During cold starting only the fuel that is vaporized participates in combustion and the remaining gets accumulated in the bowl. Repeated injection of fuel in consecutive misfiring cycles raises the fuel vapor concentration and aids combustion in the following cycle.

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