

The Effect of Reducing Compression Ratio on the Work Output and Heat Release Characteristics of a DI Diesel under Cold Start Conditions

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ABSTRACT

An experimental investigation has been carried out to compare the indicated performance and heat release characteristics of a DI diesel engine at compression ratios of 18.4:1 and 15.4:1. The compression ratio was changed by modifying the piston bowl volume; the bore and stroke were unchanged, and the swept volume was nominally 500cc. The engine is a single cylinder variant of modern design which meets Euro 4 emissions requirements. Work output and heat release characteristics for the two compression ratios have been compared at an engine speed of 300 rev/min and test temperatures of 10, -10 and -20°C. A more limited comparison has also been made for higher speeds representative of cold idle at one test temperature (-20°C). The reduction in compression ratio generally produces an increase in peak specific indicated work output at low speeds; this is attributable to a reduction in blowby and heat transfer losses and lower peak rates of heat release increasing cumulative burn. High compression however extends the range of injection timings where useful work output can be generated. The coefficient of variation of work output increases with increasing engine speed and decreasing loading, especially at low compression ratio, giving rise to potentially poor idling quality.

INTRODUCTION

There is a design trend towards using lower compression ratios for light duty DI diesel engines for performance and emissions benefits, including reduced NO_x and BSFC [1-3]. One potential limit on how much the compression ratio can be reduced is a deterioration of cold start performance associated with lower work input during compression. Also the cold start behaviour of diesel engines is prone to deteriorate as ambient temperature is reduced [4-6]. At warm ambient temperatures, start times are typically one or two seconds. Under extreme winter ambient conditions start times may extend to tens of seconds. This is a reflection of how engine friction work increases rapidly as temperature falls whilst it is

difficult to find measures which will raise the indicated work output. Indeed, indicated work output at engine speeds between cranking and idling is often relatively low as a result of poor combustion efficiency and high blow-by and heat transfer losses. The importance of engine friction at low temperatures was investigated by Shayler et al. [7]. Hengqing et al. [8] developed a model of the cold starting process which correlated well with experimental results in predicting multiple misfiring cycles at low temperatures due to high blowby and heat transfer as well as rapidly varying engine speeds. In this paper an experimental investigation has been carried out to compare the indicated performance and heat release characteristics of a DI diesel engine at compression ratios of 18.4:1 and 15.4:1. Simple calculations based on motored pressure traces at these levels indicate a reduction of compression work input in the order of 8.5%. The engine is a single cylinder variant of modern design which meets Euro 4 emissions requirements. This has an HPCR fuel injection system with solenoid injectors. The swept volume was nominally 500cc. Work output and heat release characteristics for the two compression ratios have been compared at an engine speed of 300 rev/min and test temperatures of 10, -10 and -20°C. A more limited comparison has also been made for higher speeds typical of cold idle to assess any degradation in the idle regime at one test temperature of -20°C. The focus of the work has been the first few firing cycles of the engine at any given test condition; these have most influence on start-to-idling times. Pilot and main fuel injection timings have been varied; the glow plug pre-heat time was set to match service settings. The reasons for the trends observed in the results are described and their physical significance discussed. Effects on smoke and other pollutant emissions have not been examined. Although European regulations set limits on cold start HC and CO emissions at -7°C, the key concern in the current work has been the influence of compression ratio on indicated work characteristics because these directly influence start-up behaviour.

EXPERIMENTAL SETUP

All experiments were performed on a single cylinder variant of a production Ford four cylinder, naturally aspirated, 2.0 litre direct injection diesel engine. The cylinder head, piston and crankshaft components were fitted in a Ricardo Hydra unit. This retained the same bore and stroke, which were both 86mm, and the same cylinder displacement of 0.49956 litres. The initial compression ratio was measured to be 18.4:1 (High Compression Ratio, referred to as HCR). For reduced compression testing, a piston was manufactured with the same bowl geometry but to a larger scale. This reduced the compression ratio to 15.4:1 (Low Compression Ratio, LCR). The engine had four valves per cylinder and a centrally mounted, six-hole solenoid injector. Diesel was supplied to the injector via a low pressure fuel pump feeding a 1350bar Bosch high pressure pump delivering fuel to the fuel rail. The engine was soaked down to test temperature by circulating chilled coolant through the block, head and fuel tank. The principal method of cooling was the direct chilling of coolant by a 5.8kW cooler, capable of chilling to -30°C. An 80 litre plenum before the intake was also chilled using an atmospheric cooling radiator to provide a reservoir of low temperature intake air. The fuel used throughout these tests was a reference winter fuel to avoid paraffin waxing [Cetane Index >47, T90 <325°C]. Pressure transducers and thermocouples were used to monitor operation. Cylinder pressure was measured using a Kistler 6125B quartz pressure sensor rated to 250 bar peak pressure. An optical shaft encoder with half degree crank angle resolution outputs 720 DC pulses per engine revolution (1440 per cycle); this signal is the trigger for much of the dSPACE data acquisition system which captures system variables at each half crank angle step. ATI Vision was used to isolate the engine control unit and give full control over injection parameters.

The form of test used was intended to reflect the low speed, cold-surface conditions of start up with better control over variability. The glow plug was the same steel-sheathed design used in production; this has an operating surface temperature of around 850°C. The glow plug was switched on for a pre-heat period of 4 seconds and remained on throughout the test. The engine was motored for 10 cycles at 300rpm, by which time both engine speed and fuel rail pressure had stabilised to their respective set point values. The speed was chosen to be between engine cranking and idling speeds. On cycle 11, fuel injection was enabled and injection quantities and timings were held constant for the following 30 cycles. The dynamometer was used to maintain the set point engine speed throughout the test.

EXPERIMENTAL RESULTS AND DISCUSSION

The potential of HPCR fuel injection strategies involving multiple injections to improve cold starts was demonstrated by Brown et al. [9], who observed that IMEPg was maximised when fuel injection was split into two parts, as a pilot injection followed by the main

injection after a small separation. It is unclear from this previous work how sensitive the output is to injection strategy when compression ratio is changed. In this investigation the results for single and pilot-plus-main injection patterns are presented. In the pilot plus main pattern, the pilot was fixed at 2mm³/str and the pilot to main separation was 5°CA. Total fuelling remained constant by reducing main injection by 2mm³/str if a pilot was present. Two fuel levels were selected, both above and below stoichiometric. It is typical to run with excess fuelling at cold start to overcome problems with combustion initiation and ensure a quick run-up to idle [9,10]. For tests at -10°C, an overall equivalence ratio of one (based on fuel delivered) is reached with a fuel delivery of around 45mm³/str. To straddle this, tests were run at fuelling levels of 30mm³/str and 60mm³/str. The calculated values of overall equivalence ratio at these fuel settings for each of the tests temperatures are shown in Table 1. The test temperature was taken to be the initial temperature of the cylinder liner. Intake air temperature was a few degrees lower than liner temperature at the end of a cold soak, down to -25°C for the -20°C liner case.

Temperature	Φ 30mm ³	Φ 60mm ³
10°C	0.71	1.41
-10°C	0.64	1.29
-20°C	0.62	1.24

Table 1 – Overall equivalence ratios at three temperatures and two injection quantities

Injection timing sweeps for 30mm³/str covered the range of main timings from 25°BTDC to 25°ATDC. For the 60mm³/str fuelling case, timing advance was restricted by a safe limit on peak cylinder pressure of 120 bar. The maximum advance was TDC at HCR and 5°BTDC at LCR. The principal measurement of cold start performance for these timing sweeps was the average gross Indicated Mean Effective Pressure over the first five injected cycles (IMEPg₅).

The results presented in Figure 1 are for the high compression ratio case. The results show the effect of injection timing on work output for a single injection at the three test temperatures. For both 30mm³/str and 60mm³/str the highest IMEPg₅ values are achieved with main injection timing close to TDC. The peak value of IMEPg₅ increases with decreasing test temperature for the 60mm³/str fuelling. The increase is much less marked for the 30mm³/str fuelling case. At well retarded timings, the trend is reversed for both fuelling levels.

The corresponding results for the low compression case are presented in Figure 2. The variations of IMEPg₅ with injection timing and the sensitivity to test temperature are similar to the HCR case but, for the 60mm³/str fuelling, IMEPg₅ is higher by approximately 2 bar. In effect, doubling the quantity of fuel doubled the IMEPg₅ output for the LCR case whilst the increase was much less for the HCR case. There is a minor reduction in peak IMEPg₅ injecting 30mm³/str at LCR. The introduction of a pilot injection

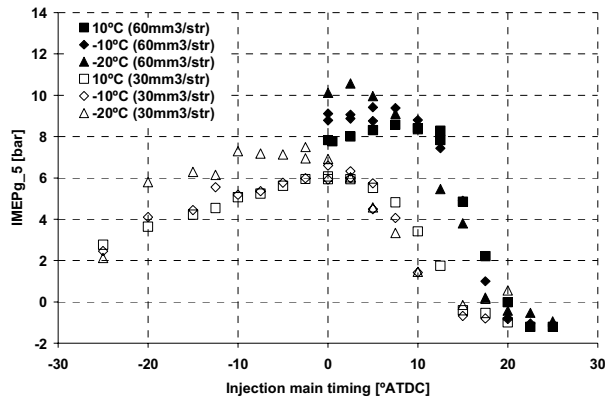


Figure 1. IMEP_{g_5} obtained at 300rpm using a single main Bq=30mm³/str and Bq=60mm³/str at three different temperatures, HCR

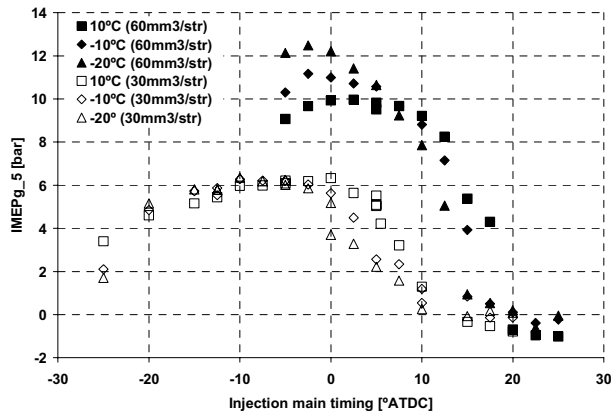


Figure 2 - IMEP_{g_5} obtained at 300rpm using a single main Bq=30mm³/str and Bq=60mm³/str at three different temperatures, LCR

before the main has a similar, small effect at both high and low compression ratios. The test results for the pilot plus main injection strategy are presented in Figures 3 and 4. There is very little effect on maximum work output. At the 60mm³/str quantity, pilot injection actually decreases the maximum IMEP_{g_5} by a small amount for the LCR case. For both compression ratios, the most significant change is a broader range of main timings giving high IMEP_{g_5} values into the retard.

To examine the relationship between rates of heat release and the corresponding cumulative totals for the various cases, the gross heat release rates were calculated using

$$\frac{dQ_{gross}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} + \frac{dQ_{bby}}{d\theta} \quad [11]$$

The energy loss to heat transfer, Q_{ht} , was calculated using the Hohenberg correlation and energy loss to blowby, Q_{bby} , was neglected for the purposes of gross heat release rate calculation [12]. The end of combustion was defined as the point at which the apparent gross heat release rate fell to within 2J/° of that at end of the expansion stroke. The relationship between peak rate of heat release and the cumulative total released by combustion is shown in Figures 5 for

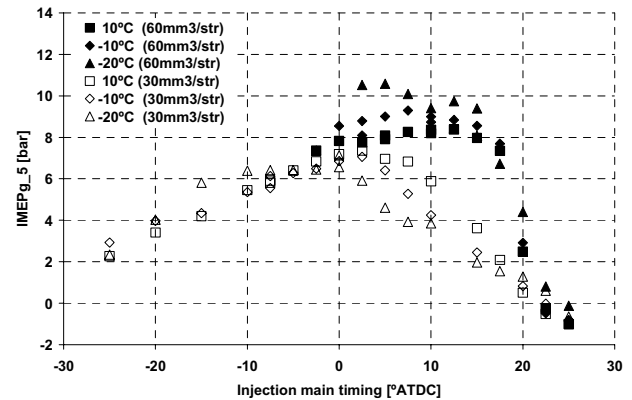


Figure 3 - IMEP_{g_5} obtained at 300rpm using a pilot plus main Aq=2mm³/str Bq=28mm³/str and Bq=58mm³/str at three different temperatures, HCR

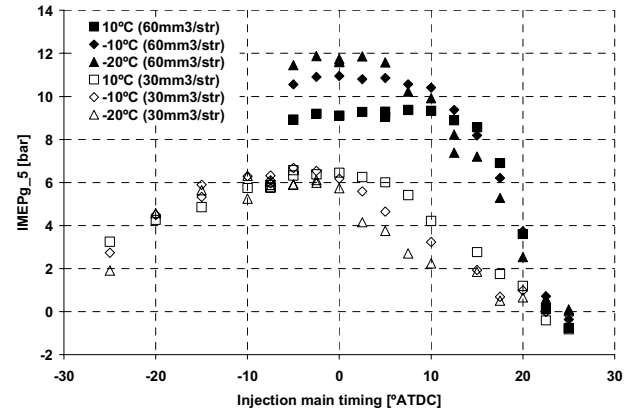


Figure 4 - IMEP_{g_5} obtained at 300rpm using a pilot plus main Aq=2mm³/str Bq=28mm³/str and Bq=58mm³/str at three different temperatures, LCR

the HCR case and in Figure 6 for the LCR case. At both compression ratios, there is a clear correlation between low maximum rates of heat release and a low cumulative total for poor burning cycles which have a maximum rate less than about 50J/°. The increase in cumulative total is of the order of 10J per 1J/° increase in maximum rate. This indicates that for weak burning cycles, the rate of heat release plays an important role in sustaining combustion. However, the cumulative totals level off and become almost independent of the maximum rate of heat release once this exceeds values of 100-200J/°. By implication, for a given fuel quantity injected there are then a range of heat release patterns burning the same fraction of the fuel. In Figure 5, for the high compression ratio case, there is no clear separation of the results for the two fuelling levels examined. In Figure 6, for the low compression case, the separation into two bands of data is obvious.

The conditions giving rise to four points marked on each of Figures 5 and 6 have been examined in more depth. All the points represent first firing cycles for the single injection strategy of 60mm³/str at -10°C and the high and low compression ratio cases are paired to have the same injection timings. Details are summarised in Table 2. The heat release profiles are given in Figures 7-10. For points 1-3, the peak rate of heat release is higher for the HCR case. Injection is