THE EFFECT OF INJECTION PRESSURE AND STRATEGY IN A JAGUAR V6 DIESEL ENGINE

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Abstract

In recent years, the improvement of engine performance and emissions has become an extremely important concern. This study focuses on the injection strategy based on the injection pressure (IP) and duration between pilot injection and the main injection (dMI) using a multi cylinder common rail multiple injections diesel engine. The study was designed to produce improvements in fuel mixing via the injection strategy, to reduce the main ignition delay. This would contribute to a minimum amount of fuel burnt in the premixed combustion phase, leading to a reduction in emissions. Recent evidence shows that premixed combustion is significant in the controlling of emissions of nitrogen oxides (NOx) and soot. Six different IPs combined with a short and long dMI were compared in the attempt to improve engine performance and emissions. The engine performance was measured in terms of brake specific fuel consumption, ignition delay, heat release and peak in-cylinder pressure and emissions, specifically nitrogen oxides (NOx), total unburned hydrocarbons (THC), carbon monoxide (CO) and smoke emissions for each engine test condition. The evidence from this study shows that the effect of IP is more dominant than dMI in terms of peak cylinder pressure, heat release, brake specific fuel consumption and emissions. However, the dMI shows a strong effect at a higher engine speed.

***Keywords**: combustion, emissions, injection strategies, premixed combustion, ignition delay*

1. Introduction

Engine design, air mixture and fuel properties are important parameters for the optimisation of engine performance and emissions [1]. A recent study shows that the emissions are greatly influenced by the spray properties, ignition delay and combustion characteristics of the fuel [2]. It is well known that the emissions from diesel engines are mainly caused by local rich mixture zones even though the overall operation is a lean condition. A large amount of research has been carried out to improve the premixed charge. The objective is to produce a consistent lean mixture both globally and locally inside the combustion chamber to minimise emissions [3-5]. Several studies have revealed that pilot injection strategy produced a shorter ignition delay, contributing to lower emissions [6-8]. Shorter ignition delay leads to a reduction of THC and smoke emissions due to less fuel adhering to the combustion chamber walls. Moreover, shorter ignition delay (up to a point) tends to take advantage of a more homogeneous mixture, resulting in lower emissions of harmful aldehydes [9]. At the same time, it tends to produce lower combustion temperatures, resulting in lower NOx formation. Alternatively, longer ignition delay tends to produce a higher level of THC emissions due to a local over-lean mixture and local non-ignition. Longer main ignition delay contributes to the global increase in bulk temperature and pressure inside the combustion chamber and potentially longer delays can promote knocking in diesel engines due to large proportions of fuel being burnt simultaneously [1].

High pressure fuel injection combined with a split injection strategy (pilot and main) have been shown to reduce engine noise and several engine out emissions, but NOx emissions increase due to increased cylinder pressures and temperatures [9]. Higher injection pressure improves the fuel injection spray characteristics by increasing the rate and depth of penetration into the cylinder. This increases the overall dispersal of an identical fuel mass within a set volume (the cylinder) tending to produce better (more complete) combustion and engine emissions due to a more complete entrainment of the more atomised fuel air charge [10].

A number of research programs have been carried out at the University of Birmingham to investigate diesel engine performance and emissions [11-19]. The present study was designed to determine the effect of injection pressure (IP) (300 bar – 800 bar) combined with the duration between pilot injection and the main injection 5 °CAD (Short) and 40 °CAD (Long) on engine performance and emissions. A multi cylinder common rail direct injection (Pilot + Main Injection) diesel engine was employed using a split injection strategy. (The possibility of a greater number of injection points is recognised). The aim of this study is to induce a better mixture formation by tailoring the injection strategy and IP. It is thought that there is potential in this area as it is known that only 80 percent of the intake air is effectively used in the combustion process. This is due in part to an insufficient period of time for the mixing process to take place [1]. Previous studies have reported that the ignition delay period depends on the quality of the combustible mixture, [9, 20] the air temperature and the size of fuel droplets. Since the ignition delay also plays a major role in combustion behaviour, the development of injection strategy is a major factor in terms of efficiency of combustion and in turn, engine out emissions. Exhaust gas recirculation (EGR) is an effective method of NOx control [21] therefore the injection strategies and various pressures of injection were also subjected to EGR.

2. Experimental Set-up

The experiments were carried out on a fully instrumented multi-cylinder Jaguar V6 diesel engine, common rail multiple direct fuel injection system, twin water-cooled variable geometry turbochargers and cooled EGR. The engine specification is shown in Tab. 1. The fuel injector model is a Siemens 4S7Q-9K546-AD (unmodified, as provided by the manufacturer) with a six-hole nozzle and 156 degrees of cone angle. An eddy-current dynamometer type Schenck W230 and an engine starter motor, are used to load and start the engine respectively. The Schenck series 2000 controller is used to control the dynamometer. The engine test rig has been described in detail in a previous publication [13].

The injections of fuel (pilot and main) had a short and long injection delay between the pilot and the main (fixed) injection 5 °CAD (Short) and 40 °CAD (Long). Therefore, the long injection delay effectively represents a pilot injection timing advance. Hereinafter this is referred to as (dMI) and this stands for the delay before the start of the main injection. The amount of fuel injected for both pilot and main injection was at a constant ratio of 10:90 for pilot and main respectively. All the tests were performed with ultra low sulphur diesel (ULSD) and the fuel properties are given in Tab. 2. The exhaust gas of the engine is passed through the gas analyser via a sample line and NO_x, THCs, CO, and smoke are measured. The exhaust sample acquisition time is approximately 10 seconds at an operating temperature of 28 to 30 °C and the relative humidity is approximately 40-50 percent. The smoke emissions were measured by using an AVL 415S smoke meter which provides results directly as a Filter Smoke Number (FSN) unit.

The effects of variation of IP and dMI, operating with and without cooled EGR, on engine performance and emissions were examined. These experiments consisted of six different levels of injection pressure (300, 430, 500, 600, 700, 800 bar) at two different engine speeds with the same engine loads (1500 rpm, 35.1Nm) and (2250 rpm, 35.1 Nm). The selected conditions have been chosen in order to investigate the effect at low and moderate engine speeds.

Tab. 1. Engine specifications

Bore	81.0 mm
Stroke	88.0 mm
Displacement volume	2720 cm^3
Maximum torque	435 Nm @ 1900 rpm
Maximum power	152 kW @ 4000 rpm
Compression ratio	17.3:1
Connecting rod length	160.0 mm

Tab. 2. Fuel characteristics

Cetane number	53.9
Density at 15°C (kg m−3)	827.1
Viscosity at 40°C (cSt)	2.467
50% distillation (°C)	264
90% distillation (°C)	329
LCV (MJ kg−1)	42.6
Sulphur (mg kg−1)	46
Aromatics Mono (wt%)	21.0
Di (wt%)	3.1
Tri (wt%)	0.3
Total (wt%)	24.4
C (wt%)	86.5
H (wt%)	13.5
O (wt%)	-

3. Engine Test Conditions

The injection pressure was varied from 300 bar up to 800 bar with two different dMI. A short dMI 5 °CAD before the main injection (main injection at 2.5 °CAD after top dead centre (ATDC) for 1500 rpm and 0.7 °CAD ATDC for 2250 rpm) and a long dMI 40 °CAD before the main injection which was investigated. Engine speeds 1500 rpm and 2250 rpm with loads of 35.1 Nm were selected to study the effect at low load. 35.1Nm represents 10% of the engines maximum load at 1500 rpm and 8.2% at 2250 rpm.

4. Results and Discussion

The engine was operated at low and moderate speeds with a constant load and the results plot was obtained by changing the IP for short and long dMI, both with and without cooled EGR. The cylinder pressure is measured inside the combustion chamber of cylinder 2 on the left side looking from the front of the engine and the related gross rates of heat release (GROHR) are displayed in the figures below (Fig. 1-4) as the engine operates with various IP and dMI. Increasing the IP decreases the fuel particle diameters giving faster and more effective vapourisation. Therefore, higher IP initially generates a faster combustion rate, resulting in higher in-cylinder peak temperatures and an advance in combustion. This is most noticeable in Fig. 2 on the ROHR plot, where the peak ROHR is shifted by as much as 7 CAD between the lowest and the highest IP. (The ROHR advance referred to here is the CAD position of the peak of the ROHR). It appears that the relationship between IP and ROHR advance is that the rate is logarithmically proportional i.e. as IP increases ROHR advances but at an ever decreasing rate (Fig. 5). On the other hand, at lower IP the initial combustion is slow due to a less efficient mixing process resulting in slower combustion in chamber. This leads to a shift in the heat release towards later CAD, a reduced peak ROHR and lower in-cylinder pressure.

Previous studies have found that the in-cylinder peak pressure is strongly affected by the mixture formation which is related to the oxygen availability in the combustion chamber [22-24] . This study shows that the long dMI produced a slightly higher in-cylinder peak pressure than the short dMI for both EGR ON and EGR OFF. The long dMI provides a longer residence time for the pilot fuel to mix resulting in more fuel ready to burn in the premixed combustion phase resulting in higher in-cylinder pressure due to the combustion of the pilot injection. Furthermore, the long pilot advance leads to the longer main ignition delay after the start of main injection, resulting in a more complete mixture, thus producing higher in-cylinder peak pressure. Fig. 4 shows that when EGR is applied, as well as the expected reduced peak pressures, there is also a shift to later CAD for the peak rate of heat release. This suggests a delay in combustion. It is thought that this is due to the increased dilution of EGR causing a delay in the rate of combustion. However, when the engine operates with higher IP we notice that this shift is certainly reduced, perhaps even eliminated. It may be viable to say that high IP can make the ROHR insensitive to EGR and that this is due to the increased penetration, atomisation (droplet size). I.e. quality of a high IP mixture.

Another important finding was that the effect of dMI on the in-cylinder peak pressure is less sensitive at low engine speed. It is firmly believed that this is due to a higher temperature combustion and increased turbulence at the higher engine speed operation. This results in enhanced fuel mixing, leading to higher peak cylinder pressure due to a more complete combustion [25].

The results show that the long dMI produces a higher premixed peak ROHR when compared to the short dMI for both a low and high IP. This is due to the longer residence time. Consequently more fuel is mixed with air and in turn, more of the fuel is burnt during combustion. In contrast, the short dMI provides a limited residence time for fuel to mix with air especially at lower IP and low engine speed. As a result, the shorter dMI produces a lower premixed peak pressure when compared with the longer dMI as shown in Fig. 2. It is noticeable that the heat released is shifted forwards to the early expansion stroke of the engine as the IP increases [26] due to an earlier combustion process at high speed in fig.2 . Clearly less energy is released from the short dMI (the area under the curve of the ROHR) suggesting that only a proportion of the fuel has been burnt and indicates poor mixing. This suggestion is reinforced on inspection of Fig. 8 which details the brake specific fuel consumption (BSFC) as fuel consumption increases for short dMI . The results also show that long dMI produces higher in-cylinder peak pressure (Fig. 1-4) as compared with the short dMI for all engine test conditions. This will also go some way to subsequently explain some of the emissions.

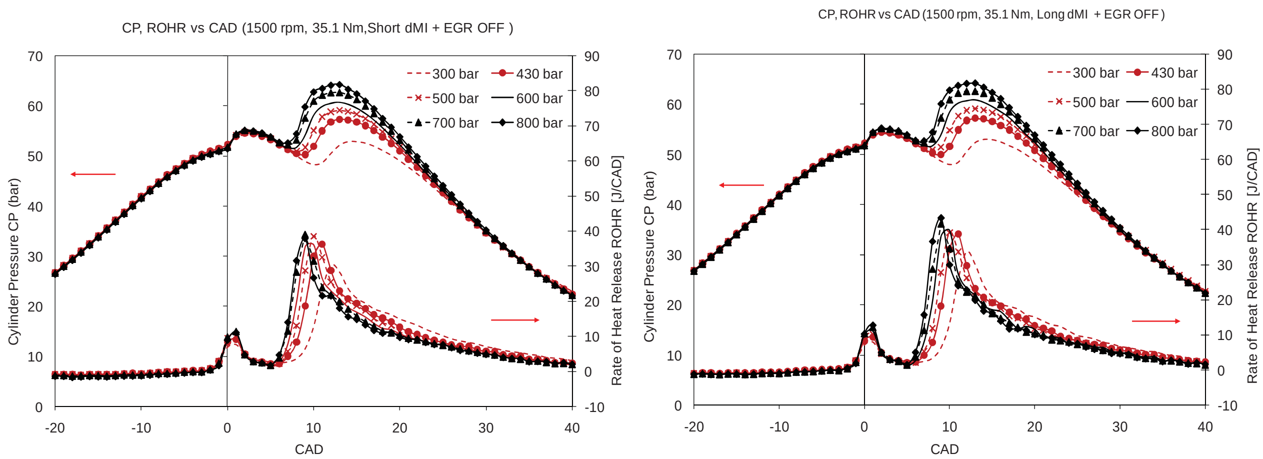


Fig. 1. Comparison of short and long dMI for 1500 rpm, 35.1 Nm and EGR OFF

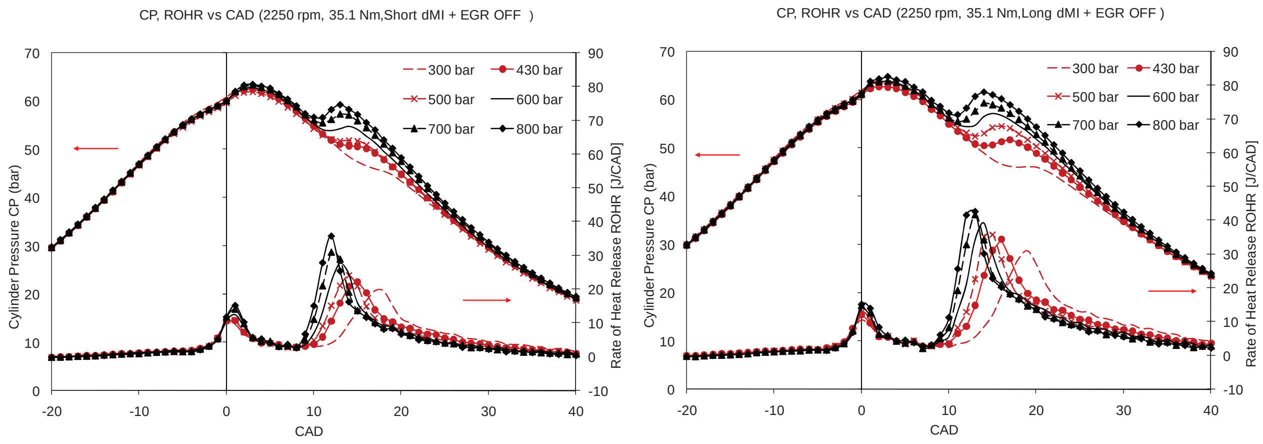


Fig. 2. Comparison of short and long dMI for 2250 rpm, 35.1 Nm and EGR OFF

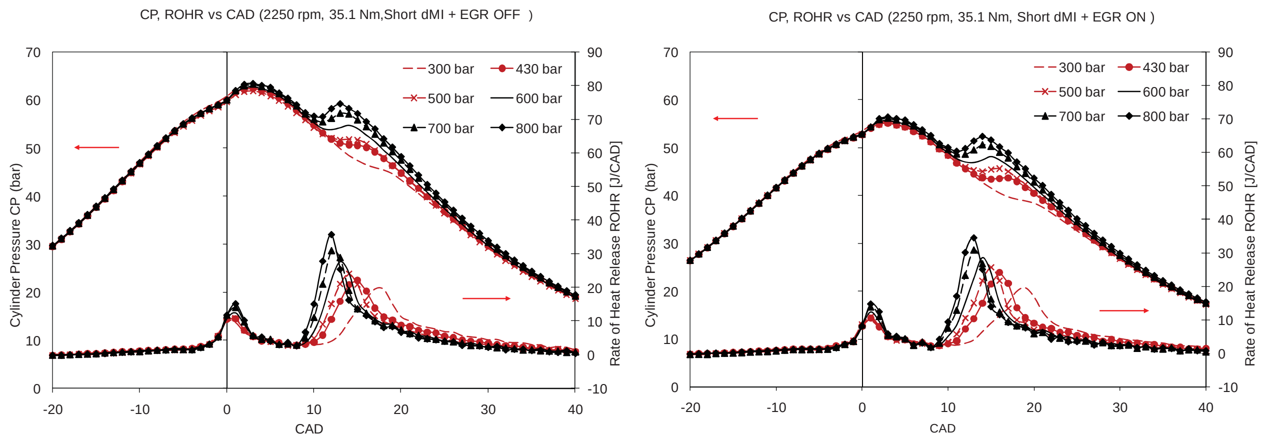


Fig. 3. Comparison of short dMI between EGR OFF with 30 % EGR for 2250 rpm, 35.1 Nm

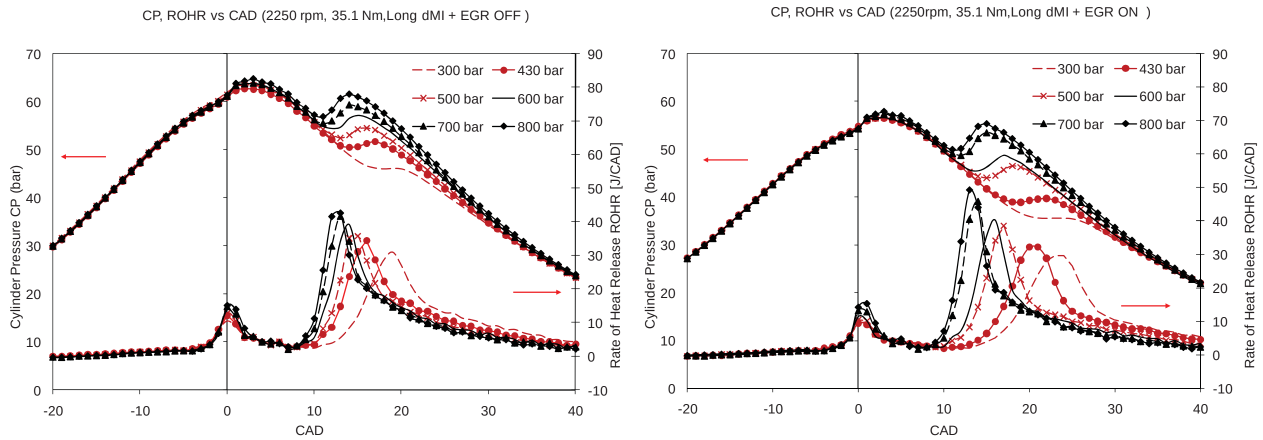
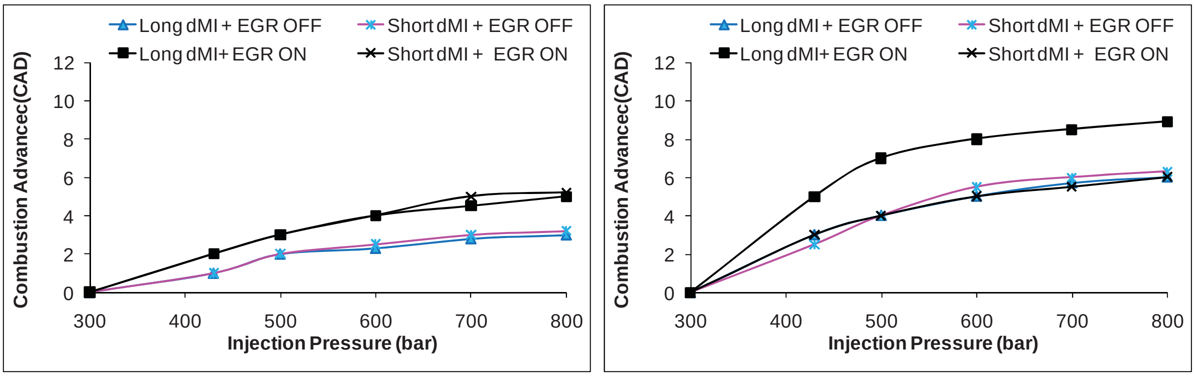


Fig. 4. Comparison of long dMI between EGR OFF with 30 % EGR for 2250 rpm, 35.1 Nm

5. Pilot Ignition Delay (the timing delay between pilot injection and first positive ROHR)

It is clearly observed from Fig. 7 and Fig. 8 that when the IP increases at constant dMI, the pilot ignition delay varied minimally for both EGR ON and EGR OFF. The effect of dMI on the pilot ignition delay is more dominant than that of the EGR. Long dMI with EGR OFF was found to have a slight trend towards a shorter ignition delay. It seems possible that these results are due to a longer residence time for the fuel to mix with air to form a combustible mixture even at lower injection pressure. This may indicate that the ignition delay is less sensitive to the IP when the engine operates with long dMI and higher oxygen availability i.e with EGR OFF.



a) b)

Fig. 5. The effect of IP on ROHR advance (compared to the lowest IP of 300 bar) a) 1500 rpm, b) 2250 rpm

It is interesting to note that the long dMI and EGR ON results show the ignition delay gradually decreases as the IP increases. The higher IP leads to the short ignition delay due to a smaller fuel particle size, consequently a better fuel/air mixture [26]. In the case of EGR ON, the IP plays a major role in producing a better mixture due to the decreased availability of oxygen in the cylinder. Hence, it could conceivably be hypothesised that the long dMI enhanced the combustion as the oxygen deficiency caused by EGR operation increased the dominance of the mixing process for complete combustion.

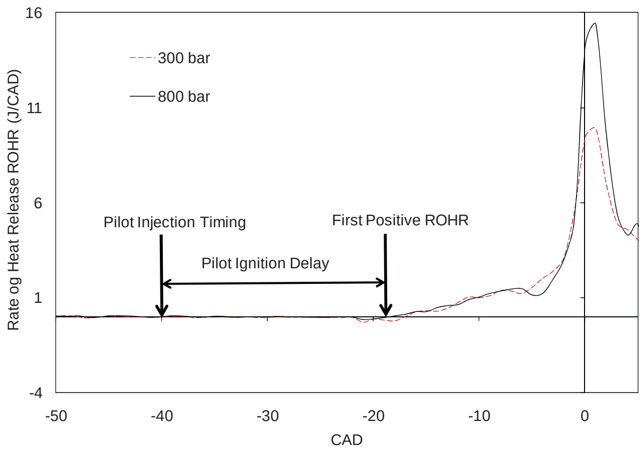
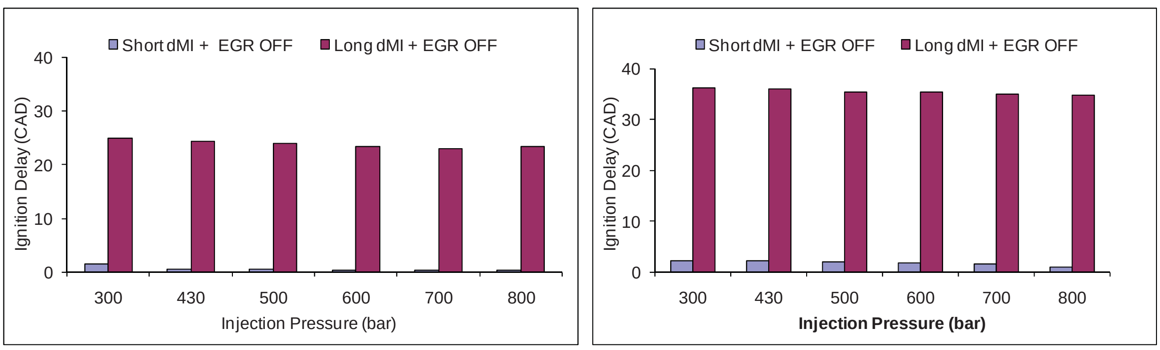
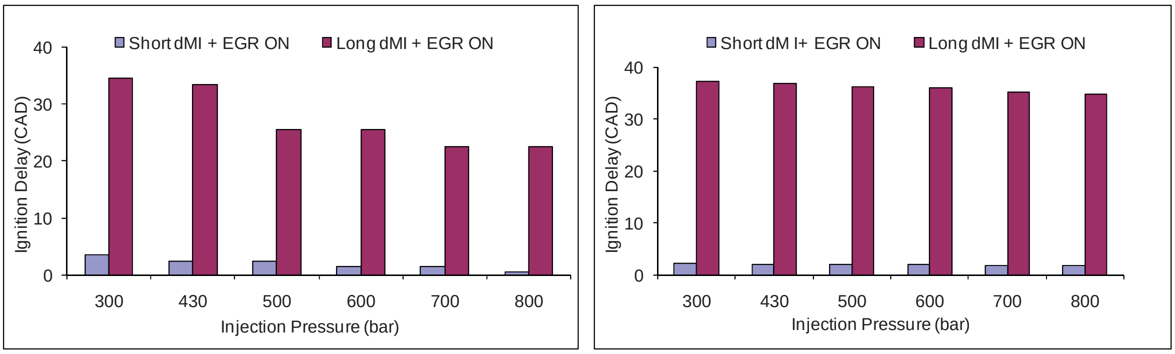


Fig. 6. Calculation of pilot ignition delay



a) 1500 rpm b) 2250 rpm

Fig. 7. The variation in ignition delay for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operating with short and long dMI and EGR OFF

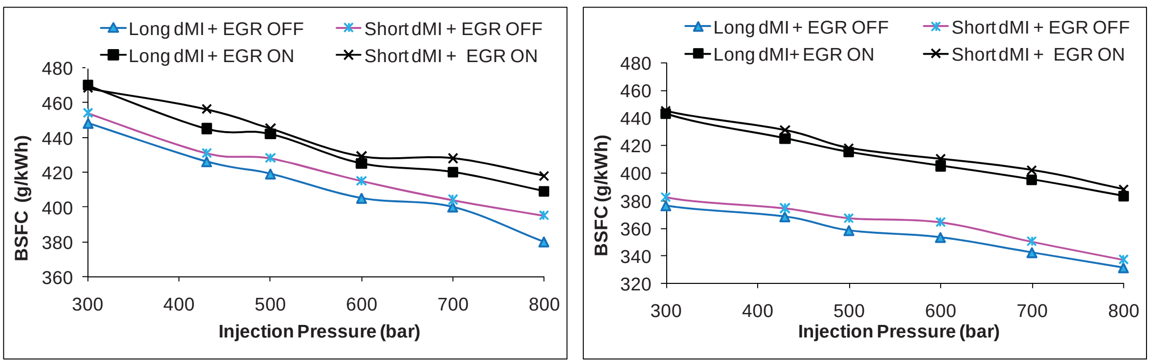


a) 1500 rpm *b) 2250 rpm*

Fig. 8. The variation in ignition delay for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operating with short and long dMI and EGR ON

6. Brake Specific Fuel Consumption (BSFC)

The results show that the effect of the EGR at lower engine speed is less sensitive than it is at higher engine speed. At lower engine speed, the effect of dMI is more dominant than the other parameters of the BSFC patterns. As expected, as the IP increased less fuel was consumed for both short and long dMI (Fig. 9). The overall results show that the long dMI consumed slightly lower BSFC compared to the short dMI when the engine is operated with EGR ON and OFF. As previously mentioned, this is thought to be due to the poorer mixing process of the short dMI which relates to the residence time of the fuel to mix with air in the combustion chamber. The shorter dMI provides a limited time for fuel to entrain with the air. In general, the BSFC decreases as IP increases due to the enhanced mixing process. As a result, energy release takes place closer to TDC, which increases the thermodynamic efficiency and thus reduces BSFC [20]. As the IP increases, the time for fuel to form a combustible mixture becomes shorter due to faster air entrainment, this results in a more active initial combustion [20].



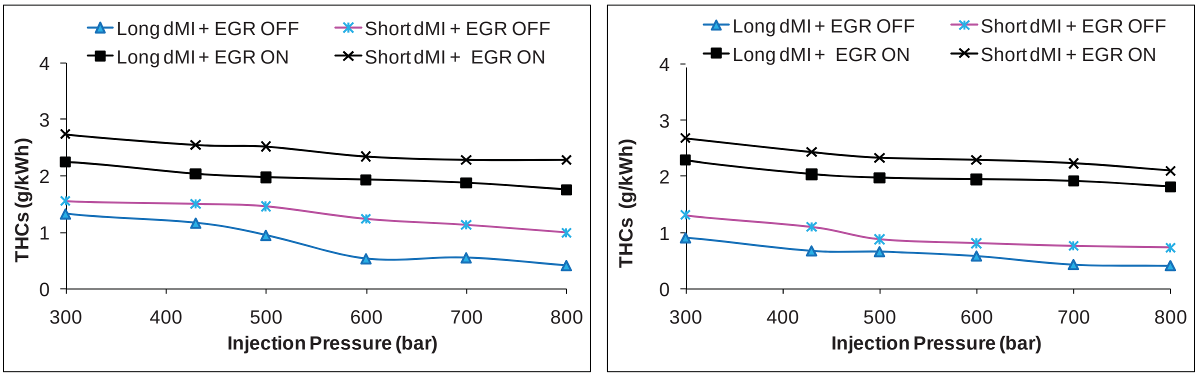
a) 1500 rpm *b) 2250 rpm*

Fig. 9. The variation of brake specific fuel consumption for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operating with EGR ON and EGR OFF

7. Total Unburnt Hydrocarbons (THCs)

Fig. 10 shows the THCs emissions trend for a diesel engine operating with different IP and dMI. Ideally, the fuel should be atomised and vapourised with air before it reaches a self-ignition temperature and initiates the combustion process. The mixing process is highly dependent on the fuel droplet size. As would be expected, the higher IP produces smaller fuel droplets and therefore, improved mixing. This results in a decrease in unburned hydrocarbons as the injection pressure increases due to improvements in the combustion [1].

The results show that the level of THCs is lower for long dMI when compared with short dMI for all variations of injection pressure when the engine is operating with EGR ON and OFF. It seems possible that these results are due to increased oxygen availability forming an improved mixture resulting in a more complete combustion [9]. On the other hand, the long dMI conceivably leads to higher wall adherence and a leaner mixture resulting in a higher THCs emissions level [25]. This concept seems sensible though this has not been observed here, perhaps this would become apparent for a dMI longer than was tested during this research. Fig. 10 is consistent with D. T. Hountalas et. al [27] who found the long dMI promoted higher combustion temperatures thus lowering the THCs emission. The reason for this is not entirely clear but it may be due mainly to improved mixing, but it is thought that perhaps higher turbulence and high combustion temperature also contribute to the more complete combustion at longer dMI [9].

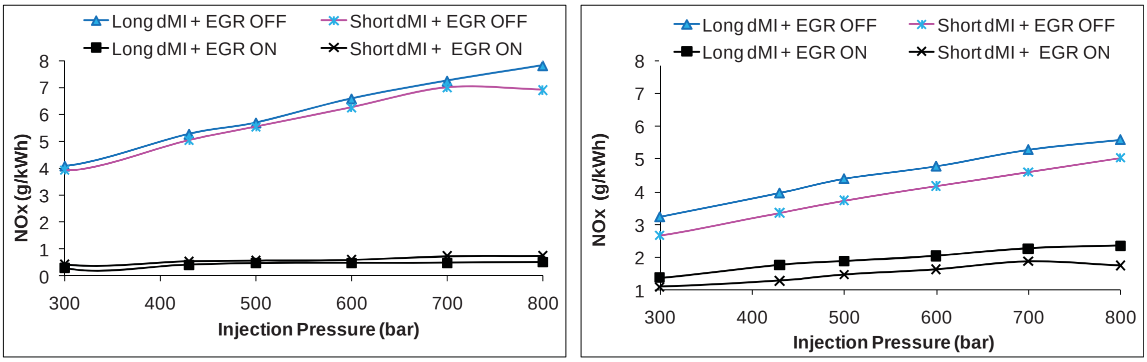


a) 1500 rpm *b) 2250 rpm*

Fig. 10. The variation of total unburnt hydrocarbons (THCs) emissions for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operating with EGR ON and EGR OFF

8. Nitrogen Oxides (NOx)

The results of this study show that the effect of dMI on NOx emissions is less significant at 1500 rpm and 35.1 Nm. The overall results show that the effect of dMI on the NOx formation is more significant at moderate speed (2250 rpm, 35.1 Nm) than low speed (1500 rpm, 35.1 Nm) due to the higher combustion temperature and increased turbulence. As IP increases NOx increases, this follows the trend detailed earlier of increased efficiency and would be expected. EGR is beneficial for NOx emissions across the range of IP and for this reason EGR is regularly used for NOx control. Reduction of NOx by EGR is due to a lower air/fuel ratio i.e. less oxygen and dilution increasing the volume of inert gas within the cylinder (the re-circulated gas). This inert gas raises the specific heat capacity of the overall mixture, so more energy is required to produce the same change in temperature, as when NO EGR is applied. The result of this is a reduced adiabatic flame temperature [11].When EGR was applied both the NOx emission and the rate at which NOx increases across the range of IP was reduced. The higher the IP the further the fuel penetrates into the cylinder, dispersing through an increased volume and the smaller the fuel droplets, leading to a leaner mixture (less localised fuel rich regions) and faster active combustion resulting in a higher level of NOx emissions [26, 28]. As would be expected, for the engine test condition of 2250 rpm and 35.1 Nm, the longer dMI produces a higher level of NOx when compared with a shorter dMI for both EGR ON and EGR OFF. As it is well known, NOx formation is related to the temperature, oxygen availability and residence time. The longer dMI produces a more complete combustion leading to higher local temperatures. These higher local temperatures and availability of oxygen promote NOx formation. The longer dMI tends to increase the flame temperature due to the preheated reactants, this will tend to result in higher NOx emissions, according to the Zeldovich mechanism as mentioned in [29]. In addition, the NOx emissions are influenced by the premixed combustion period which is related to the gas burned temperature [30].



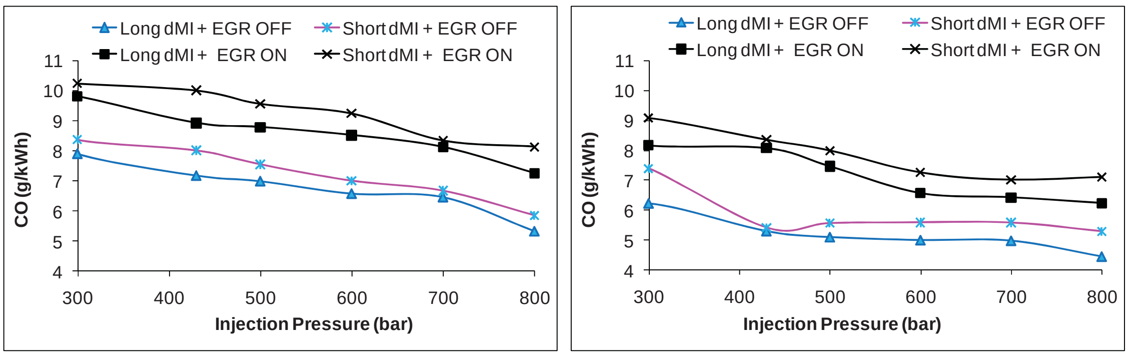
a) 1500 rpm *b) 2250 rpm*

Fig. 11. The variation of NOx emissions for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operating with EGR ON and EGR OFF

The results also revealed that NOx decreased as the engine speed increases. A study by J Benajes et. al [20] showed that this effect is a result of a local lean mixture induced by an air swirl motion, resulting in a reduction of the local temperature, this in turn leads to a reduction of NOx formation.

9. Carbon Monoxide (CO)

Fig. 12 shows the carbon monoxide emissions of the engine operating at different IP and dMI. As expected, CO levels increased when EGR was applied, due to the reduced combustion temperatures through dilution and decreased oxygen availability [31-33]. In fact, CO emissions are greatly influenced by fuel-air mixing and the air-fuel ratio [1, 24]. It is observed that the CO emissions were lower for longer dMI across the range of IP for both EGR ON and OFF. This result may be more apparent at high speed due to the increased effect of swirl and turbulence at the higher engine speed, influencing the emissions even at short dMI. Hence, it could perhaps be concluded that the higher engine speed induced swirl motion leads to the improved mixture formation for combustion, even at short dMI. The general trend for CO emission when IP is increased is downward, again this seems to follow the efficiency of combustion increase. This is detailed earlier and is in agreement with similar studies [34, 35].

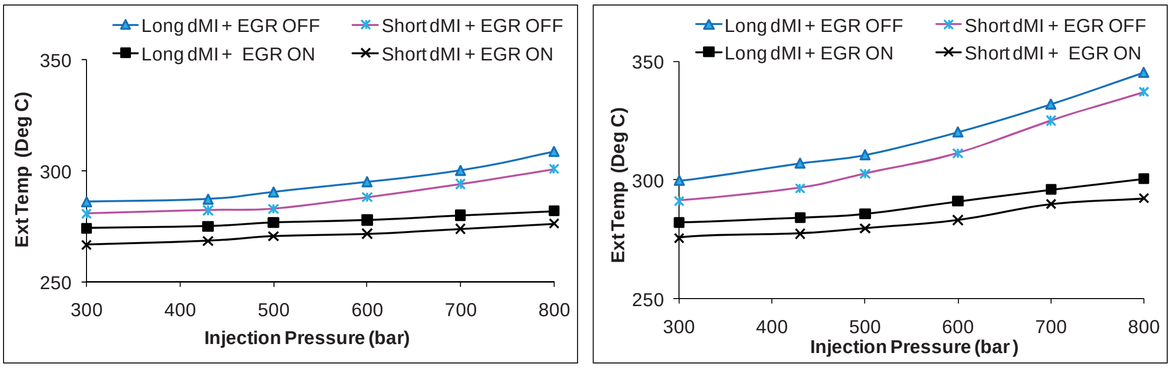


a) 1500 rpm *b) 2250 rpm*

Fig. 12. The variation of CO emissions for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operating with EGR ON and EGR OFF

10. Exhaust Temperature

The results show that the longer dMI produces higher exhaust temperatures when compared with the shorter dMI for all engine conditions. The more fuel that is burnt after TDC the higher the exhaust gas temperature. Longer dMI provides more time for the fuel to evaporate and mix with surrounding air leading to a higher exhaust temperature. As IP increases, the fuel droplet size decreases leading to fast (active) combustion [10, 23, 33, 36] thus producing a higher exhaust gas temperature. The low exhaust temperature at lower IP is due to a poor combustion process leading to a lower combustion temperature. In general, the exhaust gas temperature is higher when the engine is operating without EGR. With EGR ON the combustion is slower due to dilution producing a lower combustion temperature and in turn, exhaust gas temperature. This difference in temperature is not so noticeable for lower injection pressures where the application of EGR leads to increased delay in the ROHR (Fig. 4). This shifts the combustion closer to the exhaust stroke, thus reducing the time between combustion and exhaust. We can perhaps then say with some certainty that at, higher IP, exhaust temperatures are mainly dependant upon the peak combustion temperature. Also we know that the higher IP do not give a delay in the ROHR for EGR ON. As this has the opposite effect compared to the reduction in peak cylinder temperatures due to EGR the difference between exhaust temps for EGR ON and EGR OFF for low IP is far less than at high IP, for which EGR has no effect on the position of the peak of the main ROHR (Fig. 12).



a) 1500 rpm *b) 2250 rpm*

Fig. 12. The variations of exhaust gas temperature for six multiple injections for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operates with EGR ON and EGR

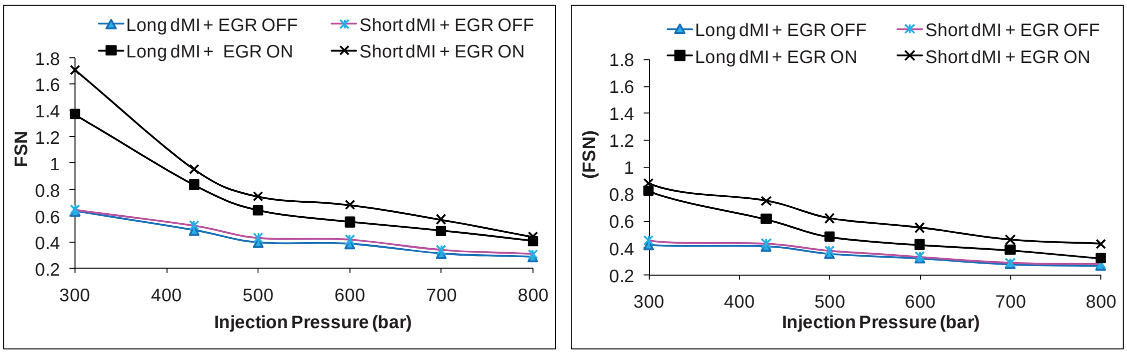
Interestingly, as IP increases for all operating conditions the exhaust temperatures increased due mainly to the improvements in combustion as already detailed. A potentially positive consequence of this is that increased exhaust temperature can aid the efficiency of aftertreatment devices that are to be found in modern exhaust systems. Examples are: the diesel oxidation catalyst (DOC), a selective catalytic reduction catalyst (SCR), and the diesel particulate filter (DPF).

11. Smoke

The variation of smoke relative to the variation of the IP and dMI are shown in the Fig. below. The effect of dMI on the smoke emissions is not significant when the engine operates with EGR OFF since both dMI produce similar behaviour. However for EGR ON, the shorter dMI produces higher levels of smoke, compared to the longer dMI. This is as expected and is explained by the increased quality of mixing achieved with a longer dMI. In this case, the smoke level is more sensitive to the IP and EGR operation, rather than the difference between the long and short dMI. The increase in the IP from 300 bar to 800 bar resulted in a reduction of smoke. These results are consistent with other studies by Siddappa et al. and Dhananjaya D A et al. [1, 10] and suggest that the higher IP can be beneficial due to better fuel atomisation leading to improved combustion behaviour. In contrast, the shorter dMI emits higher levels of smoke due to less time being available for the fuel to mix with air, this leads to less complete combustion.

It is clearly observed from the graph that EGR operation produces higher levels of smoke emissions, due to a large displacement of oxygen (compared to EGR OFF) in the charge. This leads to a reduction of the rate of soot oxidation. However, the smoke emissions decrease as the IP increases due to an improved mixing process and less fuel-rich regions that can contribute to soot formation [26, 37]. A late combustion process can benefit soot oxidation, resulting in low emission levels due to higher temperatures extending the combustion into the expansion stroke. These findings suggest that the higher IP can be considered for smoke reduction when EGR is applied.

Previous studies [3, 38] have mentioned that smoke formation occurs in rich zones inside the flame reaction, while the oxidation is mainly produced at the reaction surfaces due to an insufficient amount of oxygen being available in these localised regions. In addition, the soot formation is extremely sensitive to the air entrainment [39] during the injection process. The longer dMI leads to reduced smoke emissions, this is due to the increased period for mixing to occur, this enhances the mixing process and concurs with many studies [3, 20, 24, 40].



a) 1500 rpm *b) 2250 rpm*

Fig. 13. The variation of smoke emission for split injections for a range of injection pressures (300, 430, 500, 600,700 and 800 bar) operating with EGR ON and EGR OFF

Long dMI produces lower amounts of smoke than short dMI, this might allow the use of an increased EGR rate (EGR rate can be limited by soot emission levels). This would lower NOx levels further. Perhaps to levels lower than for short dMI. This might have potential for an optimised injection strategy incorporating higher EGR rates (e.g. >60% EGR). The potential for lower levels of NOx than the less efficient short dMI is a subject envisioned for further study.

12. Conclusions

Experimental analysis has been carried out to evaluate the effect of dMI and IP on the engine performance and emission of a multi cylinder with split injection with and without EGR.

Combustion

- when the IP is increased the peak combustion pressure and peak ROHR increases,
- as the IP increases the combustion advance follows a logarithmical trend,
- long dMI leads to an increased combustion pressure and peak ROHR,
- as IP increases, fuel consumption goes down particularly for the longer dMI.

Emissions

- the most interesting finding was that the higher injection pressures of 600 to 800 bar are favourable for most emissions (Smoke, CO, HCs), but introduced a small penalty in terms of NOx,
- it was also shown that the longer dMI produced better engine performance and emissions than shorter dMI,
- both long and short dMI are suitable for EGR,
- increased IP increases the exhaust gas temperature, providing potential for improvements in the efficiency of aftertreatment technologies,
- smoke levels decrease as IP increases,
- this leads to the possible potential to reduce emissions in terms of nitrogen oxides, THCs and soot by employing a combination of dMI, IP and EGR.

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