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REVIEW ARTICLE

ANALYSIS OF CYLINDER PRESSURE AND HEAT RELEASE RATE WITH EXHAUST EMISSION AND FUEL CONSUMPTION IN A DIESEL ENGINE

Analysis of Cylinder Pressure and Heat Release Rate with Exhaust Emission and Fuel Consumption in a Diesel Engine

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1. INTRODUCTION

Individual cylinder pressure based feedback control is an ideal method to optimize engine operation. In-cylinder pressure is a fundamental combustion variable, which can be used to characterize the combustion process for each combustion event. Optimal engine control can be maintained by monitoring the pressure in each cylinder and using this information for feedback control in order to minimize the exhaust gas emissions. The availability of low cost and robust pressure sensors, such as the glow plug installed pressure sensor (GPPS) is expected to lead to the wide adoption of cylinder-pressure-based engine control diesel engines. In order to identify the most effective parameters to use when controlling combustion based on cylinder pressure information, a series of experiments were carried out with varying injection timings and EGR levels. This paper presents the results obtained and discusses the potential key parameters that may be used for closed loop control using an in-cylinder pressure transducer. This paper will also look at the combustion properties for combined hydrogen/diesel combustion. It was necessary to find an appropriate running point for the engine in purely diesel mode to act as a base point. Once this point was found, an experimental test matrix for the combined EGR and hydrogen investigation was set up.

2. INJECTION TIMING AND EGR EXPERIMENTAL TEST MATRIX

The injection strategy for this part of the investigation was a single injection at 1200 bar, maintained via the high-pressure common rail, at injection timings varying from 14 CAD BTDC and 0 CAD BTDC. The EGR level was varied, in 5% steps, from 0% and 40%.

A set operating point of 1500 rpm and 2.7 bar BMEP was used. The exhaust gas emissions were all measured. This information was then used to plot

correlations between the in-cylinder pressure and the related combustion properties and the resultant exhaust gas emissions.

3. RESULTS AND DISCUSSION FOR THE INJECTION TIMING INVESTIGATION

The results are obtained for a range of EGR and fuel injection timings.

3.1 Combustion Parameters

The followings are a summary of results obtained. Figure 1 shows that with increasing levels of EGR the 50 % MFB point is delayed. It can also be seen that higher levels of EGR have more of an effect than lower levels as the lines diverge with increasing EGR. This effect becomes more pronounced as the injection timing is retarded.

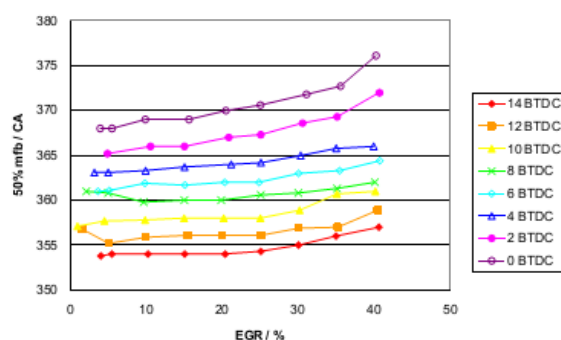


Figure1 Correlation between 50 % mass fraction burnt, EGR and injection timing

From Figure 2 it can be seen that as the EGR level is increased, the maximum in-cylinder pressure decreases. This is a known effect of EGR addition. The maximum pressure decreases as the injection timing is retarded. As the injection timing

is retarded the effect of EGR addition becomes more pronounced.

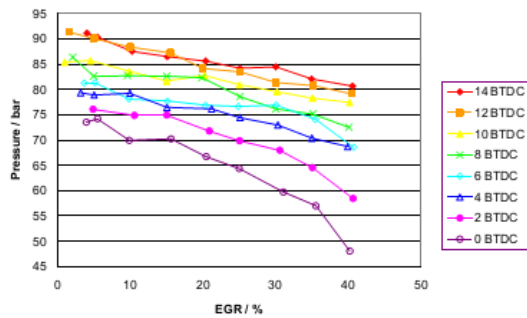


Figure 2 Correlation between maximum pressure, EGR and injection timing

From Figure 3 it can be seen that increasing the EGR level delays the position of the maximum in-cylinder pressure. The position of the maximum pressure is also delayed as the injection timing is retarded. As the injection timing is retarded, the effect of the EGR is increased.

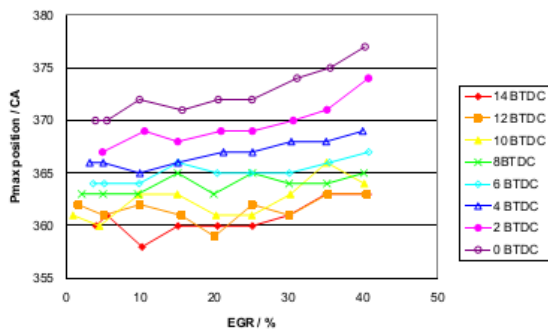


Figure 3 Correlation between the position of the maximum pressure, EGR and injection timing

From Figure 4 it can be seen that with increasing levels of EGR, the maximum pressure rise rate decreases. As the injection timing is retarded, the pressure rise rate decreases. The effects of EGR appear to be greater with retarded injection. The results of different injection timings are not well separated. Hence, this is not a particularly reliable parameter.

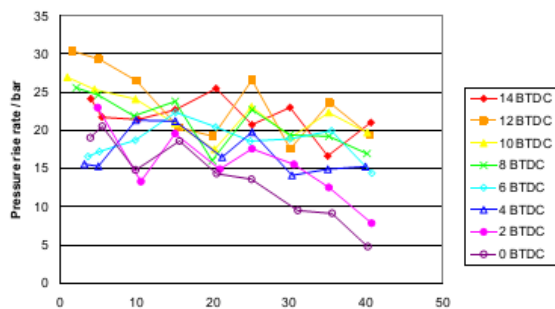


Figure 4 Correlation between the maximum pressure rise rate, EGR and injection timing

Figure 5 shows that the position of the maximum pressure rise rate is delayed with increased levels of EGR. It can also be seen that as the levels of EGR increases, the delay is also increased. As the injection timing is retarded, the position of the maximum pressure rise rate is delayed. This is all due to the increased ignition delay.

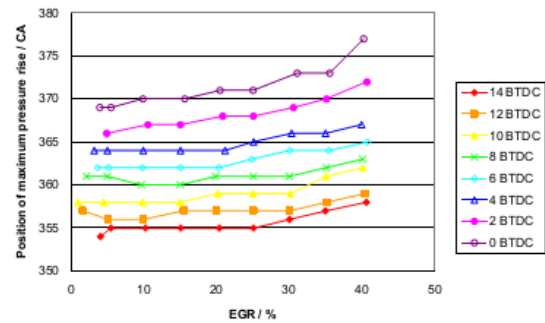


Figure 5 Correlation between the position of the maximum pressure rise rate, EGR and injection timing

Figure 6 shows that the motoring pressure at 20 crank angle degrees before TDC decreases as the level of EGR increases, as was the case for the maximum in-cylinder pressure. The motoring pressure decreases as the injection timing is retarded. This is a parameter which needs further investigation before it could be recommended for a closed loop control system.

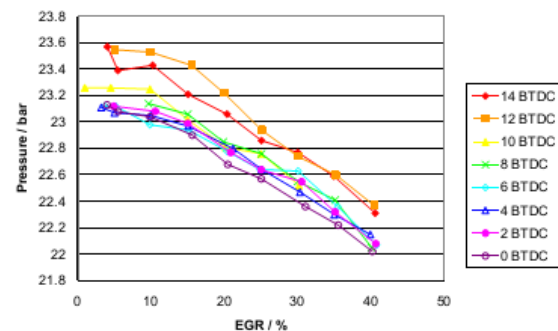


Figure 6 Correlation between the pressure at 20 degrees before TDC, EGR and injection timing

3.2 Emissions

From Figure 7 it can be seen that increasing EGR decreases the emission of nitrogen oxides (NOX). Retarding the injection timing also decreases the emission of NOX. At earlier injection timings, the effect of EGR is more pronounced, as can be seen from the converging lines in Figure 7.

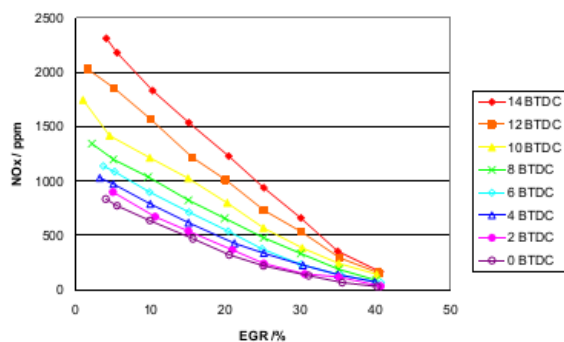


Figure 7 Correlation between Nitrogen Oxides emissions, EGR and injection timing

Figure 8 clearly shows that retarding the injection timing increases the total unburnt hydrocarbons (THC). As is known, increasing EGR level also increases the total unburnt hydrocarbons. The combined effect of retarded injection and high levels of EGR means a significant increase in the total unburnt hydrocarbons. This makes it necessary to optimise the injection strategy to minimise the THC emissions.

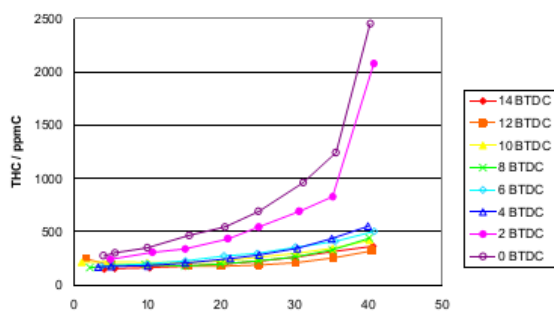


Figure 8 Correlation between total unburnt hydrocarbons, EGR and injection timing

Although not quite so clear, Figure 9 shows that increasing the EGR by a small percentage does not have much of an effect on the smoke number. With more than approximately 20% EGR, increasing the EGR significantly increases the smoke number. Broadly, retarded injection timing increases smoke number after the 20% EGR level.

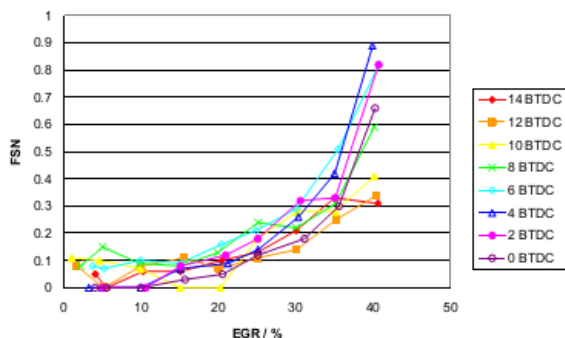


Figure 9 Correlation between the smoke number, EGR and injection timing

It can be seen from Figure 10 that increasing EGR levels increases the emissions of carbon monoxide (CO). It can also be seen that retarding the injection timing increases the emission of carbon monoxide. The effect of EGR increases both with retarded injection timing and increasing EGR levels.

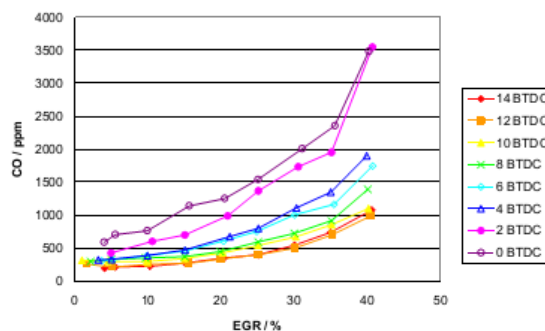


Figure 10 Correlation between carbon monoxide, EGR and injection timing

Figure 11 shows that the specific fuel consumption decreases with EGR. The specific fuel consumption also decreases with retarded injection timing. Figure 32 also shows that this parameter is variable, and not appropriate for use as a closed loop control.

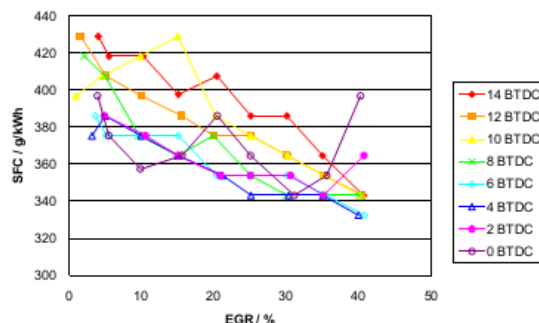


Figure 11 Correlation between the brake specific fuel consumption, EGR and injection timing

3.3 Correlation between Parameters

In Figure 12 EGR increases towards the bottom of the graph. Injection timing is retarded from left to right. Figure 33 shows that the position of the 50% MFB is retarded as nitrogen oxides are decreased.

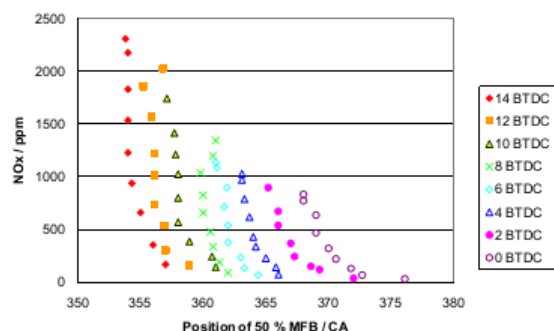


Figure 12 Correlation between the position of 50% MFB, nitrogen oxides, EGR and injection timing

Unlike in the previous figure, in Figure 13 EGR increases towards the top of the graph. The injection timing is still retarded from left to right. From Figure 13 it can be seen that as the 50% MFB position is delayed the total unburnt hydrocarbons increases. This is the usual diesel trade off: as NOX is decreased, THC increases.

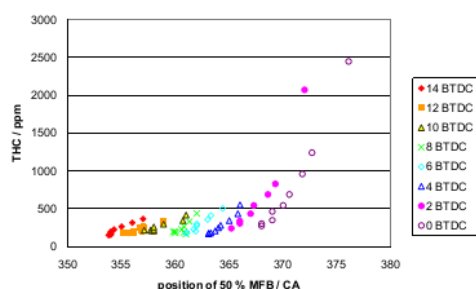


Figure 13 Correlation between total unburnt hydrocarbons, the position of 50% MFB, EGR and injection timing

Similar to Figure 13, Figure 14 follows EGR increases towards the top of the graph. Injection timing is retarded from left to right. As was the case for the total unburnt hydrocarbons, as the 50% position is delayed, the filter smoke number is increased, as shown in Figure 14. This directly follows the THC emissions, which contributes to the smoke number, and is a direct result of the higher incomplete combustion.

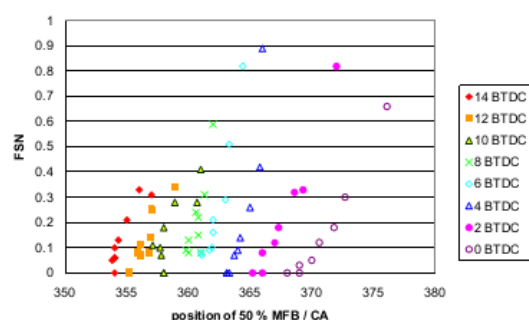


Figure 14 Correlation between the position of 50% MFB, smoke number, EGR and injection timing

As with the previous two figures, with Figure 15 EGR increases towards the top of the graph. Injection timing is retarded from left to right. It follows that as the 50% MFB position is delayed, and therefore the ignition delay, there is less time for the combustion event to occur. This means that incomplete combustion was increased, resulting in higher carbon monoxide emissions, as can be seen in Figure 15

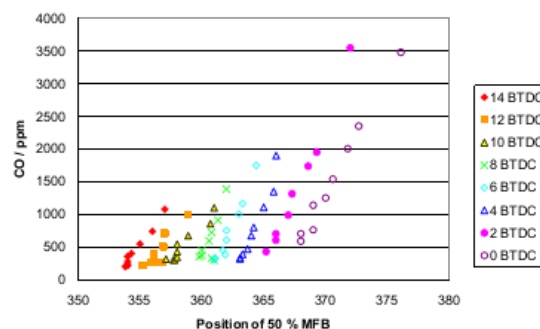


Figure 15 Correlation between the position of 50% MFB, carbon monoxide, EGR and injection timing

For Figure 16 EGR increases towards the bottom of the graph. Injection timing is retarded from left to right. As the position of the 50% MFB is delayed, the specific fuel consumption increases, as is shown in Figure 16. The increase in incomplete combustion means lower fuel efficiency, and therefore higher fuel consumption.

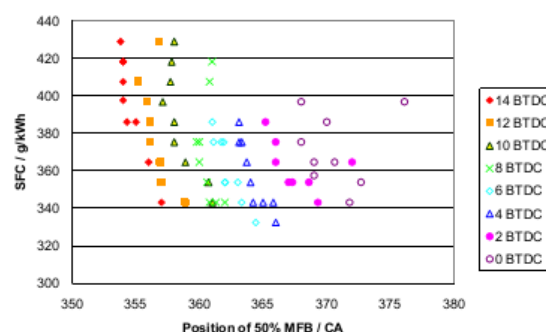


Figure 16 Correlation between the position of 50% MFB, the specific fuel consumption, EGR and injection timing

For Figure 17 EGR increases from the top to the bottom of the graph. Injection timing is retarded from left to right. As the position of the maximum rate of pressure rise rate is retarded, the NOX emissions decrease, as shown in Figure 17. This decrease in NOX emissions corresponds to the in-cylinder pressure decrease brought about by the increasing EGR levels, as well as the decreased

time for the combustion event, and therefore NOX formation time.

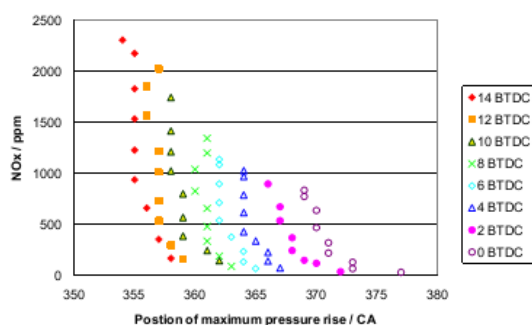


Figure 17 Correlations between the Position of Maximum Pressure Rise Rate, Nitrogen Oxides, EGR and Injection Timing

4. INJECTION TIMING AND PRESSURE INVESTIGATION

In order to find an appropriate base point for the hydrogen experiments, an investigation of injection pressures and timings for a single diesel injection strategy was conducted. Injection pressures were varied from 400 to 1400 bar. The injection timing was varied between 0 and 14 Crank Angle Degrees Before Top Dead Centre (CAD BTDC).

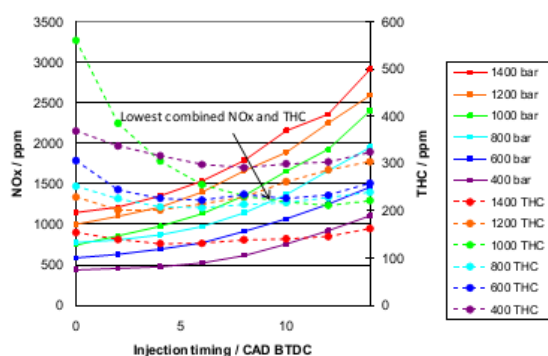


Figure 18 Lowest combined NOX and THC over various injection pressures and timings

As can be seen from Figure 18, NOX emissions increase as the injection timing is advanced, and as the injection pressure is increased. This is to be expected, as NOX emissions increase with temperature and with lower flame speeds. Earlier injection timings allows for a higher portion of the diesel fuel to be burnt during the premixed phase of combustion. This leads to higher rates of in-cylinder pressure rise, higher maximum in-cylinder pressures, and therefore higher temperatures. Conversely, the total unburnt hydrocarbons decrease as the injection timing is retarded and there is more time to allow for complete combustion. As the injection

pressure is increased, the fuel atomises better, allowing for better fuel air mixing and therefore more complete combustion. This tradeoff is typical of diesel combustion.

5. HYDROGEN AND EGR EXPERIMENTAL TEST MATRIX

In order to investigate the effects of combined hydrogen/diesel combustion with EGR addition, various levels of EGR and hydrogen addition were used. The EGR levels were varied between 0% and 40%, where the engine was capable of running this level, in 5% steps. Hydrogen was supplied as shown in the previous chapter, and the levels were varied between 0 and 10% of the volume of the inlet charge, in 2% vol. steps. In the case of running both hydrogen and EGR, the hydrogen always replaced air.

Two engine speeds were run, 1500rpm and 2500rpm, and 3 engine loads. The 3 engine loads were 0 bar BMEP, 2.7 bar BMEP and 5.4 bar BMEP, referred to as no load, medium load and high load throughout this paper. Since not all combinations of the speeds and loads were capable of supporting the maximum levels of hydrogen and EGR, Table 1 shows the achieved levels for each operating point.

Operating Point	Hydrogen Level	0% vol.	2% vol.	4% vol.	6% vol.	8% vol.	10% vol.
1500 rpm 0 bar	40 % EGR	30% EGR	30% EGR	Not run	Not run	Not run	Not run
1500 rpm 2.7 bar	40% EGR	35% EGR	40% EGR	10% EGR	Not run	Not run	Not run
1500 rpm 5.4 bar	30% EGR	30% EGR	30% EGR	15% EGR	15% EGR	10% EGR	10% EGR
2500 rpm 0 bar	40% EGR	40% EGR	35% EGR	Not run	Not run	Not run	Not run
2500 rpm 2.7 bar	40% EGR	40% EGR	40% EGR	35% EGR	Not run	Not run	Not run
2500 rpm 5.4 bar	15% EGR	15% EGR	15% EGR	15% EGR	Not run	Not run	Not run

Table 1 Experimental Test Matrix

6. RESULTS AND DISCUSSION FOR THE HYDROGEN INVESTIGATION

6.1 Pressure and Heat Release Rate

As described in the previous chapter, the in-cylinder pressure data was captured using a pressure transducer connected to a PC running LabVIEW via a charge amplifier. This in-cylinder pressure data, combined with the crank angle data captured simultaneously from the shaft encoder, was used to analyse the combustion process. The data

capture was averaged over 20 cycles. The heat release rate was calculated using Equation.

$$HRR = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta}$$

where $\gamma=1.3$

The heat release rate trace was used to determine the nature of the combustion. As can be seen by comparing Figure 19 and Figure 20, the addition of hydrogen has increased the maximum heat release rate. This is due to the increased ignition delay, and the great proportion diesel fuel being burned in the premixed phase of combustion. This increased rate of heat release is only true while stable combustion can be maintained, as will be discussed later in this section. Looking at Figure 20 it can be seen that the maximum rate of heat release actually decreases slightly, although the hydrogen level supplied has increased. This shows that the limit of stable combustion has been reached, and the benefits of the hydrogen addition will begin to decrease.

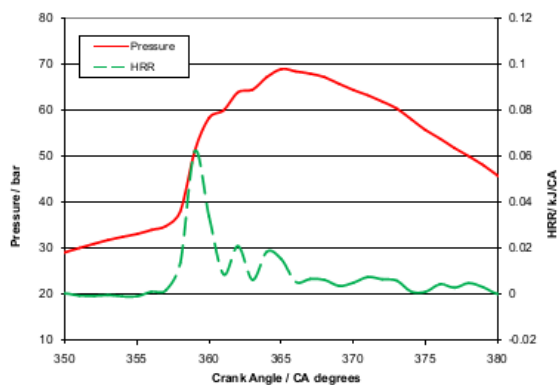


Figure 19 In-cylinder pressure and Heat Release Rate at 2500 rpm and 5.4 bar BMEP, 0% vol. hydrogen, 20% EGR

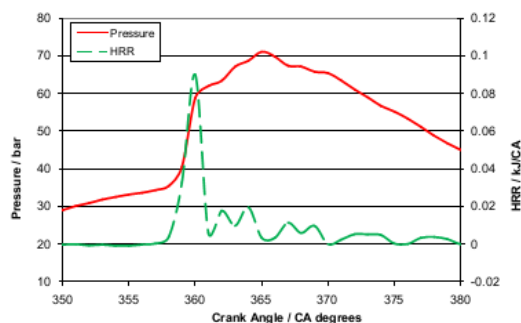


Figure 20 In-cylinder pressure and Heat Release Rate at 2500 rpm and 5.4 bar BMEP, 2% vol. hydrogen and 20% EGR

As can be seen from Figure 21 the maximum pressure for medium and high loads increases with hydrogen at 2500 rpm. For the case of no load, increasing

hydrogen decreases the maximum pressure. This is due to the unstable nature of hydrogen combustion at low loads. Unstable combustion was taken to be a greater than 5% Coefficient of Variation (CoV) of maximum in-cylinder pressure. (This has also been observed in previous studies where at low loads, the in-cylinder conditions were not always favourable for the efficient oxidation of hydrogen and the engine combustion stability could deteriorate [Tsolakis (2004)].) There is higher maximum cylinder pressure with no EGR, due to the effect of EGR of lowering the maximum pressure.

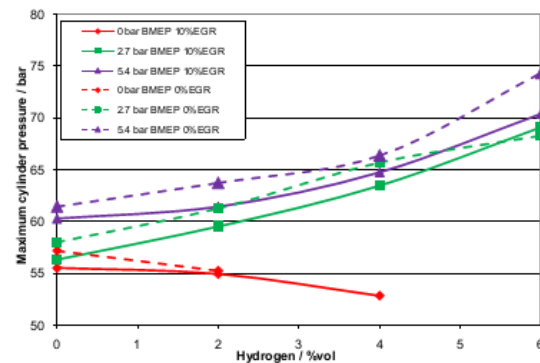


Figure 21 Maximum in-cylinder pressure at 2500 rpm

The 1500 rpm case is slightly different from the 2500 rpm case, as can be seen from Figure 48. The onset of unstable combustion at 1500 rpm occurs at a lower load than at 2500 rpm. The advantages of hydrogen induction are clearer at high loads at low speeds.

6.2 Ignition Delay

As can be seen from Figure 22 as the hydrogen addition level increases the ignition delay increases, except in the no load case. Ignition delay was taken as the time between the start of fuel injection signal and the start of combustion. The start of combustion in this instance was taken as the point on the heat release rate curve after injection where the value changed from negative to positive [Heywood (1988)]. The decreased ignition delay in the no load case is interesting as it is not what would normally be expected with hydrogen addition, but is also reflected in the unusual trends in the emissions. The no load case needs further investigation to establish the exact chemical kinetics at this point. It may also be useful to measure the hydrogen emissions to establish whether or not the hydrogen is burning as the combustion at this point is unstable.

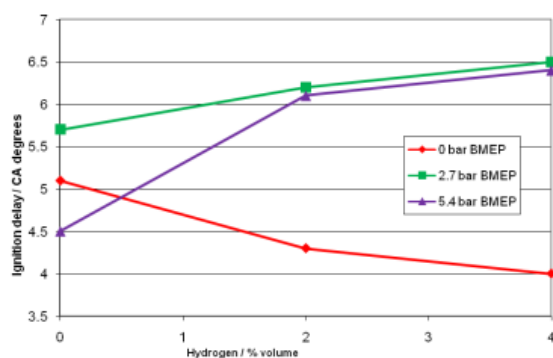


Figure 22 Ignition delay at 2500 rpm

Hydrogen addition also resulted in increase of the peak heat release rate until the onset of unstable combustion. This is a result of the longer ignition delay, and therefore a greater volume of diesel fuel being burnt during the premixed phase of combustion. This in turn makes the combustion noisier.

6.3 Maximum Rate of In-cylinder Pressure Rise

Similarly as shown in Figure 23, the maximum rate of cylinder pressure rise increases with hydrogen (until the combustion becomes unstable) for medium and high loads, but decreases in the case of no load. With EGR the maximum rate of pressure rise was decreased. The increased maximum rate of cylinder pressure rise means that the combustion using hydrogen is noisier than using exclusively diesel.

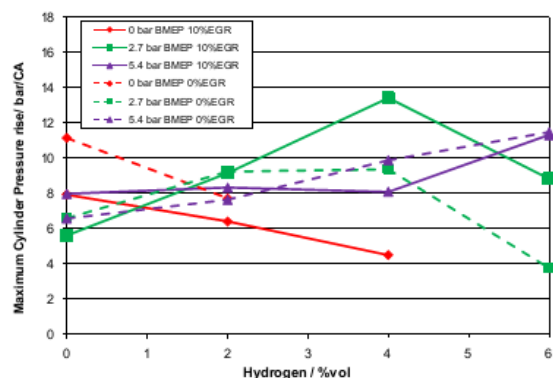


Figure 23 Maximum rate of in-cylinder pressure rise rate at 2500rpm

In the case of the 1500 rpm speed, there is an unexpected dip in the rate of maximum in-cylinder rise at 4% vol. hydrogen addition for medium and high loads.

7. CONCLUSION

This paper has looked at various combustion parameters and their correlations with the exhaust

gas emissions in order to make suggestions for a closed loop control system using an in-cylinder pressure transducer. The effects of hydrogen addition on the combustion parameters are also examined.

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