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Ignition Delay in an Ethanol Fumigated Common Rail Diesel Engine

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Abstract

A novel method for determining ignition delay is presented. This method utilises combustion resonance as a means of determining the onset of ignition. Results are shown from an ethanol fumigation study comprising of substitutions up to 50% at full, three-quarter and half load. It has been demonstrated that at full load there is a decrease in ignition delay with increasing ethanol substitutions, whereas at half load there is an increase in ignition delay with increasing ethanol substitutions. It is suggested that this conflicting result is a consequence of the auto-ignition of ethanol.

Keywords: Ignition Delay, Combustion Resonance, Ethanol Fumigation.

Nomenclature

D100E000	100% diesel fuel
D090E010	90% diesel fuel and 10% fumigated ethanol on an energy basis
D080E020	80% diesel fuel and 20% fumigated ethanol on an energy basis
D070E030	70% diesel fuel and 30% fumigated ethanol on an energy basis
D060E040	60% diesel fuel and 40% fumigated ethanol on an energy basis
D050E050	50% diesel fuel and 50% fumigated ethanol on an energy basis
D000E050	Engine operation after D050E050 has been established and the diesel fuel is switched off to cylinder one
D000E000	Motored, or misfired, operation

1. Introduction

A recent study by Shafiee and Topal shows it is probable after 2042 the only fossil fuel available will be coal [1]. Fifteen percent of Australia's greenhouse gas emissions are from the transport sector [2]. This value in the United States is considerably higher at 28% [3]. Indicators such as these place pressure for viable, cleaner bio-origin fuels as alternatives to fossil fuels for transport to be developed and implemented [4].

Currently, diesel fuel dominates the transport sector because of its superior thermal efficiency to other fuels; at full load a diesel engines uses only ~70% of the fuel that a comparable gasoline engine consumes for the same power output [5]. Fumigation, which is the introduction of finely atomised supplementary fuels into the intake air, has been a proposed method for diesel fuel substitution since the 1940s [6]; although it was first mentioned in Rudolf Diesel's original patent for internal combustion engines in 1898 [7]. In diesel engines fumigation can be achieved with many gaseous fuels.

Although lower alcohols such as methanol or ethanol are not gaseous, there is still a large amount of interest in using them as secondary fuels in diesel engines [8,9]. Ethanol, as a secondary fuel, has the potential to significantly reduce the dependence on fossil fuels in internal combustion engines in the immediate future. The biodiesel industry is only being developed, whereas ethanol can be readily produced economically and is easily stored and handled [10].

A great advantage of dual-fuel operation in a diesel engine is that it only requires a minimal change to the current engine operation. This is because the injector for the ethanol is placed at the intake manifold. However, this solution does require a separate fuel tank. It has been suggested by Chauhan et al. [10] that although a lot of research has been done in ethanol fumigation with diesel engines, the work has been limited and warrants further investigation. Ethanol fumigation has proved to be a viable solution to alleviate the diesel fuel demand. Ethanol can be used to displace up to 50% of the diesel fuel during engine operation; however, substitutions this high have significant drawbacks and a 15-20% substitution is more practical [9-12].

Ignition delay has marked effects on the operation of an engine. Specifically, it can drastically alter the performance of the engine and the exhaust emissions [12]. The presence of a secondary fuel in the intake air reduces the oxygen concentration of the intake air and the secondary fuel also undergoes pre-ignition reactions. These have a significant effect on the ignition delay [12].

It is well accepted that the presence of ethanol decreases the temperature at the end of the intake process [11,13]. This, coupled with the changes in air density and the low cetane value of ethanol, is attributed by most authors to cause an increase in ignition delay with ethanol fumigation [8,9,12,13].

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This paper investigates the effect of ethanol fumigation on ignition delay. The novelty of this work lies with the method used to determine ignition delay. Unlike other investigations into ignition delay, the work presented in this paper uses combustion resonance. Combustion resonance has been shown to be useful for the calculation of in-cylinder temperature and trapped mass [14,15].

2. Experimental Configuration

Experiments were conducted on a modern turbo-charged inline 6 cylinder Cummins diesel engine with common rail injection (ISBe220 31) at the QUT Biofuel Engine Research Facility (BERF). The engine has a capacity of 5.9 l, a bore of 102 mm, a stroke length of 120 mm, a compression ratio of 17.3:1 and maximum power of 162 kW at 2500 rpm. The engine was coupled to an electronically controlled hydraulic dynamometer with load applied by increasing the flow rate of water inside the dynamometer housing. In-cylinder pressure was measured by a Kistler piezoelectric transducer with a Data Translation (DT9832) simultaneous analogue-to-digital converter connected to a desktop computer running National Instruments LabView. Data was collected at a sample rate of 200,000 samples per second.

Specific data collected was in-cylinder pressure, band-pass filtered in-cylinder pressure (4-20 kHz, both collected as a differential voltage signal), injection information and crank-angle information. The engine was run at 2000 rpm on neat diesel and with ethanol fumigation substitutions of 10%, 20%, 30%, 40% and 50% on an energy basis at full, three-quarter and half loads. Energy basis in this context is defined as the amount of ethanol required to increase the engine torque from a depressed value back to the set point. The depressed value was a reduction in torque by 10%, 20%, 30%, 40% and 50%, corresponding to the ethanol substitution. For example, at 2000 rpm / full load the engine delivers 760 Nm of torque; therefore, at full load 2000 rpm D080E020 represents an initial diesel torque of 608 Nm with the remaining 152 Nm made up by ethanol fumigation substitution. The injection timing was controlled by the engine management system which varied the injection timing from 356 to 358 crank-angle degrees during the experiments. This change represents a change in volume of less than 2% in the combustion chamber and hence should not have had a significant effect on the outcomes of the experiments.

3. Methodology

In-cylinder pressure is commonly used to investigate the indicated work, indicated power, mean effective pressure, peak pressure, maximum rate of change of pressure, heat release, and thermal efficiency of engines [16-18]. This information is determined by observing various aspects of pressure versus volume and pressure versus crank-angle data. Figure 1 shows an example of pressure versus crank-angle data. As work is related to pressure, investigating in-cylinder pressure can yield many insights into combustion phenomena.

The area located at the peak of Figure 1 is related to the auto-ignition of the fuel. When isolated and analysed; the dominant frequency through this area is the first circumferential mode frequency and henceforth will be known as the combustion resonance [14,15]. As this resonance is caused by the combustion of the fuel, it can be used to accurately determine the precise location that combustion occurs. Figure 2 shows an example of a signal that will be used to find this; Figure 2 is a band-pass (4-20 kHz) filtered pressure signal.

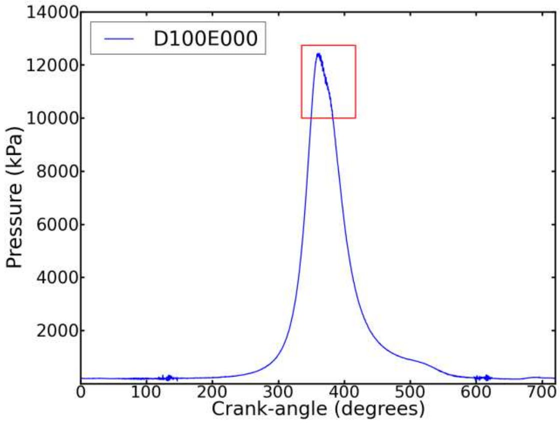

Figure 1 Pressure vs crank-angle, 2000 rpm, full load, neat diesel fuel

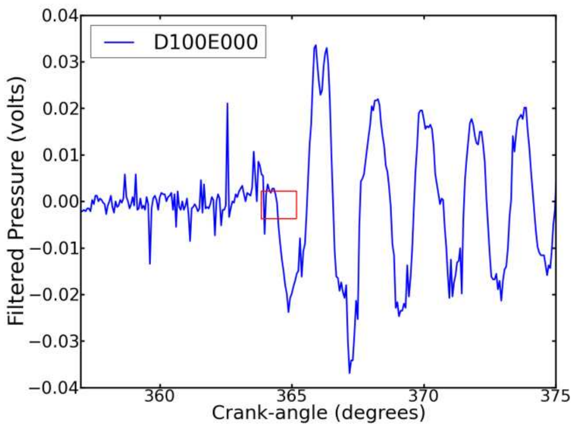

Figure 2 Band-pass filtered pressure signal at 2000 rpm, full load, neat diesel fuel from highlighted area of Figure 1

The point of injection is obtained directly by interrogating the electronic diesel injector driver signal. However, if this is not feasible it is still possible to use the filtered data to determine the point of injection by observing the level of high frequency fluctuation on the signal. The data in Figure 2 starts exactly where the injection occurs. Fluctuations in pressure associated with diesel injection can be seen between 356 and 364 crank-angle degrees. Ignition can be seen to commence at approximately 364 degrees crank-angle after which a strong resonance can be seen for the rest of the signal displayed in Figure 2. An extensive analysis of the combustion resonance has been undertaken in [15]. For the purposes of this investigation the ignition delay is defined as the number of crank-angle degrees from injection, detected from the electronic diesel injector driver signal, to the point of ignition which was taken to be when the signal no longer exhibited noise-like behaviour and combustion resonance commenced.

In this preliminary study the data was analysed manually. For each engine condition 200 consecutive cycles were used. Analysing this data in an automated process has some limitations; namely, the noise-like beginning can present itself in a variety of ways. Some promising results have been obtained using autocorrelation models and some preliminary work was also undertaken using Bayesian models. Additionally, for the basis of testing any automated methods, substantial results would be required for comparison.

4. Results

Figure 3 shows the 2000 rpm, full load results running the engine on neat diesel fuel and 10, 20, 30, 40 and 50% ethanol substitutions. The results are presented as a kernel density estimate, a probability density function (pdf), based on 200 consecutive cycles for each engine condition. It can been seen that on neat diesel fuel and at 10 and 20% ethanol substitutions there is minimal difference in the timing of ignition delay. However, it is also clearly seen that the widths of these three distributions are not the same. A 10% ethanol substitution has resulted in a much higher peak, indicating lower inter-cycle variability. When using neat diesel fuel and at 10 and 20% ethanol substitutions the pdfs exhibit a slightly bimodal distribution. Moreover, once the substitution exceeds 20%, there is a significant decrease in ignition delay which conflicts with the current literature [8-10,12,13]. Interestingly, the 40% ethanol substitution is showing less of a decrease in ignition delay than the 30% ethanol substitution. This is attributed to having only analysed 200 cycles. Because of these inter-cycle changes it is desirable to use a minimum of 800 consecutive cycles; this value will be dependent on individual engine testing setups. However, for the purpose of this investigation analysing more cycles was not necessary. What is important in this find is that there is a substantial decrease in ignition delay at some point between 20 and 30% ethanol substitution.

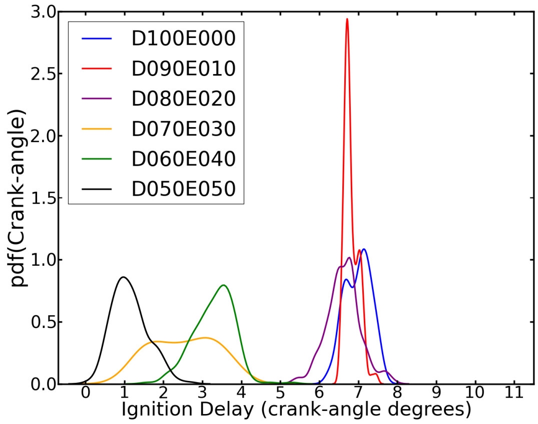

Figure 3 Pdfs of Ignition delay, 2000 rpm, full load.

As shown in Figure 4, at three-quarter load there is a slight increase in ignition delay at low ethanol substitutions (10% and 20%), and a slight decrease in ignition delay at 30% ethanol substitution. At 40% ethanol substitution there is a substantial decrease in ignition delay. There is no 50% substitution at this load due to high levels of engine knock.

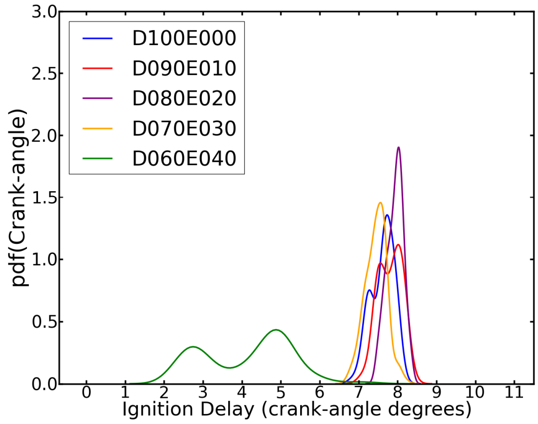

Figure 4 Pdfs Ignition delay, 2000 rpm, three-quarter load

Figure 5 shows that as the ethanol substitution increases the ignition delay also increases. This is in agreement with the current literature [8-10,12,13]. An exception to this trend is found for the 40% ethanol substitution, although it is still showing retarded ignition compared to neat diesel fuel operation.

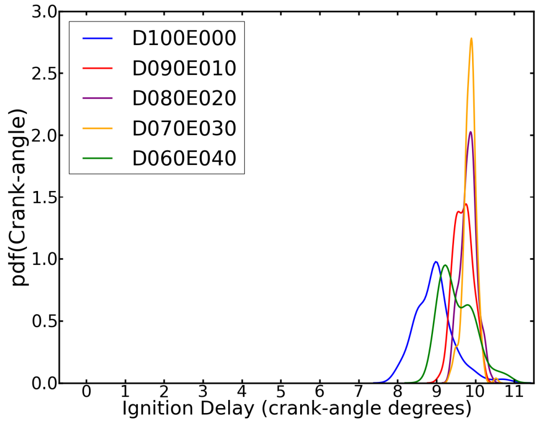

Figure 5 Pdfs of Ignition delay, 2000 rpm, half load

In order to investigate the unexpected reduction in ignition delay with increasing ethanol substitution a further experiment was conducted. This experiment aimed to determine if auto-ignition of ethanol was possible in our engine setup. The engine was run at 2000 rpm, full load at a 50% ethanol substitution. Once the engine operation was established and the data acquisition had begun, the diesel fuel to cylinder one was switched off. The corresponding in-cylinder pressure signal was then monitored for approximately 170 cycles.

Results for the neat diesel cases were compared by examining net rate of heat Release diagrams created using AVL BURN. The variation in calculated ignition delay was found to be less than 1 crank-angle degree from the method shown in this paper.

Figure 6 shows 5 in-cylinder pressure traces. The top trace depicts a representative cycle of the engine running at 2000 rpm, full load with 50% ethanol substitution. The next 3 traces show ignition whilst there was no diesel present, the highest of these 3 traces was from the cycle immediately after the diesel was switched off. The third trace displayed in Figure 6 is from the second cycle after the diesel was switched off, and the trace following this was a representative cycle of the established operation of the cylinder without diesel fuel, running on ethanol only approximately 30 cycles after the diesel was switched off. The last trace shown is of the cylinder being motored, with no diesel fuel and no ethanol.

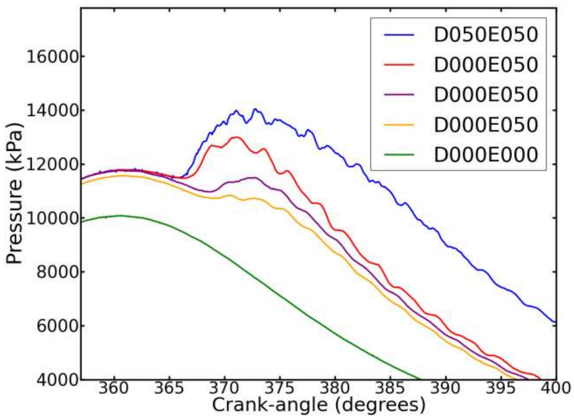

Figure 6 In-cylinder pressure traces showing evidence of auto-ignition in ethanol-only operation of a diesel engine

5. Conclusion

Compared with other types of diesel engines, common rail diesel engines have much later diesel injection timing. Injection in a common rail diesel engine happens near top dead centre (TDC) – the location where the piston is at its highest point. In standard diesel engines injection occurs well before TDC, often 15-20 degrees before TDC. This has a large effect on ignition delay. As a consequence of later injection, the compressed charge air is at a higher temperature when the fuel is injected.

The air/ethanol mixture in a common rail diesel engine undergoes substantially more compression before the diesel fuel is introduced. Figure 6 shows that under a motored condition the in-cylinder pressure exceeds 10 MPa. With additional heat, the peak in-cylinder pressure prior to combustion is almost 12 MPa. This increase in pressure, and the corresponding increase in temperature, allows for the auto-ignition of ethanol. In the case where there is a substantial amount of ethanol the effect of this is easily seen, Figures 3 and 4.

At lower loads, and hence lower in-cylinder temperature and a lower ethanol/air ratio, the results mirror those of diesel engines which have earlier injection found in the literature. This result is unsurprising. It is at the higher loads and ethanol substitutions where the interesting outcomes of this work can be found, showing that under high load and ethanol substitutions ignition delay is substantially decreased. Figure 3 shows that this reduction in ignition delay was as large as 7 degrees crank-angle.

Given the effect that ignition delay has on important engine factors such as emissions and noise, this paper gives evidence to suggest that research is needed in the use of fumigated fuels in common rail diesel engines.

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7. References

[1] S. Shafiee and E. Topal, Energy Policy **37** (1) (2009), pp. 181-189.

[2] Quartely Update of Australia's National Greenhouse Gas Inventory, Aust National Greenhouse Accounts, 2010.

[3] P. McArdle, P. Lindstrom, S. Calopedis, Energy Information Administration Tech. Report. DOE/EIA-0573, Office of Integrated Analysis and Forecasting, U.S. Department of Energy, Washington, DC, 2006.

[4] B. Skelton, Process Safety and Environ. Protection **85** (2) (2007), pp. 347-347

[5] A.C. Lloyd and T.A. Cackette, J. Air & Waste Mange. Ass. **51** (6) (2001), pp. 809-847

[6] M. Alperstein, W.B. Swin, P.H. Schweitzer, SAE Paper Number 580058, 1958.

[7] R. Diesel, Internal Combustion Engine, Letters Patent No. 608,845, 1898.

[8] T.K. Hayes, L.D. Savage, R.A. White, SAE Paper Number 880497, 1988.

[9] M. Abu-Qudais, O. Haddad, M. Qudaisat, Energy Convers. Mange. **41** (2000), pp. 389-399.

[10] B.S Chauhan, N. Kumar, S. Sunder Pal, Y.D Jun, Energy, **36** (2011), pp. 1030-1038.

[11] T.A. Bodisco and R.J. Brown, 17th Australasian Fluid Mechanics Conference Proc. (2010).

[12] G. Prakash, A. Ramesh, A.B. Shaik, SAE Paper Number 1999-01-0232, 1999

[13] Z. Sahin and O. Durgun, Energy Converse. Mange. **48** (2007), pp. 1952-1964.

[14] R. Hickling, D.A. Feldmaier, F.H.K. Chen, J.S. Morel, J. Acoust. Soc. Am. **73** (4) (1983), pp. 1170-1178.

[15] T. Bodisco, R. Reeves, R. Situ, R. Brown, Mech. Sys. Sig. Proces. **26** (2012), pp. 305-314.

[16] C. Amann, SAE Paper Number 852067, 1986.

[17] J. Heywood, Internal Combustion Engine Fundamentals, McGraw-Hill, 1988.

[18] A. Randolph, SAE Paper Number 900170, 1990