Understanding the relationship between ignition delay and burn duration in a constant volume vessel at diesel engine conditions

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Abstract

Experiments were performed in a constant volume vessel, with fuel sprays injected into the vessel at various different pressure and temperature conditions chosen to represent diesel engine operation at various loads.

A range of diesel primary reference fuels (i.e. mixtures of cetane and heptamethylnonane) of varying cetane number (CN) were tested, and as expected lower CN fuels have longer ignition delays. Burn period was plotted against ignition delay and two distinct trends can be seen: "mainly diffusion" diesel combustion in which burn period decreases with ignition delay and "mainly pre-mixed" diesel combustion in which burn period increases with ignition delay. There is typically a minimum in plots of burn period versus ignition delay which represents the transition between the two types of combustion mode. Higher CN, higher engine load and higher boost pressure favour "mainly diffusion" combustion whilst lower CN, lower loads and non boosted conditions favour "mainly pre-mixed" combustion.

Keywords

Diesel, Spray, Cetane Number, Ignition Delay, Burn Duration,

1. Introduction

Diesel vehicles (both passenger cars and heavy duty vehicles) are required to meet increasingly stringent controls on particulate matter (PM) and oxides of nitrogen (NOx), whilst at the same time operating with high efficiency to minimize greenhouse gas emissions. Although exhaust gas aftertreatment systems remain critical to achieving the emissions norms, optimized combustion to reduce engine-out emissions also plays a pivotal role.

Engine-out PM can be reduced by improving fuel-air mixing, so that a smaller fraction of combustion occurs in ultra-rich regions. Optimized fuel injection and swirl play an important role, but some advanced diesel combustion concepts also involve injection earlier in the cycle (often combined with lower compression ratio) to allow more time for mixing [1, 2, 3, 4].

Engine-out NOx reduction is usually achieved by exhaust gas recirculation (EGR) to reduce the combustion temperature. In traditional diesel combustion, increasing EGR tends to increase local mixture fraction φ, leading to higher PM emissions. However advanced diesel combustion modes with a very high degree of pre-mixing can allow a simultaneous reduction in NOx and PM [5, 6]. Some workers have gone as far as to suggest that gasoline-like fuels are the best for this type of combustion system because their autoignition resistance means that they can be injected very early, thereby allowing ample time for pre-mixing [7, 8, 9].

The original Sandia conceptual model of diesel combustion [10] envisages a steady state diffusion flame in which most of the combustion occurs under very rich conditions within the head of the jet. However this model is not valid when most of the combustion takes place after injection has finished and the Sandia group have developed new conceptual approaches to describe diesel combustion modes with a high degree of pre-mixing [11].

The literature contains examples of diesel combustion systems whereby a higher pre-mixed burn fraction gives rise to a greater level of combustion noise which can be mitigated with increased fuel Cetane Number (CN); see Hartikka and Nuottimäki [12] for a recent example. The reason generally given is that a longer ignition delay gives more time for fuel and air to mix prior to auto ignition and propagation of flame through the pre-mixed region is faster than via a diffusion flame. Faster flame propagation causes a higher maximum rate of pressure rise and hence more combustion noise.

However the relationship between cetane number and burn rate/combustion noise is not as straightforward as commonly supposed, especially for highly pre-mixed diesel combustion systems described by the new Sandia conceptual model [11]: In recent work Hu et al. [13] have presented results in which n-heptane and iso-octane were injected into a constant volume combustion vessel at pressure and temperature conditions representative of a diesel engine. N-heptane has a CN which is representative of a typical European diesel fuel, albeit much more volatile, whereas iso-octane is extremely resistant to autognition. Because of the long ignition delay, iso-octane was found under some conditions to have become too mixed and dilute for autoignition to occur at all. In higher temperature and pressure conditions where iso-octane did ignite, the combustion was found to be much slower than for n-heptane under the same conditions.

The objective of this work is to explore the relationship between ignition delay and burn rate in a systematic way. To this end a variety of diesel primary reference fuels were tested in a constant volume combustion vessel operated under a variety of different conditions chosen to represent diesel engine operation at various loads. The relevance of the results to actual engines is discussed.

2. Experimental

The Combustion Research Unit (CRU) in Shell Global Solutions is a constant volume vessel, manufactured by Fueltech that can mimic combustion conditions in modern diesel engines. A schematic diagram of the CRU is shown in Figure 1.

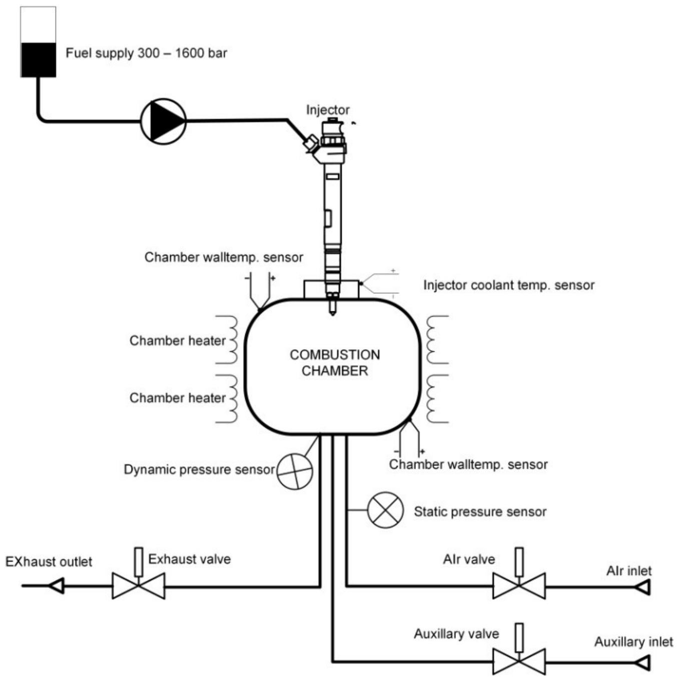

Figure 1: Schematic diagram of Combustion Research Unit*. (Reproduced with permission from Fueltech)*

Table 1: Operating parameters of CRU

Parameter	**CRU Mark II**
Initial chamber pressure	2 – 75 bar
Initial chamber wall temperature	350 – 590°C
Fuel pressure	200 – 1600 bar
Main injection pulse width	0.3 – 1.5 ms
Pilot injection pulse width	0 – 1.4 ms
Pilot-main separation	0.1 – 3.0 ms
% Auxiliary gas	0 - 100%

The unit is supplied with a common rail injection system of type Bosch CRIP2 (PartNo: 0445110157) and a 7 hole nozzle. Fuel is injected into the pressurized heated chamber where it mixes with hot air and ignites.

The combustion process is monitored with a pressure sensor inside the chamber whilst a needle lift sensor inside the injector monitors the injection event. The chamber pressure, chamber temperature, fuel pressure, chamber gas composition and injector pulse width can all be varied by the operator. Some technical parameters of the CRU are listed in Table 1.

The needle lift sensor and the two dynamic pressure sensors in the combustion chamber and fuel line all sample at a rate of 50 kHz (intervals of 0.02ms), giving outputs including needle lift, chamber pressure and fuel pressure. The needle lift enables the measurement of the start of injection (SOI) and the end of injection (EOI). The chamber pressure is used to determine the ignition delay and burn period. The specific definitions of ignition delay and other terms used in this paper are given in Table 2.

Table 2: Definitions of ignition delay and burn period used in this work.

Parameter	Definition	Comment
SOI	Start of injection	
EOI	End of injection	
PW	Pulse Width	EOI -SOI
MPI	Maximum pressure increase	The peak pressure in the smoothed *p-t* plot minus the initial pressure
$ID^{5\%}$	Ignition delay 5%	The time from t = *SOI* to the moment when the chamber pressure is equals its initial value plus 5% of *MPI*.
$BP^{10\text{-}90\%}$	10-90% Burn	The time from the moment

	period	where the chamber pressure equals its initial value plus 10% of *MPI* to the moment when the chamber pressure equals its Initial value plus 90% of *MPI*.

The CRU was operated under a number of conditions to represent different points in the speed load map of a typical diesel vehicle (Table 3).

Table 3: Conditions tested in the combustion research unit.

Condition	**Vessel temperature T_i / °C**	**Vessel pressure p_i / bar**	**Fuel rail pressure $p_{i,fuel}$ / bar**	***Injector pulse width PW* / ms**
A	590	75	900	0.9
B	590	65	900	0.9
C Max Power	590	65	1600	1.5
D	590	48	900	0.9
E Idle	590	48	400	0.6
F	590	25	900	0.9

The primary reference fuels, PRFs, used for cetane measurement are cetane (n-hexadecane), which has a designated cetane number of 100, and heptamethylnonane (HMN or 2,2,4,4,6,8,8- heptamethylnonane), which has a designated cetane number of 15. Note that these are different to the gasoline primary reference fuels n-heptane and iso-octane.

By definition, mixtures of cetane and HMN have a cetane number (CN) given by the linear relationship:

$$CN = \text{cetane (\%vol.)} + 0.15\ \text{HMN (\%vol.)} \tag{1}$$

Various PRFs from 15CN to 100CN were blended for use in the CRU (Table 4). Cetane was 99% purity from Alfa Aesar and HMN was 98% purity from Aldrich

Table 4   Primary Reference Fuels Used.

Cetane %vol.	**HMN %vol.**	***CN***
100	0	100
90	10	91.5
80	20	83
70	30	74.5
60	40	66
50	50	57.5
45	55	53.25
40	60	49
35	65	44.75
30	70	40.5
20	80	32
10	90	23.5
0	100	15

3. Results

A typical set of pressure traces is shown in Figure 2. The example chosen is for condition C, which was chosen to be representative of diesel combustion at full load. As would be expected, the ignition delay reduces with increased cetane number. A pre-mixed combustion phase can be easily discerned which is characterized by a much more rapid rate of pressure rise – this is followed by a slower rate of pressure rise in which the combustion is mixing controlled (N.B. the end of injection occurs around after combustion has begun for the higher CN fuels) It can be seen also that the fuel with the highest CN has the lowest pre-mixed pressure rise and vice versa. This is consistent with a conventional view of the effect of higher cetane reducing the amount of pre-mixed combustion. The fact that what we have termed the pre-mixed pressure rise is greatest for the lower CN fuel with more time for mixing provides confirmation that the higher initial heat release rate is indeed mainly due to a physical mixing phenomenon rather than a cool flame.

Figure : Average *p-t* plots for *CN* = 40.5, 53.25 and 83 PRFs at condition C designed to represent full load. The pre-mixed and mixing controlled contributions to the pressure rise are marked for the *CN* = 83 PRF. The black spots mark 5% of maximum pressure increase, a point which is used to calculate $ID^{5\%}$

Figure 3 shows a plot of burn period versus ignition delay for condition C (full load) (see definitions of $BP^{10\text{-}90}$ and $ID^{5\%}$ in Table 2). It can be seen that the burn period decreases with increasing ignition delay which is indicative of a greater proportion of pre-mixed combustion. The pulse width for condition C is 1.5 ms so that combustion and injection are occurring simultaneously for all but the fuels with the lowest CN.

Figure 4 shows the same plot for condition E which is designed to be representative of idle or low load. For this condition the injection pressure is much lower and the pulse width is only 0.6 ms, so that injection and combustion are largely separate events. However in this case the plot of BP versus ID goes through a minimum and then increases with increased ignition delay. Slower combustion results from the fuel-air mixture becoming progressively more dilute with increased ignition delay. The results for the lowest CN numbers (CN15 and CN23.5) are not shown on figure 4 because the combustion is so slow as to be off the scale.

Figure 3: Dependence of burn period ($BP^{10\text{-}90}$) with $ID^{5\%}$ for different primary reference fuels (PRFs) at condition C (designed to represent full load). The results for PRF15 are not shown.

Figure 4: Dependence of burn period ($BP^{10\text{-}90}$) with $ID^{5\%}$ for different primary reference fuels (PRFs) at condition E (designed to represent idling conditions). The results for PRF15 and 23.5 are not shown.

For the full load condition (Figure 3) the results for CN15 which are not shown on the figure gave an ID of ~5ms and a BP of ~11ms. This is contrary to the trend displayed by the other fuels suggesting it is possible to switch combustion mode (between pre-mixed and mixing controlled combustion) under all conditions if a sufficiently low CN fuel is used.

Figure 5 illustrates the effect of vessel pressure on the burn period and ignition delay. The vessel temperature, injector rail pressure and pulse width were constant for the conditions shown (see Table 3). At p_i =75bar, the pre-dominant trend is for BP decrease with ID for as the fuel becomes progressively more pre-mixed. A minimum in BP occurs around an ID of 1.5ms, after which there is a slight rise in burn period as the fuel air mixture becomes more dilute. At lower pressures the trend to increasing BP becomes more important until at p_i =25bar the pre-dominant trend is for BP to increase with ID for all fuels. Note that the position of the minimum is constant for all 4 cases shown in Figure 5 but will depend on factors such as the pulse width and injector rail pressure.

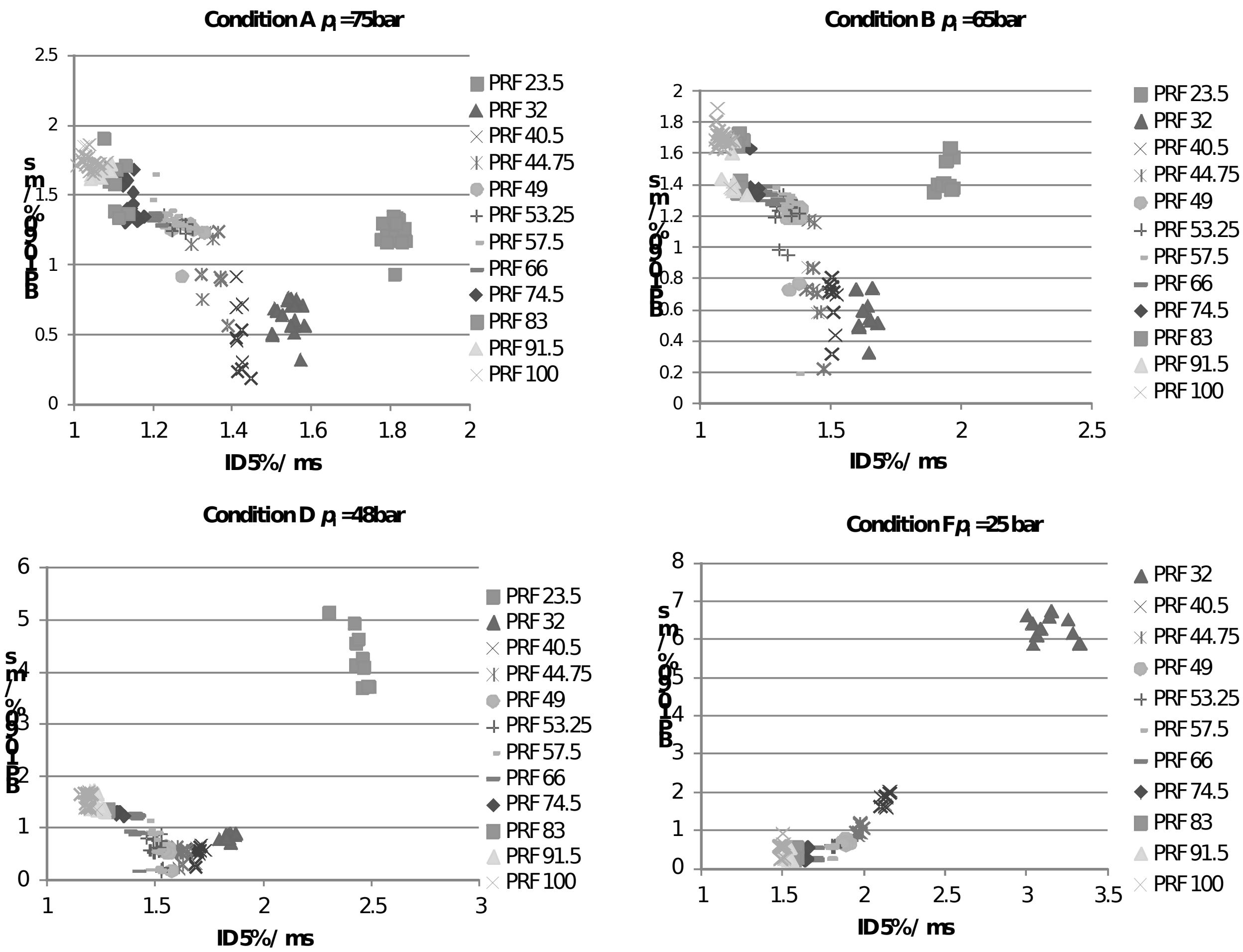

Figure 5: Dependence of burn period (BP^{10-90}) with $ID^{5\%}$ for different primary reference fuels (PRFs) and 4 different vessel pressures. The injector pressure, pulse width and vessel pressure are kept constant

The reason for the change in curve shape with vessel pressure lies in the relationship between ignition delay and pressure (Figure 6). The relationship between $ID^{5\%}$, CN and vessel pressure (for conditions A, B, D, F) can be approximately fitted to

$$ID5\%\ (ms)=(7.12-0.028CN)pi(bar)-0.335 \quad (2)$$

The decrease of ID with p_i is well known for homogenous systems [14, 15]. For diesel–like systems Heywood [16] cites examples [17, 18] of ID decreasing with p_i. Nevertheless the impact on the combustion mode is striking: in figure 5, an ignition delay of 1.5 ms represents the minimum in the plot of BP vs. ID – this is the transition between “mainly diffusion” diesel combustion and “mainly pre-mixed” diesel combustion. In Figure 6 it can be seen that at p_i =25bar, a 100CN PRF fuel has an ID of 1.5 ms but fuels of lower cetane will have longer ignition delays than this and display the "mainly pre-mixed" combustion characteristic. By contrast at p_i =75bar, a 40.5 PRF fuel has an ignition delay of 1.5ms. At this pressure fuels with higher cetane numbers would have much shorter ignition delays and show the "mainly diffusion" combustion characteristic.

Figure 6: Dependence of ignition delay ($ID^{5\%}$) with vessel pressure for different primary reference fuels (PRFs). Vessel temperature = 590ºC, fuel rail pressure = 900bar, injector pulse width = 0.9 ms.

4. Discussion

The CRU has been run at a number of different conditions representing different speed/load points in an engine using different diesel PRFs of varying CN.

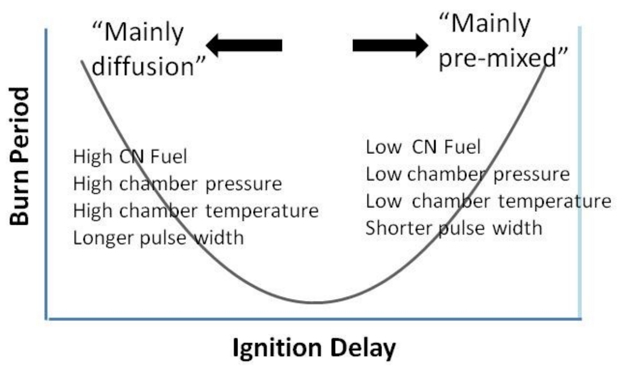

Figure 7: Schematic diagram showing the dependence of burn period with ignition delay for different primary reference fuels.

A plot of burn period versus ignition delay is found to show a characteristic parabolic shape (figure 7) which can be divided in to two regions:

> **"Mainly diffusion"**: increasing burn rate (decreasing burn period) with increasing ignition delay; a longer ignition delay gives more time for fuel and air to mix prior to auto ignition and propagation of flame through pre-mixed region is faster than via a diffusion flame
>
> **"Mainly pre-mixed"**: burn rate slows down (increasing burn period) with increasing ignition delay. The ignition delay becomes very large causing the fuel and air to pre-mix to such an extent that mixture becomes lean.

The factors which affect the transition between "mainly pre-mixed" and "mainly diffusion" combustion are as follows:

Fuel Cetane Number: The higher the CN, the shorter the ignition delay.

Chamber pressure: The higher the chamber pressure the shorter the ignition delay (Figure 6). This means that boosted (i.e. supercharged or turbocharged engines) are more likely to operate with diffusion combustion modes.

Temperature: Arrhenius-like behaviour would suggest that a higher temperature would lead to shorter ignition delays, although the possibility of negative temperature coefficient effects cannot be discounted.

Pulse width: The smaller the amount of fuel injected the more likely the fuel is to be in the "mainly pre-mixed" combustion region. The minimum occurred in conditions A,B,D and F (each with a 0.9ms pulse width) at around 1.5 ms. For conditions C and E with longer and shorter pulse widths respectively (and different rail pressures), the minima occurred later and earlier respectively.

The conditions employed in the CRU were designed to be representative of a range of different loads in a diesel engine, but there are a number of features of engine combustion, such as swirl that cannot be easily replicated in a constant volume vessel. Additionally this study has assumed that that the gas in the CRU is air, when in reality exhaust gas re-circulation (EGR) is an extremely common approach for reduction of engine-out NOx. The question arises therefore as to whether the distinct regions of combustion identified in this paper and shown schematically in Figure 7 also be identified in engine experiments?

A joint study between CONCAWE, FEV and RWTH Aachen has investigated how fuel properties could influence the effectiveness of advanced diesel combustion technologies [1, 2, 3]. A fully warmed-up diesel bench engine was able to achieve Euro 6 NOx emissions limits without additional NOx aftertreatment on a wide range of test fuels with different properties. The engine hardware enhancements used to achieve low engine–out NOx performance included a lower compression ratio, higher maximum cylinder peak pressure and rail pressure, adjustment of fuel injection timing, and intensified exhaust gas recirculation (EGR). The engine operation simulated closed loop combustion control (CLCC) by matching the point in the cycle at which 50% of the fuel's heat is released (CA50) and optimizing the combustion performance on each test fuel. Figure 8 shows the combustion sound level (CSL) plotted against CN for various fuels investigated in [2]. CSL is proportional to the maximum rate of pressure rise and it can be seen that at the higher part load points CSL decreases with increased cetane, whilst at the lower part load points CSL increases with cetane number. In other words the engine described in [2] is operating in a "mainly diffusion" mode at the higher part load points and in "mainly pre-mixed" mode at the lower part load points.

Figure 8: Relationship between Combustion Sound Level and Cetane Number in a diesel bench engine capable of meeting Euro 6 NOx emissions limits without additional NOx aftertreatment. Data re-plotted from reference [2].

5. Conclusions

Experiments were performed in a constant volume vessel, with fuel sprays injected into the vessel at various different pressure and temperature conditions chosen to represent diesel engine operation at various loads. A range of diesel primary reference fuels (i.e. mixtures of cetane and heptamethylnonane) of varying cetane number (CN) were tested, and as expected lower CN fuels have longer ignition delays.

Burn period was plotted against ignition delay and two distinct trends can be seen: “mainly diffusion” diesel combustion in which burn period decreases with ignition delay and “mainly pre-mixed” diesel combustion in which burn period increases with ignition delay. In “mainly diffusion” combustion a higher degree of mixing favours faster combustion, whereas in “mainly pre-mixed” diesel combustion higher ignition delay can lead to progressive enleanment of the fuel and hence to slower combustion.

The trends observed in the CRU were compared with trends in combustion noise in published engine experiments at different speed/load points. From this it can be concluded that a constant volume vessel such as the CRU can represent many of the important features of diesel engine combustion.

Vessel pressure was found to have a significant effect - the higher the chamber pressure the shorter the ignition delay. The consequence of this is that boosted engines are more likely to operate in a diffusion combustion mode. More generally, higher CN, higher engine load and higher boost pressure favour “mainly diffusion” diesel combustion whilst lower CN, lower loads and non-boosted conditions favour “mainly pre-mixed” combustion.

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Fueltech are gratefully acknowledged for permission to publish the schematic diagram of the Combustion Research Unit.

Figure Captions

Figure 1: Schematic diagram of Combustion Research Unit. (Reproduced with permission from Fueltech)

Figure : Average *p-t* plots for CN = 40.5, 53.25 and 83 PRFs at condition C designed to represent full load. The pre-mixed and mixing controlled contributions to the pressure rise are marked for the CN = 83 PRF. The black spots mark 5% of maximum pressure increase, a point which is used to calculate $ID^{5\%}$

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