HCCI WITH SELECTED STANDARD AND ALTERNATIVE FUELS: CHALLENGES AND SOLUTIONS

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Abstract

Work with different fuels: gasoline, gasoline - diesel blends and propane at the authors' Future Power Systems Group in Birmingham University reveals great potential for the use of moderate compression ratio engines (from 10.4 to 15), particularly when additional engine facilities such as EGR (external and internal via negative valve overlap and low lift cams), some degree of intake heating and/or hydrogen addition (from fuel reforming) are employed. Possibilities, benefits and demands of these technologies are outlined. There is a brief outline of the team's work on modelling of the kinetics of autoignition in HCCI using CHEMKIN and its interrelation with the gas dynamics of the engine system as studied with the Ricardo WAVE platform. The trade-off between the single-zone heat release model and the multi-zone model is outlined, as the latter can offer much better accuracy, particularly with reference to emissions, but at the cost of assumptions/validation techniques ranging from a well educated guess to more advanced optical diagnostics. The team who contributed to this work includes colleagues from Birmingham University, Jaguar Cars Ltd, Johnson Matthey plc and Shell Global Solutions.

Keywords: HCCI, gasoline, propane, mixtures of gasoline and diesel, hydrogen, exhaust gas fuel reforming, moderate compression ratios

1. Introduction

HCCI is known as an attractive combustion mode on account of its potential for achieving high efficiency and ultra-low NOx emissions because of low temperature combustion achieved under uniformly lean condition, has been studied for over twenty years since its potential benefits were first realized and demonstrated [1, 2]. Since then a significant amount of work has been done particularly on four stroke gasoline HCCI engines. Initially high compression ratios (typically in the 15 to 21 region), and/or with intake air heating were used to initiate HCCI combustion. The effect of increasing the inlet charge temperature is to advance the auto ignition timing and decrease the combustion duration. The auto ignition process tends to have very rapid heat release rates, leading to violent combustion with very rapid pressure rise rates. Thus, charge dilution was provided in the form of excess air (very lean air fuel ratios) or by external exhaust gas recirculation (EGR). This dilution effectively slows down the rate of combustion [3-5]. The requirements for dilution limit the maximum power density of HCCI engines as violent combustion occurs when the excess air ratio (lambda) is reduced. As such, the maximum load achieved is dictated by the amount of air or EGR that can be inducted into the engine to provide dilution. Forced induction such as supercharging has been shown by Christensen et al. to be an effective method in raising the power density of HCCI engines [6]. With sufficiently high intake manifold pressures, loads up to 16 bar indicated mean effective pressure (IMEP) were achieved by the same group with intake air heating in a diesel type engine fuelled with natural gas [7]. As such, it appears that increasing the dilution amounts is instrumental to increasing the useable load range for HCCI combustion as other methods such as stratification show increased NOx emissions and high cycle to cycle variation [8].

The HCCI ignition and combustion processes are mainly driven by chemical kinetics of hydrocarbon oxidation chemistry, thus with no definite controlling event, such as spark ignition or fuel injection, good understanding of the underlying chemistry and good control of in-cylinder temperatures and charge reactivity through engine breathing, actual compression ratio, residual gas trapping and fuel composition is needed for successful HCCI engine operation over increasing load and speed ranges. A great deal of work has been done in recent years and the research area has extended to all aspect of the combustion process. It has been gradually presenting a picture of energy saving and cleaner exhaust emissions.

Increasing environmental concerns regarding the use of fossil fuels and global warming have prompted researchers to investigate alternative fuels. Besides gasoline [9] and diesel fuel [10], a variety of alternative fuels, such as methanol [11], ethanol [12, 13], natural gas [14], biogas [15], hydrogen [16], DME [11] and their mixtures [17-19], including also gasoline and diesel mixtures and different mixtures of iso-octane with heptane [20], have been experimentally proved as possible fuels for HCCI combustion in both two-stroke and four-stroke engines. The authors have investigated almost all of aforementioned fuels on the same test bench as this study. Some of the publications and the results have listed in references [13, 15, 21-26]. This paper presents some results obtained with gasoline, gasoline-diesel blends and propane, while more results with natural gas, biogas and ethanol as well as challenges associated with HHCI operation of a multi-cylinder engine in a vehicle environment will be presented in a parallel paper later this month [27].

HCCI modelling is a relatively new area and thus a complete code for HCCI engine modelling is not yet available. Because of the fast occurrence of chemical reaction within the HCCI engine, a single zone model with an assumption of homogeneous in-cylinder mixture can be used to calculate the chemical kinetics of the combustion, despite that this assumption can be invalid in many cases. A parallel paper to be presented later this month [28] compares the results from a single zone chemical kinetic model coupled by a 1-d gas dynamic model and a more detailed consideration of non-homogeneous characteristics of the in cylinder gas (thus employing a multi-zone chemical kinetic model).

2. Equipment

Most engine tests that yielded results presented here were carried out using a “Medusa” single cylinder research engine (based on the design by Richard Stone of Oxford University) that was built in-house using a modified Rover K series cylinder head (Table 1). The engine was fitted with several different pistons and two different camshafts, thus providing different combinations of geometric compression ratios and valve lift. Valve timing could also be adjusted using vernier adjusted pulley, the timings of inlet and exhaust valves were set manually with the pulley before tests to acquire special valve strategies. Compression ratios between 10.4:1 and 15:1 were obtained using a standard Rover piston, a racing piston, and using a specially modified piston blank that was designed and machined in house. This adaptation also allowed the combustion chamber shape to be more similar to typical advanced engine designs used for future HCCI operation.

The engine was installed in a fully instrumented test cell, with all the auxiliary facilities required for the operation and control of the engine. A 3 kW electric air heater was installed in the intake duct to preheat the air required for HCCI operation with some fuels in the “intake preheat” mode.

A Kistler 6125A pressure transducer was fitted flush with the wall of the combustion chamber connected via a Kistler 5011 charge amplifier to a National Instruments data acquisition card fitted in an IBM compatible PC. A shaft encoder was used to provide synchronization crank angle degrees.

Table 1. Single cylinder “thermal” engine specification summary

Engine type	4- cycle, single cylinder
Bore x Stroke	80 x 88.9 mm
Connecting Rod Length (mm)	165 mm
Geometric Compression Ratios	Several values between 10.4 - 15.0
Fuelling type	Gaseous: Low pressure mains gas: Induction via gas carburettor, high pressure (bottled gasses or propane): injection into inlet port, upstream of inlet valve Liquid: port-injected, injection at 3 bar (gauge)

3. Experimental procedures and results

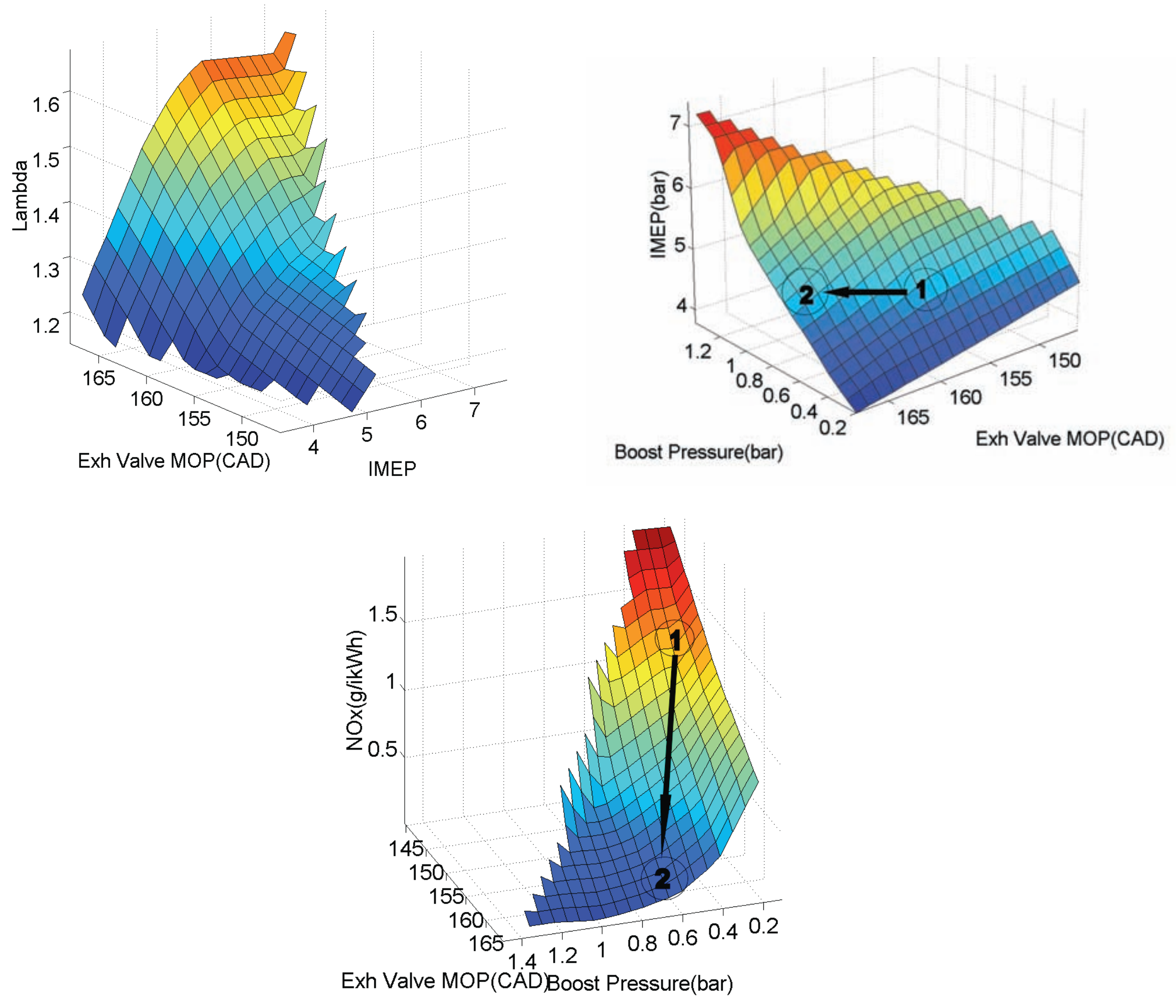
3.1. Procedures

The engine was also equipped with a traditional electronic spark ignition system. This system was used only for engine start-up, warm-up and SI mode; and were turned off when running the engine in HCCI mode. The engine was coupled to a DC dynamometer which maintained the engine at a constant set speed. For the 1-cyl engine tests, analysis software was developed in house using the LabVIEW programming environment to record the in-cylinder pressure versus crank angle for a representative number of consecutive engine cycles (usually 100), and to analyze the resulting data. Pressure data were not filtered electronically, but a numerical filter was used with normally three passes of 3-point averaging, and an adjustable trigger limit, normally set at 30% threshold for noise identification. Pressure oscillations were not studied. In all “knocking” cases audible knock criterion was used, while rates of pressure rise remained. the rate of pressure rise with crank angle dp/dtheta was generally below 5 bar/deg, only in one or two cases reaching 7 bar/deg, well below 10 bar/deg in all acceptable (i.e. non-knocking) cases. Heat release analysis was performed using the classic Rossweiler and Withrow method for mass fraction burnt modified to include continuous linear change of the polytropic index between the ignition point and end of combustion.

The engine was run in different modes and air/fuel ratios while the coefficient of variation of IMEP was kept below 5% whenever possible. Carbon dioxide, carbon monoxide, unburned hydrocarbons, oxygen and NO_x emissions were recorded using various standard emissions equipment.

3.2 Results with gasoline and gasoline-diesel blends

The recent main experimental effort on the single cylinder thermal engine has been to study the effect of boosting pressure and valve timing on the HCCI operating window boundary. Boosting is considered a promising way of increasing upper load boundary as it overcomes flow constraints associated with low valve lifts and enables lean operation reducing combustion NVH and NOx under high load conditions. For the trade-off of thermal efficiency and combustion smoothness, it has been found that the optimum phasing of 5% burn point should be controlled at TDC [22]. Figure 5a shows the lambda required in order to maintain the optimum combustion phasing for various exhaust valve timings and engine loads at 1500 rpm and it is shown that a higher load operating condition requires a higher lambda. This implies a higher boost pressure with advanced inlet valve timing. Meanwhile, there is another interdependence where lambda required for a specific combustion phasing varies with exhaust valve timing. The values of lambda required for stable combustion are highest with advanced exhaust valve timing and increased internal EGR. It is the combined dilution effects from the high rate EGR and the high air-fuel ratio that control the combustion rate and maximum in-cylinder pressure when approaching upper load boundary. Clearly, the extension of the upper load boundary is also limited by NOx emissions. Figure 1c shows that NOx emissions decrease as boosting pressure increases (for a higher lambda), or as exhaust valve timing advances (for a higher internal EGR rate). The two engine conditions marked '1' and '2' have the same engine load of 4.8 bar IMEP. It is shown that by increasing the boosting pressure from 0.4 bar to 0.6 bar and advancing the exhaust valve timing MOP from 158 CAD to 168 CAD, NOx emissions are significantly reduced from 1.12 to 0.16 g/kWh. This has clearly demonstrated the combined effect of boosting and internal EGR in NOx control for the extension of upper load boundary for HCCI operation.



(a) **(b)** **(c)**

Fig. 1. The effect of boosting and internal EGR on gasoline HCCI: (a) Lambda; (b) IMEP, (c) NOx. 1-cyl "Medusa" engine, non-heated inlet, nominal compression ratio (swept volume based) = 10.4, 1500 rpm, 5% burn point at TDC, Figs (b) and (c) IMEP = 4.8 bar [22]

Dual- and multi-fuels have been reported as a good method for promoting the HCCI combustion. A brief review of research is presented in a parallel paper by the authors [26]. Various researchers have found that diesel fuel has a high ignition quality to initiate effectively the onset of HCCI combustion. However, their methods generally needed two completely separate fuel supply systems in the engine, leading to larger system complexity. Pre-mixed blends of gasoline and diesel have also been successfully and comprehensively tested, although only in preheated intake mode with one value of excess air ratio lambda and at quite slow engine speed in a large cylinder (1.6 litre swept volume) [20].

The Birmingham FPS Group has investigated the HCCI combustion behaviour of the pre-mixed blends of gasoline and diesel (injected into the intake port at 3 bar injection pressure) as the two fuels with opposite but complementary properties, in a basically SI gasoline design "Medusa" engine. The engine was operated in two modes, using two different means to achieve the HCCI conditions: intake heating and NVO (negative valve overlap), with moderate compression ratios of 15 and 10.4, correspondingly. The intake heating mode brings benefits in easier evaporation and formation of the homogeneous charge especially for fuel mixtures with higher diesel fuel ratio, but the unheated NVO mode offers larger practical significance. No external EGR was used at all. Degree of internal EGR available in the NVO mode is not easily measurable. Methods of estimation of the internal EGR have been published by the authors [23].

In the heated intake mode, without internal EGR, the intake charge must be heated to over 370 K, for all fuel blends studied, even when compression ratio 15.0 is used. However, by adopting the gasoline/diesel blended fuel, the intake temperatures required to achieve HCCI can be lowered by at least 10 degrees compared with pure gasoline operation. Even more interestingly, with internal EGR (residual gas trapping) produced by NVO, appropriate conditions for HCCI combustion can be achieved without pre-heating even for the lower compression ratio. By adopting the gasoline/diesel blended fuel, the HCCI operating region for the unheated NVO can be significantly extended into lower IMEP values. Diesel fuel has remarkable influence on gasoline HCCI combustion. Ignition timing is advanced, duration of combustion is shortened and the IMEP range achievable in the unheated NVO mode (with acceptable COV of IMEP) is widened as diesel proportion increases, as shown in Figures 2 and 3.

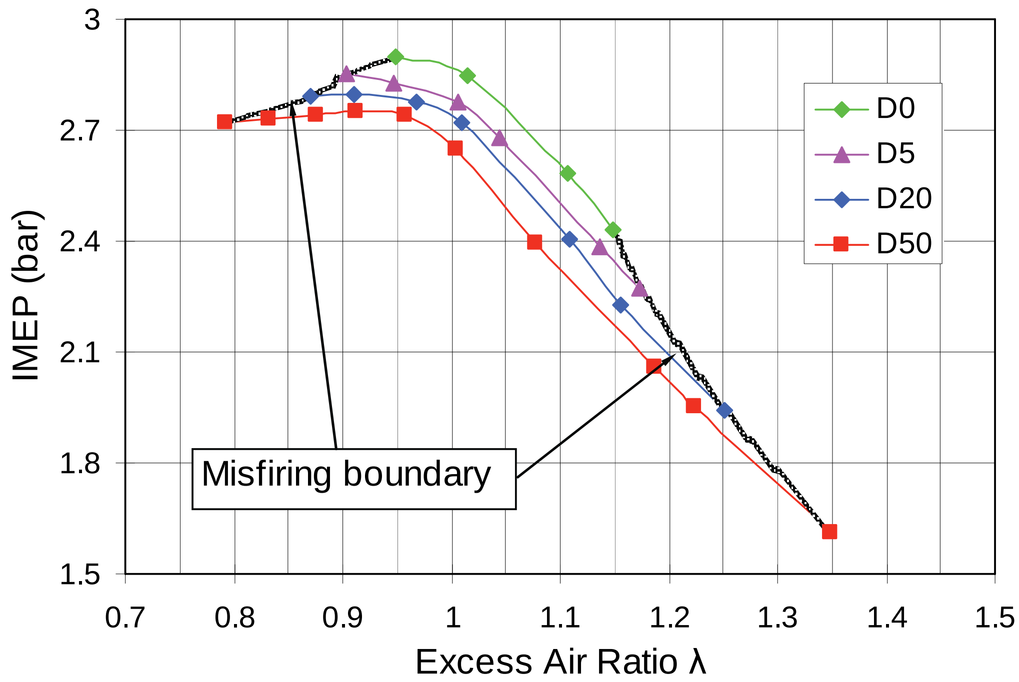


Fig. 2. Comparison of IMEP and lambda range possible with the four fuels of D0 (pure gasoline), D5, D10 and D50 when engine operates in NVO HCCI mode. 1500 rpm, unheated intake, low lift cams (3mm as opposed to standard 8mm) , NVO = -170 deg [26]

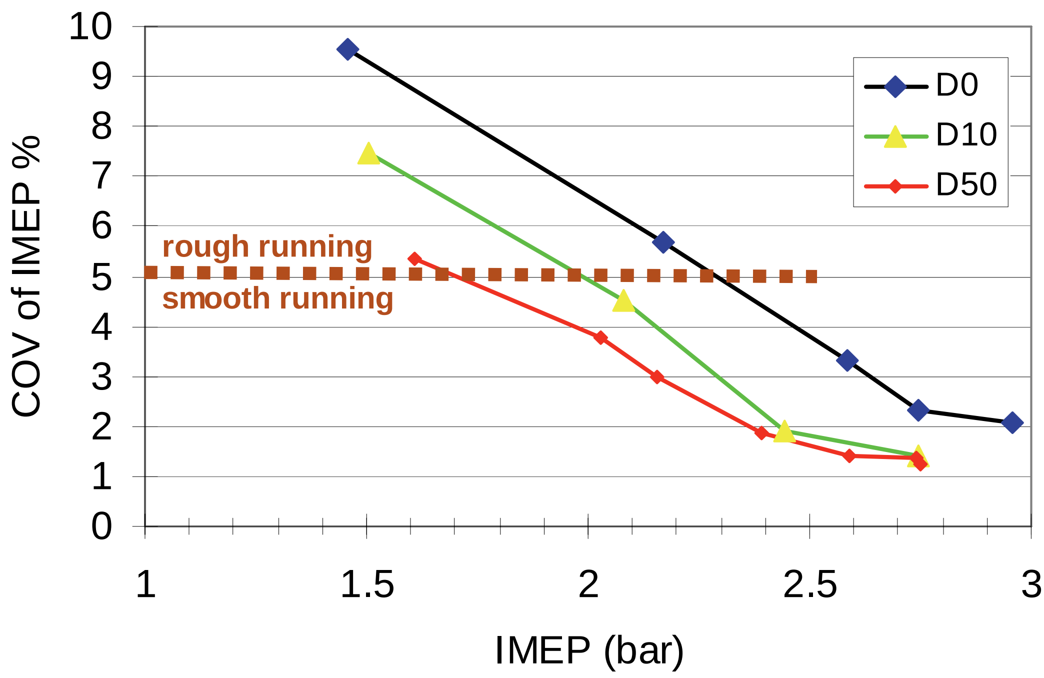


Fig. 3. Combustion stability comparison between three fuels: D0 (pure gasoline), D10 and D50 when engine worked with unheated NVO HCCI mode, geometric CR = 10.4, 1500 rpm [26]

Exhaust emissions for the gasoline-diesel blends, especially HC [26] and NOx shown in Figure 4, show a large improvement compared with pure gasoline HCCI. Unlike in other studies reported in the literature for the same operating conditions, blended fuel HCCI combustion in fact produced much less harmful emissions than pure gasoline HCCI combustion.

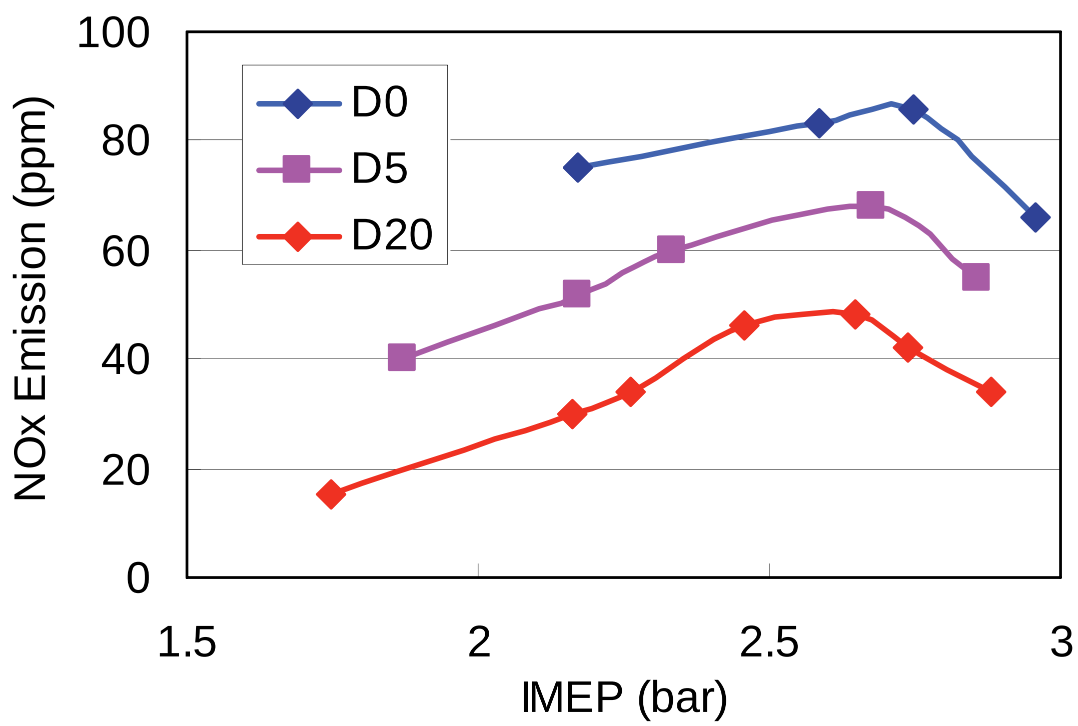


Fig. 4. NOx emissions for unheated NVO HCCI mode. 1500 rpm, unheated intake, geometric CR = 10.4, low lift cams, NVO = -170 deg [26]

3.3 Results with propane

Propane is available commercially for use in conventional internal combustion engines as an alternative fuel for gasoline. However, its application in the developing homogeneous charge compression ignition (HCCI) engines requires various approaches such as high compression ratios and/or inlet charge heating to achieve auto ignition. Previous work with propane in a diesel engine with a compression ratio (CR) of 18.8 and inlet air heating reaching temperatures of 140 deg C showed the viability of propane for use as an HCCI fuel [29]. The load range (IMEP) achieved was between IMEP 0.8 bar and 3.3 bar. As propane is intended for use as an alternative to gasoline, it is impractical to have such conditions for propane HCCI. Gasoline fuelled SI engines usually have moderate compression ratios and there is no much inlet air heating, thus making it challenging to achieve the high in-cylinder temperatures required for auto-ignition. The approach documented by the authors' research group in a recent paper [30] utilizes the trapping of internal residual gas (as used before in gasoline controlled auto ignition engines), to lower the thermal requirements for the auto ignition process. In that work, with a moderate engine compression ratio the achievable engine load range was controlled by the degree of internal trapping of exhaust gas supplemented by inlet charge heating. Increasing the compression ratio decreased the inlet temperature requirements; however, it also resulted in higher pressure rise rates, as seen in Figures 5 a and b.

a)

b)

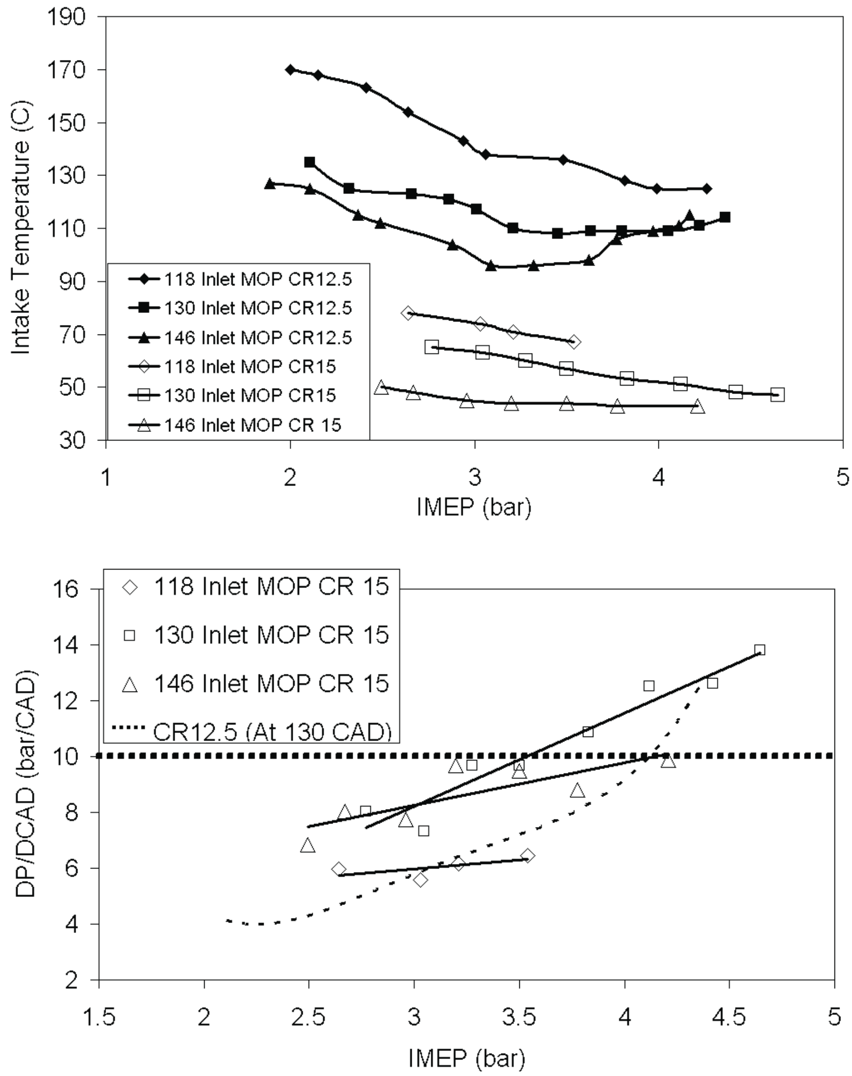


Fig. 5. Required intake temperature and rate of pressure rise for propane HCCI [30] 1500 rpm, lambda =1

Varying the inlet valve timing affects the combustion phasing, as seen in Figure 6, which can help to decrease the maximum pressure rise rates.

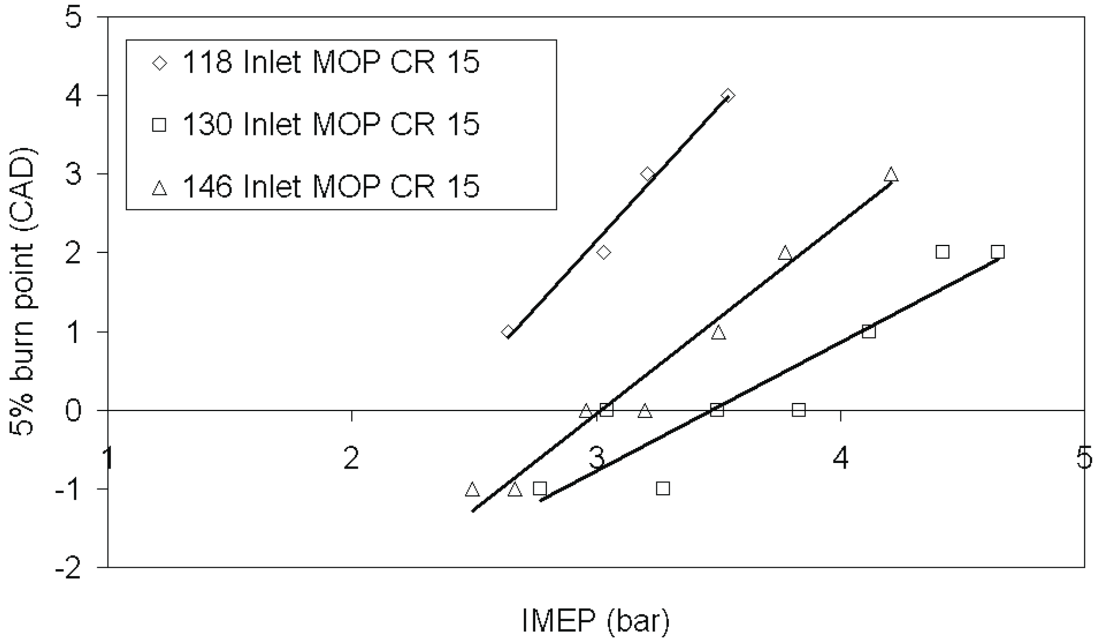


Fig. 6. Timing of ignition for propane HCCI [30] 1500 rpm, lambda =1

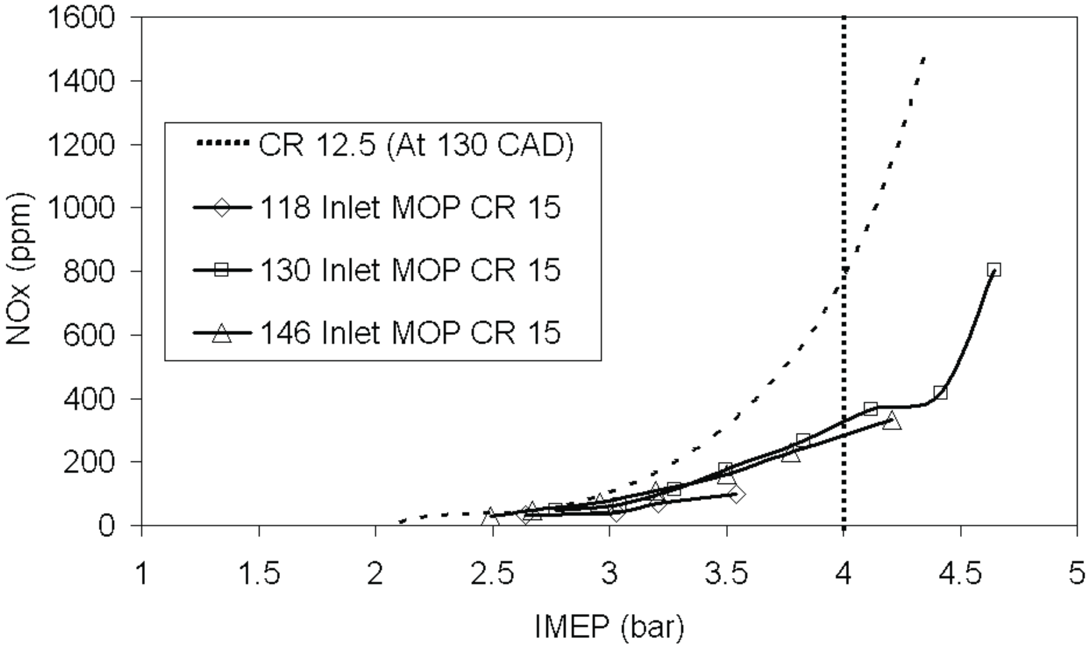


Fig. 7. NOx emissions for propane HCCI [30] 1500 rpm, lambda =1

NOx emissions were characteristically low due to the nature of homogeneous combustion. NOx emissions for CR 15, shown in Figure 7, are lower compared to CR 12.5, especially at higher loads. This is due to the later start of combustion, as a significant part of the load range now has the 5% burn point after TDC. This can be seen also when the inlet MOP is set to 118 CAD. At this valve timing, the combustion phasing is between TDC and 3 CAD after TDC. For the load range of 2.6 to 3.6 bar IMEP, the NOx emissions for the 118 CAD inlet valve timing are lower than for the other two timings. With the higher compression ratio of 15 and without inlet air heating, variation of the inlet valve timing results in changes of the combustion phasing which consequently affect the maximum rates of pressure rise and NOx emissions. A varying inlet valve strategy could be used to take advantage of this for the control of combustion phasing.

3.4 Modelling

Although it has been widely accepted that the single-zone model is adequate and satisfactory in calculating the auto-ignition timing, the work of Babajimopoulos et al [31] shows that the single zone model does not always predict the same ignition timing as the multi-zone model. Also it is known that the burn rate of single-zone model is very fast and the resultant pressure rise is over sharp, compared with experimental data. The situation is different in multi-zone modelling.

The single-zone CHEMKIN model coupled with 1-D gas dynamic simulation code WAVE has been used in the previous work of the authors [21, 25, 32]. Cam profile switching and variable cam phasing as well as fuel reforming are suggested to achieve smooth transitions between SI and HCCI [33]. This combustion model has been further developed into a multi-zone model.

Details of the multi-zone model will be presented in a separate paper [28]; here only a fragment of the simulated results from the single-zone and multi-zone model are presented and compared with experimental data. For detailed investigations, a 9-zone model has been used with initial conditions listed in Table 2. The charge is divided into zones in relation to temperature distribution within the cylinder, with no distinct physical boundary between the zones and the zone numbers are sorted by their values of temperature. Zone number 1, near the wall, is the coldest zone and number 9 is the hottest zone found the first ignition region at the time of combustion occurrence. The mass fractions including EGR distributions are the main factors that can was investigated while the equivalence ratio was maintained at 0.971. The averaged in-cylinder temperature at IVC calculated by the WAVE model of the V6 engine was 573K for the engine condition studied, and two different mass and temperature distributions (Table 2) were investigated. Case 1 was chosen according to [34], on the basis of an assumption that 40% of the gas in the cylinder is within the highest temperature zone. The assumption made in case 2 is based on the consideration of [35] that the hottest zones, as well as the coldest zones, have the lowest mass fractions and the middle temperature zones are associated with the highest mass fractions within the cylinder. It appears that this assumption cannot be far from reality of the present engine, as the observations of [36] and [37] using flame imaging are that auto-ignition in HCCI engines with EGR trapping starts in relatively small regions. The comparison between the predicted in-cylinder pressure data, using the single-zone and multi-zone models, is given in Figure 8. The results show that not only the burn rate predicted by the single-zone model is over fast but also the ignition timing is very late, while multi-zone model seems to be able to give a better prediction of the ignition timing and burn rate.

Table 2. Initial conditions for the 9-zone model simulation for CASE 1 and CASE 2 [28]

Zone	Mf (%)	Residual Mf	T (K)	Zone	Mf (%)	Residual Mf	T (K)
			Case 1				Case 2
1	1	0.432	430	1	3	0.502	500
2	1	0.442	460	2	5	0.502	527
3	1	0.452	490	3	5	0.442	548
4	2	0.462	510	4	10	0.462	558
5	2	0.472	530	5	16	0.482	568
6	6	0.482	550	6	30	0.502	578
7	17	0.492	565	7	16	0.522	588
8	30	0.502	580	8	10	0.542	598
9	40	0.522	590	9	5	0.562	608

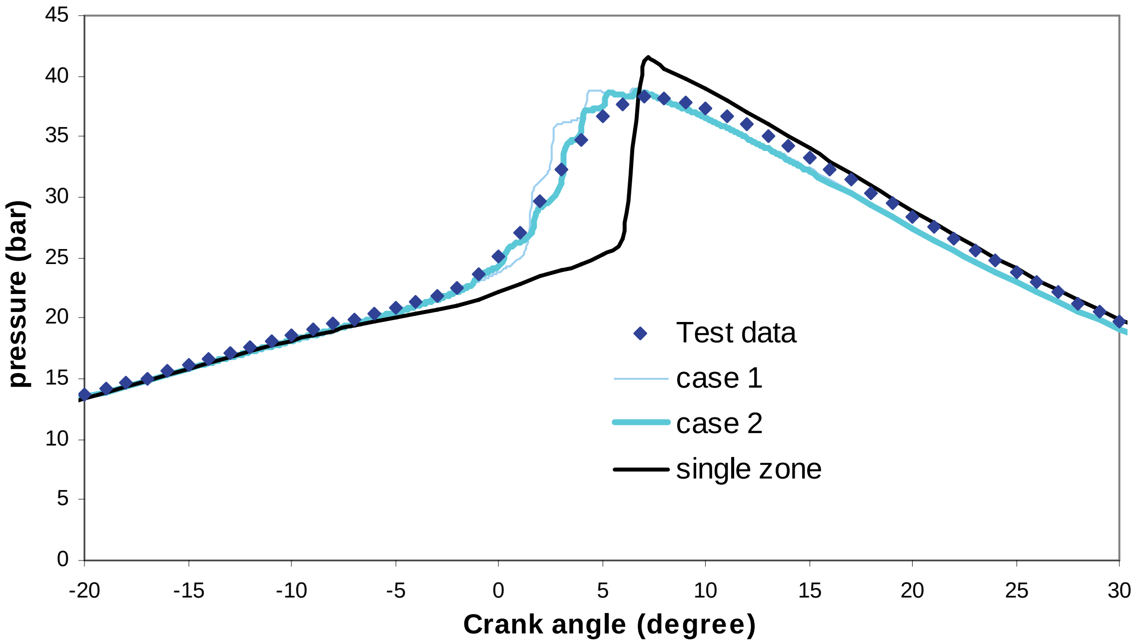


Fig. 8. Comparison of the pressure history calculated by the single-zone, multi-zone models and experimental data [28]

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