Combustion in a heavy-duty direct-injection engine using hydrogen–methane blend fuels

G P McTaggart-Cowan[1]*, **S N Rogak**[2], **S R Munshi**[3], **P G Hill**[2], and **W K Bushe**[2]

[1]Wolfson School of Mechanical and Manufacturing Engineering, Loughborough University, Loughborough, UK
[2]Department of Mechanical Engineering, University of British Columbia, Vancouver, Canada
[3]Westport Power Inc., Vancouver, Canada

Abstract: Adding hydrogen to the fuel in a direct injection natural gas engine offers the potential significantly to reduce local and global air pollutant emissions. This work reports on the effects of fuelling a heavy-duty engine with late-cycle direct injection of blended hydrogen–methane fuels and diesel pilot ignition over a range of engine operating conditions. The effect of hydrogen on the combustion event varies with operating condition, providing insight into the fundamental factors limiting the combustion process. Combustion stability is enhanced at all conditions studied; this leads directly to a significant reduction in emissions of combustion byproducts, including carbon monoxide, particulate matter, and unburned fuel. Carbon dioxide emissions are also significantly reduced by the lower carbon–energy ratio of the fuel. The results suggest that this technique can significantly reduce both local and global pollutant emissions associated with heavy-duty transport applications while requiring minimal changes to the fuelling system.

Keywords: direct injection, hydrogen, natural gas, diesel pilot

1 INTRODUCTION

Reducing the environmental impact of heavy-duty engines is crucial in the search for efficient and sustainable road transportation. Alternative motive power technologies that retain the performance, efficiency, and reliability of existing engines have the best possibility of being accepted by users. To date, direct injection compression-ignition diesel engines have dominated this class; in the near term, these engines will likely remain the motive power source of choice. Replacing the liquid diesel fuel in these engines with a cleaner-burning alternative, such as natural gas, can reduce emissions of both local air pollutants and greenhouse gases (GHGs). The use of natural gas can also provide energy source diversity benefits, while making use of the existing fuel processing and distribution infrastructure. Blending hydrogen into the natural gas offers further advan-tages, including the near-term application of hydrogen to an in-use transportation application with few requirements for new infrastructure. However, the presence of significant concentrations of hydrogen in the fuel can have a marked effect on the combustion process. This work investigates how a heavy-duty natural-gas-fuelled compression-ignition engine's combustion process and pollutant emissions are influenced by adding hydrogen to the fuel.

1.1 Natural gas fuelling of heavy-duty engines

Most in-use natural-gas-fuelled heavy-duty engines use a spark to ignite a premixed natural gas–air charge. Compared with conventional diesels, these natural gas engines typically emit less ambient air pollutants such as oxides of nitrogen (NO_x) and fine particulate matter (PM), at the cost of reduced fuel conversion efficiencies and impaired engine perfor-mance. They also tend to emit more methane (CH_4),

Corresponding author: Wolfson School of Mechanical and Manufacturing Engineering, Loughborough University, Lough-borough LE11 3TU, UK. email: g.mctaggart-cowan@lboro.ac.uk

which is the primary constituent of natural gas and is a significant GHG. The reduced efficiency and higher CH_4 emissions tend to offset the GHG benefits achieved through the lower carbon–energy ratio of the natural gas fuel. A more comprehensive review of natural gas fuelling of heavy-duty engines is available elsewhere [1, 2].

One technology for natural gas fuelling of heavy-duty engines, developed by Westport Innovations Inc., uses natural gas injected directly into a diesel engine's combustion chamber late in the compression stroke. This technology is particularly aimed at heavy-duty transport truck applications. A small amount of diesel fuel is injected prior to the natural gas; the autoignition and combustion of this pilot fuel provides an ignition source for the gaseous fuel. The natural gas then burns as a predominantly non-premixed combustion event. This provides a combustion heat release rate that is very similar to that for pure diesel fuelling, resulting in engine performance and efficiency equivalent to those of the baseline diesel-fuelled engine [3]. CO_2 emissions are reduced by the lower carbon–energy ratio of the natural gas, while the fuel's lower adiabatic flame temperature significantly reduces NO_x formation. PM emissions are also lower, because of the lower sooting tendency of natural gas. Based on these effects, test-cycle emissions have been certified by the California Air Resources Board at levels below the 2007 standards [4]. Further, as the natural gas is not premixed in the combustion chamber, diesel engine compression ratios and efficiency can be retained, while emissions of CH_4 are significantly lower than from other natural gas fuelling technologies [1].

To reduce pollutant emissions sufficiently to meet upcoming standards, engine developers are pursuing various advanced combustion strategies. One of these is the use of exhaust gas recirculation (EGR) as a means of reducing combustion temperatures and hence limiting nitrogen oxide (NO) formation. This technology has been shown to be effective in reducing NO_x emissions from both diesel and direct-injection natural gas combustion systems [5, 6]. However, at the very high EGR levels required to meet future NO_x regulations, combustion stability is impaired and emissions of CO, PM, and unburned fuel increase substantially [7]. One alternative, which could maximize the potential benefits of EGR, is to use a fuel additive to enhance combustion stability; hydrogen is one such additive that has been studied extensively for natural gas-fuelled engines.

1.2 Hydrogen-blended natural gas in engines

The concept of using hydrogen as an additive to improve the combustion in internal combustion engines was first suggested for conventional gasoline fuelling [8, 9]. Several more recent studies have investigated the effects of blending natural gas and hydrogen for use in homogeneous-charge, spark ignition engines [10–16]. The results from these studies vary, with the most important influence being enhanced combustion stability [10, 11, 14]. Fundamental studies demonstrate that hydrogen's enhanced diffusivity results in a higher turbulent flame propagation rate [17]. The hydrogen also increases the flame's resistance to stretch, reducing local extinction events, and generates higher levels of O, OH, and H radicals [18]. Adding 20 per cent hydrogen to the fuel can increase OH radical concentrations by as much as 20 per cent, significantly increasing methane oxidation rates [15, 19].

The effects of hydrogen addition on a premixed combustion event vary with the air–fuel ratio. For a given air–fuel ratio, NO_x emissions are greater with hydrogen addition because of its higher flame temperature (see Appendix 2), while CO and HC emissions are reduced [11, 14]. The influence of hydrogen becomes more significant at higher air–fuel ratios, where unstable and incomplete combustion leads to high emissions and poor performance. By enhancing combustion stability, hydrogen addition allows operation at higher air–fuel ratios. As the combustion temperature decreases with increasing air–fuel ratio, combining hydrogen addition with higher air–fuel ratios can lead to a net reduction in NO_x emissions [12, 14]. Flame stability in the presence of EGR is also improved [13, 15]. These effects depend on the volumetric hydrogen concentrations investigated; values of 15–20 per cent generally achieve substantial improvements without significant negative effects [12, 13, 20]. Above 30 per cent, substantial reductions in the charge energy density, coupled with a higher potential for harmful pre-ignition, pose substantial handicaps while providing little further benefit in emissions or stability [15, 16].

For premixed combustion, as is found in most spark ignition engines, adding hydrogen to the gaseous fuel lowers its volumetric energy density, reducing the maximum power that the engine can deliver. This can be overcome by increasing the intake charge pressure (turbocharging or supercharging) [12, 14] or by fuel injection directly into the combustion chamber. In a spark ignition engine, direct injection of a gaseous fuel during the intake

stroke typically generates a well premixed charge [21]. Delaying the injection until the beginning of the compression stroke results in reduced fuel–air mixing, a slower combustion event, and impaired engine performance [21]. However, this system is still based on premixing of the fuel with the air prior to ignition from a spark plug, a fundamentally different combustion process from a late-cycle direct injection engine with pilot ignition.

The non-premixed combustion of hydrogen–methane blends, as found in a diesel engine operating on late-cycle direct injected natural gas, has not been as extensively studied as the premixed case. Fundamental studies suggest that non-premixed flame stability is enhanced by the higher flame speeds and improved mixing associated with hydrogen addition [22]. In partially premixed combustion, flame thickness increases with hydrogen addition [23]. In industrial gas turbines and boilers, hydrogen addition increases NO formation (owing mainly to high H and OH radical concentrations), and flame stability is improved [24]. Preliminary work on a late-cycle direct injection natural gas-fuelled engine with pilot diesel ignition suggests that NO_x emissions increase, while combustion stability improves and unburned fuel and PM emissions are reduced with hydrogen addition to the fuel [25]. However, these tests were conducted only at a single, low-load, high-EGR operating condition; they are not necessarily applicable to more conventional higher-load operating conditions.

2 EXPERIMENTAL APPARATUS

The research facility used in this project is a Cummins ISX series heavy-duty diesel engine modified for single-cylinder operation and adapted to operate on direct injection of natural gas fuelling. The single-cylinder engine, described in Table 1, has performance and emissions similar to an equivalent natural-gas-fuelled multicylinder ISX engine [26, 27]. The diesel and gaseous fuel injection processes are controlled electronically using a single centrally mounted dual-fuel injector. Combustion air is supplied from an industrial rotary-screw air compressor fitted with water and oil separators. EGR flow is controlled with intake and exhaust pressure controllers and a valve in the EGR loop. This air exchange system allows the charge conditions to be controlled independently from the engine operating mode and fuel composition.

The engine facility is fully instrumented, with measurements of air and fuel flow (both diesel and gaseous fuel) as well as exhaust composition. The intake air flow is measured by a venturi, while EGR rate is determined from the relative concentrations of CO_2 in the intake and exhaust streams. The gaseous fuel flow measurement uses a Coriolis-force mass flow sensor, which provides the gaseous mass flowrate independent of the fuel composition. The fuel composition is determined via gas chromatography. The pilot diesel flow is measured using a gravimetric system. Emissions, including CO, HC, CO_2, and NO_x, are measured undiluted by individual analysers in a dedicated emissions bench. PM emissions are sampled using a microdilution system, where raw exhaust is diluted (at a volume ratio of 15:1) by clean, dry nitrogen. The total PM mass in the diluted stream is evaluated using a tapered-element oscillating microbalance (validated against gravimetric filter measurements [27]). The black carbon (BC) content of the PM is measured using an aethaelometer, where PM is collected on a quartz filter. The attenuation in the intensity of a monochromatic light shone through the sample indicates the exhaust concentration of light-absorbing black carbon. Further information on the operation and validation of the PM sampling instrumentation is available elsewhere [27].

The combustion process is monitored using a water-cooled in-cylinder pressure transducer and a $\frac{1}{2}°$ crank angle (CA) encoder. This information is used to calculate the gross indicated work per cycle (work done during the compression and power strokes only) and the gross indicated power. The latter is used to normalize the fuel consumption and

Table 1 Engine and injector specifications

Engine	Cummins ISX single-cylinder four-stroke, four-valve
Fuelling	Direct injection; diesel pilot, gaseous main fuel
Displacement (per cylinder)	2.5 L
Compression ratio	17:1
Bore/stroke/connecting rod length	137/169/261 mm
Injector	Westport Power Inc. dual-fuel concentric needle
injection control	Separate diesel and natural gas solenoids
injector holes	Seven pilot, nine gas
injection angle	18° below fire deck

the emissions measurements. The heat release rate (HRR) can also be calculated from the pressure trace [28]

$$\frac{\mathrm{d}Q_{\text{net}}}{\mathrm{d}\theta} = \frac{\gamma}{\gamma-1} p \frac{\mathrm{d}V}{\mathrm{d}\theta} + \frac{1}{\gamma-1} V \frac{\mathrm{d}p}{\mathrm{d}\theta}$$

where θ is the crank angle, p is the in-cylinder pressure at a given crank angle, V is the cylinder volume at that point, and γ is the specific heat ratio (c_p/c_v – assumed constant at 1.37). The net heat release rate represents the rate of energy release from the combustion processes less wall heat transfer and crevice flow losses. The definition of the combustion timing and duration are based on the relative fraction of the total energy released between the start of combustion (identified from the HRR) and the timing of interest, presented as the integral of heat release (IHR). Combustion timing is defined as the crank angle location at which half the heat has been released (50%IHR), while the burn duration refers to the total duration of combustion (from 5%IHR to 90%IHR). Subdividing the IHR into 'initiation' (5–10%IHR), 'early' (10–50%IHR), and 'late' (50–90%IHR) provides an indication of the relative durations of the gaseous fuel combustion initiation, premixed combustion, and non-premixed combustion phases.

2.1 Test conditions

The effects of hydrogen addition to the natural gas are evaluated at two baseline operating conditions, as indicated in Table 2. A low-load, low-speed, high-EGR condition is representative of a highly unstable operating condition, where hydrogen is anticipated to have the greatest benefit. The details of these experimental results have been presented previously

[25]. The second condition involves a higher load at a moderate speed with a moderate level of EGR. This condition is generally representative of a steady state cruising mode for a heavy-duty engine. The use of EGR maintains relatively low levels of NO_x without degrading the combustion event at this high-load condition. The results presented here involve fuels with three hydrogen concentrations at each operating condition, as shown in Table 2.

Varying the timing of the combustion provides a range of combustion conditions while maintaining a constant charge composition and overall fuel–oxygen equivalence ratio. This timing is controlled by the start-of-injection timing of the diesel; the gaseous fuel injection starts a fixed 1.0 ms after the end of the diesel injection. In this work, the mid-point of the combustion event varies from top dead centre to 15° CA after top dead centre (ATDC), a typical range for heavy-duty engine applications. Later combustion corresponds to lower NO_x emissions but reduces efficiency and increases combustion instability. To hold the combustion timing (50%IHR) fixed while varying the fuel composition, the start-of-injection timing is shifted. To account for the variations in the gaseous fuel's energy density, the injection duration is adjusted to maintain a constant indicated power output from the engine (from 1.5 to 1.75 ms at high load, and from 1.2 to 1.3 ms at low load). For the tests reported here, the pilot diesel quantity is fixed at 6 mg/injection; this corresponds to approximately 4 per cent of the total fuel energy at the high load case and 7 per cent at the low load case. To achieve this injection mass under the test conditions required commanded durations of 0.65 ms at high load and 0.9 ms at low load. The pilot fuel commanded start-of-injection timing was approximately 30° CA before the 50%IHR point at high load and 25° CA before it at low load; this varied by as much as 5° CA with

Table 2 Operating conditions and fuel composition

Parameter	Low load			High load		
Speed (r/min)	800			1200		
Indicated power (kW) [% load]	10 [25%]			35 [75%]		
Ind. mean effective pressure (bar)	6			13.5		
Fuel–oxygen equivalence ratio	0.5			0.6		
Gaseous fuel pressure (MPa)	16			21		
EGR (mass %)	40			30		
Combustion timings (50%IHR)	0,5,10,15°ATDC			0,5,10,15°ATDC		
Fuel						
%H$_2$ by volume	0	10	23	0	15	35
%H$_2$ by mass	0	1.4	3.7	0	2.2	6.3
%H$_2$ by energy	0	3.6	9.2	0	5.6	15.2
Fuel energy density (MJ/m^3)	32.7	30.4	27.3	32.7	29.2	24.6
Fuel carbon–energy ratio (g C/MJ)	16.7	16.1	15.3	16.7	15.7	14.1
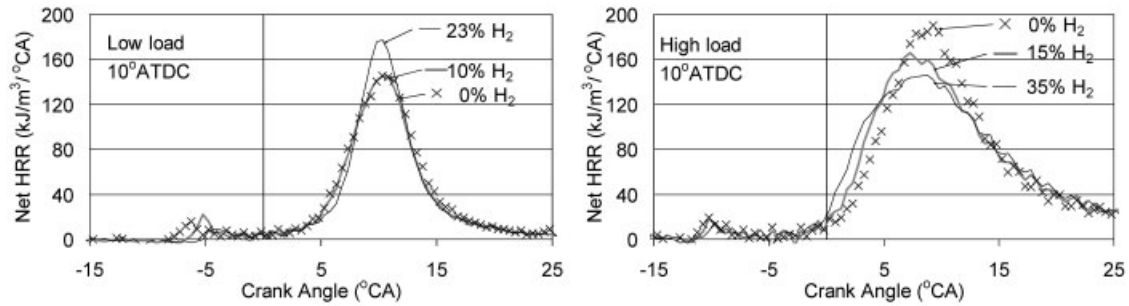
Fig. 1 Effects of hydrogen content on heat release rate. Low load (left) and high load (right) at 50%IHR = 10° CA ATDC timing

composition and combustion phasing owing to variations in the ignition delay of the gaseous fuel.

3   RESULTS

The addition of hydrogen to natural gas fuel has a significant impact on engine performance and emissions. The results presented here evaluate the influence of hydrogen addition, with a particular emphasis on the differences between the low- and high-load cases. The evaluations are conducted on the basis of in-cylinder heat release rate, combustion stability, and the corresponding effects on emissions.

3.1   Effects on heat release rate

The effects on the combustion process of hydrogen addition to the fuel vary substantially with operating condition. Figure 1 shows the calculated heat release rate at both low- and high-load cases at a sample combustion timing (50%IHR at 10° CA ATDC; the other timings show similar effects [27]). At low load, hydrogen addition increases the peak heat release

rate, which is representative of the maximum rate at which fuel is being consumed in the reaction zone. At high load this rate is reduced. These effects are consistent at all combustion timings, as shown in Fig. 2. This figure shows all four timings for each load and fuel condition, with the arrow indicating the effect of timing. In all the plots where the arrow is shown, the trend in results with combustion timing is consistent at both loads and at all fuel compositions. The gaseous ignition delay (GID), also shown in Fig. 2, is reduced with hydrogen addition. The higher concentrations of reactive species such as H and OH [18, 24] will lead to a wider flammability range and a correspondingly shorter time until an ignitable gaseous fuel–air mixture reaches the vicinity of the pilot reaction zone. For both peak heat release rate and GID, combustion timing has significantly larger effects at low load than at high load; however, the magnitudes of the effects of hydrogen content are similar at both operating conditions.

These results suggest that, at low load, chemical reaction rates, rather than physical mixing processes, are limiting the combustion rate. Adding hydrogen to the fuel increases the availability of
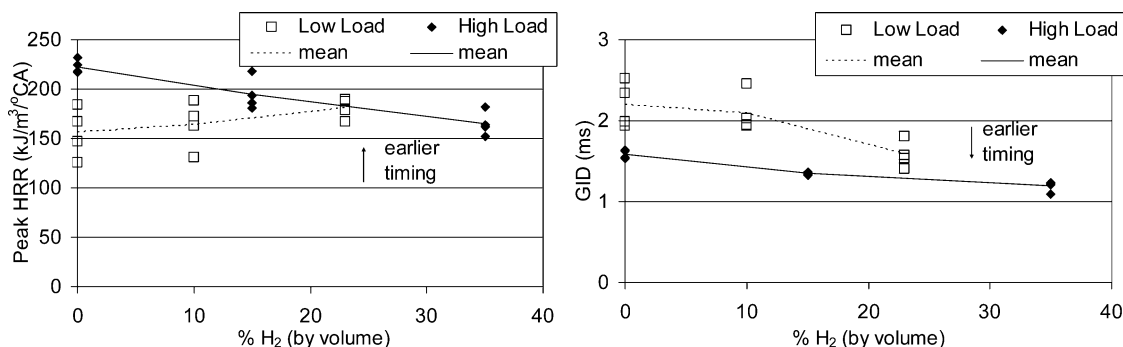
Fig. 2 Effects of hydrogen content on peak heat release rate (left) and gaseous fuel ignition delay (right). All timings shown for each load condition

reactive radicals in the reaction zone, enhancing the net combustion rate. The reduction in combustion rate at high load is attributed to the fuel's lower energy density (Table 2). This suggests that at high load the early stages of the combustion process are restricted by the availability of fuel, indicating an injection-rate-restricted reaction event. The reductions in the GID demonstrate that adding reactive species derived from the hydrogen in the fuel enhances the ignition event at both low and high loads.

Further insight into the effect of hydrogen on the combustion event is provided by the timings of the components of the combustion event. Figure 3 shows the duration of the reaction initiation (5–10%IHR), early (10–50%IHR), late (50–90%IHR), and total (5–90%IHR) gaseous combustion process. The results demonstrate significant variations with timing, especially at low load where adding hydrogen significantly reduces the magnitude of the variations with timing. However, the trends in combustion duration as a function of fuel composition are equivalent for all four timings at both load conditions. The results demonstrate that the earliest stages of the combustion event are enhanced with hydrogen addition at all loads. The effects are relatively small, and variation with timing is significant. However, for a given load and timing, the

hydrogen in the fuel enhances the rate at which the heat is released from the early, post-ignition combustion.

The second phase of the gaseous combustion shown in Fig. 3, from 10% to 50%IHR, includes the partially premixed combustion of gaseous fuel injected prior to ignition. For all the conditions tested, this phase also includes the peak heat release rate. The 10–50%IHR duration is reduced at low load, but is increased at high load, these effects are consistent with the trends in peak heat release rate shown in Figs 1 and 2. At low load, the hydrogen in the fuel is thought to enhance the reaction rates, while the reduced injection rate (on an energy basis) due to the fuel's lower energy density limits the rate of heat release at high load.

The late stage of the combustion event (demonstrated in Fig. 3 by the 50–90%IHR duration) shows significantly less sensitivity to hydrogen addition to the fuel. At high load the duration is slightly reduced, while at low load the combustion duration increases slightly with hydrogen addition to the fuel. These effects, while small, tend to offset the effects observed in the earlier combustion phases. As a result, the net combustion duration (5–90%IHR), also shown in Fig. 3, is in general unaffected by the presence of hydrogen in the fuel at both load conditions and at all the timings investigated.
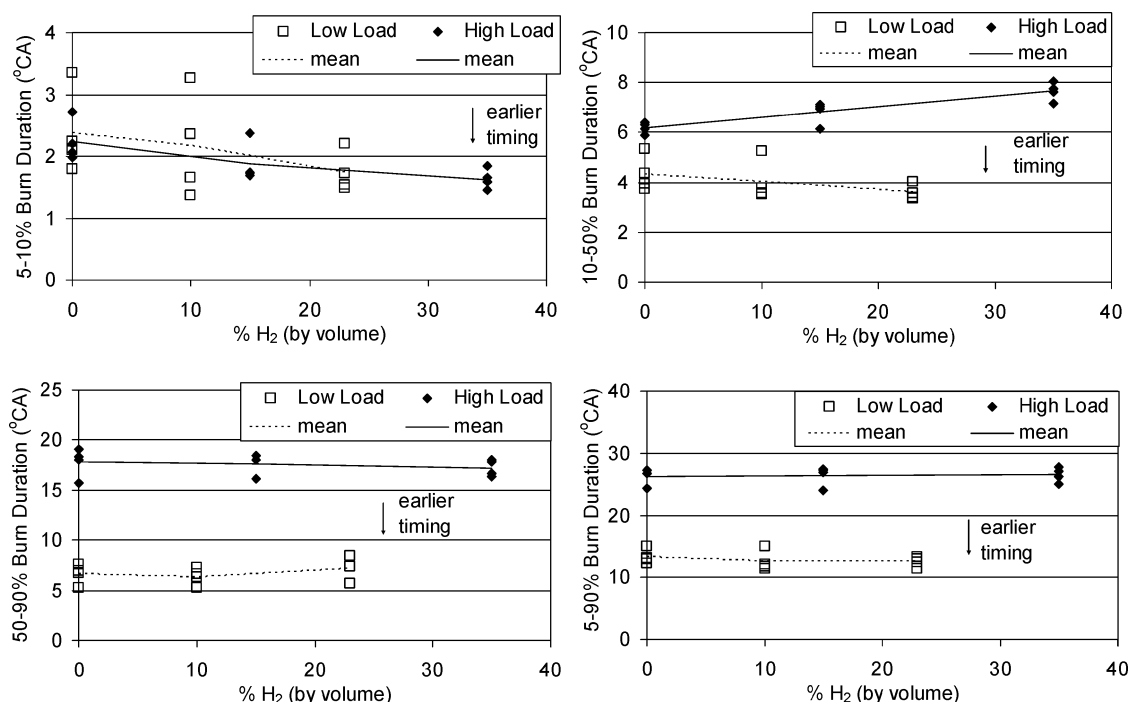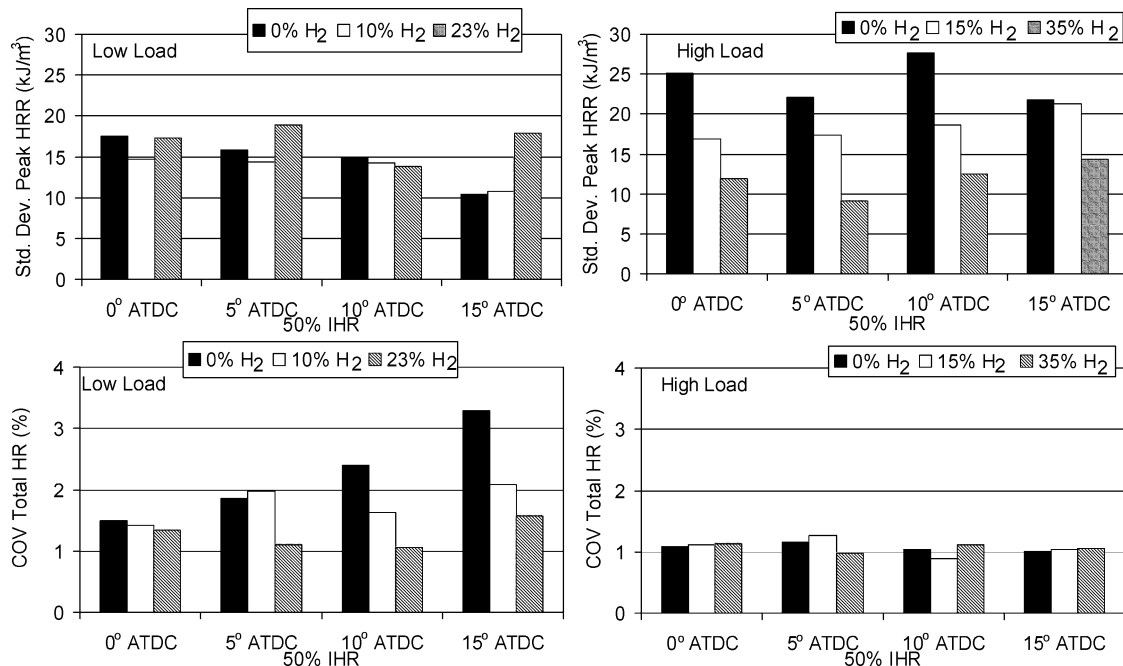

Fig. 3 Effects of hydrogen content on combustion duration. All timings shown for each load condition. Arrows indicate the effect of timing for both load conditions

3.2 Combustion stability

The presence of hydrogen in the fuel has a significant effect on the first half of the combustion event, as demonstrated in Figs 1 to 3. The heat release rate and ignition delay results shown in these figures are based on the average of 45 consecutive cycles. The observed cycle-to-cycle variability, represented by the standard deviation of the peak heat release rate (Fig. 4), provides an indication of the stability of the most intense part of the combustion. This maximum intensity occurs when the partially premixed fuel–air mixture is consumed, and includes the initial stages of the mixing-limited combustion event. While hydrogen addition has no consistent effect on the stability of the peak heat release rate at low load, it substantially reduces this variability at the high-load condition. This is probably a result of greater H and OH radical concentrations and the enhanced diffusivity of the hydrogen, reducing the frequency of local extinctions caused by high turbulent shear stresses in the reacting gaseous jet, as suggested previously [24]. The results indicate that local extinction events may not be a significant source of combustion variability at the low-load operating condition; this coincides with lower in-cylinder turbulence and lower fuel rail pressures, resulting in lower turbulence intensities in the reaction zones.

Hydrogen addition to the fuel also can influence the cycle-to-cycle variability in the net energy release per cycle (calculated from the integral of the heat release rate). This parameter indicates the degree to which the combustion event consumes all the injected fuel; greater variability indicates a larger number of cycles where less of the chemical energy is released during the bulk combustion event. This variability is demonstrated in Fig. 4 by the coefficient of variation (CoV – standard deviation normalized by the mean at each fuel composition and timing condition) of the total heat release (total HR). At low load there is a significant improvement in total energy release stability, indicating that the hydrogen is helping to achieve more complete consumption of the injected fuel. This effect is particularly noticeable at late timings, where the temperatures and pressures late in the combustion process will be dropping more rapidly owing to expansion of the cylinder contents. This drop in temperature will tend to result in bulk quenching of the reactions; the presence of hydrogen, and the associated increase in radical concentrations, is likely delaying this effect. Conversely, there is no significant improvement in the stability of the total HR at high load, nor is there any discernible influence of combustion timing. This suggests that the higher in-cylinder temperatures at high load

Fig. 4 Effects of hydrogen content on combustion stability. Top: cyclical variation (standard deviation) of the peak heat release rate. Bottom: cyclical variation (coefficient of variation) of the total amount of energy released in the combustion event

tend to result in more complete consumption of the fuel. The fact that the addition of hydrogen does not enhance the stability of the bulk combustion process at this condition suggests that bulk quenching is not a major source of combustion variability.

Improvements in combustion stability, either in the peak heat release rate or in the total amount of energy released, are likely to have a significant impact in reducing emissions of partial-combustion byproducts such as CO and HC. These emissions are often associated with increased occurrences of flame extinction, either through turbulent-shear-induced local extinctions or through bulk quenching late in the combustion event.
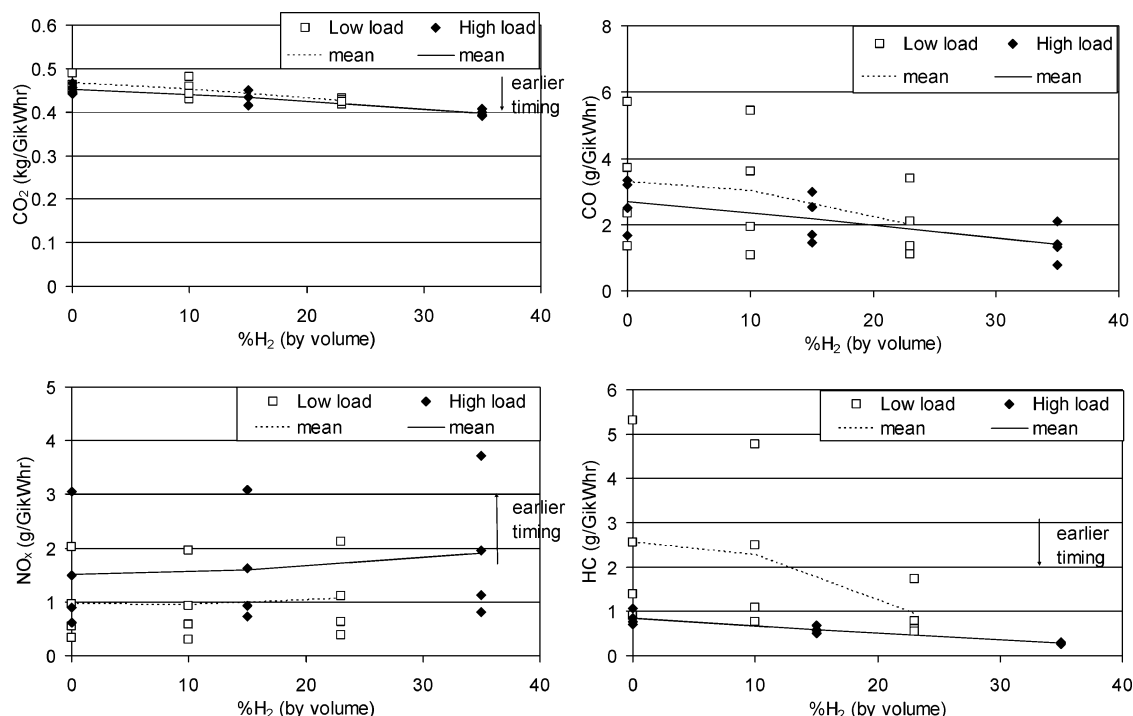
3.3 Emissions

Adding hydrogen to the natural gas has a substantial impact on pollutant emissions; concentrations of gaseous pollutants CO, HC, NO_x, and CO_2 are shown in Fig. 5. Each of these plots shows the effect of hydrogen addition at both load conditions. For each operating condition, all four timing conditions are presented. For those species where combustion timing has a significant impact, the direction of the general trend is shown by the arrow on the

right-hand side of each plot. (In the case of CO, the trend is not consistent with timing; the effects of timing are shown in Fig. 6.)

The principal cause of the reduced CO_2 emissions is the reduction in the carbon content of the fuel; the hydrogen addition does not significantly influence combustion efficiency (not shown; hydrogen addition has no discernible effect on the fuel conversion efficiency at either load condition). The observed CO_2 reduction, at a rate of 3 per cent for every 10 per cent increment in volumetric hydrogen concentration in the fuel, corresponds directly to the reduced carbon content of the fuel (Table 2). The lower carbon content of the fuel will also generate similar reductions in CO and HC emissions. However, other factors may also influence the emissions of these partial-combustion byproducts.

CO emissions, shown in Fig. 5, are reduced to a substantially greater extent than would be explained simply by the lower carbon content of the fuel. The majority of the reduction in CO is most likely due to higher concentrations of radicals such as H and OH in the reaction zone and during the early combustion period [24]. Although CO is highly sensitive to timing (as indicated by the large spread of the data in Fig. 5, especially at the low-load case), the mean

Fig. 5 Effects of hydrogen content on gaseous emissions. All timings shown for both load conditions. Trends with timing, indicated by arrows, are consistent at both load conditions and all H_2 levels for CO_2, HC, and NO_x emissions. CO trends with timing are shown in Fig. 6
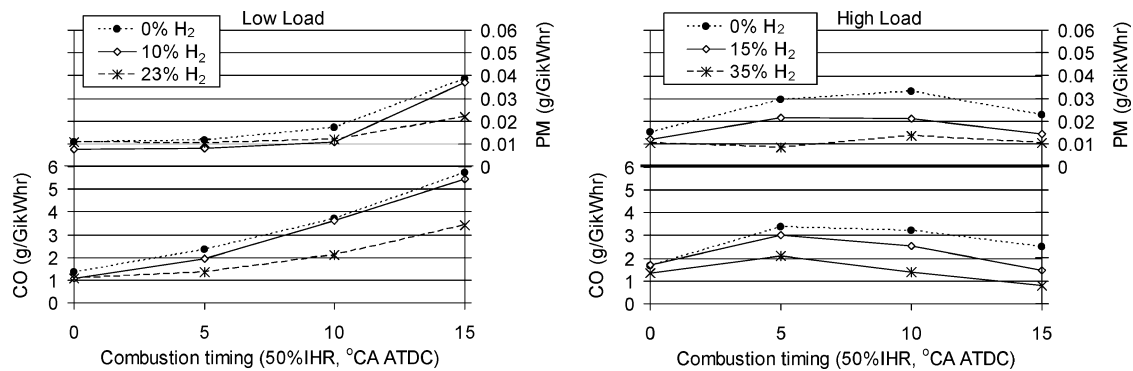
Fig. 6 Effects of timing on CO and PM emissions, at both low (left) and high (right) loads. Trends in BC (not shown) are equivalent to those of PM

reductions with hydrogen addition are similar at both load conditions.

The trends in CO emissions with timing, at both low and high loads, are shown in Fig. 6. At low load, CO emissions increase with later timing; however, at high load, they are greatest at the intermediate timings. While the fundamental causes of the high emissions at intermediate timings are still under investigation, they are thought to be a result of interaction between the burning gases and the piston. This does not influence the final conclusions of this work, as the trends with timing are consistent for all the hydrogen concentrations tested, as shown in Fig. 6. At light load, the trends with hydrogen content are also consistent, although the later timings show a greater reduction in CO with hydrogen addition (relative to no hydrogen) owing to improved combustion stability.

Hydrogen addition to the fuel significantly reduces HC emissions, which are >95 per cent methane (there is no experimentally discernible difference between measured HC and CH_4 levels). As Fig. 5 shows, the reduction in HC emissions at high load, and at the early timings for low load, is essentially linear. However, at the later timings at low load (50%IHR at 10° and 15° CA ATDC), the reduction in emissions is significantly greater. This is a result of the improvements in the combustion stability shown in Fig. 3 for these conditions, which results in reduced unburned fuel emissions. At both loads, later timings consistently resulted in higher HC emissions, with a greater variability with timing for the low-load case. This is a direct result of the greater improvement in overall combustion event stability at these late timing conditions.

NO_x emissions are significantly increased with hydrogen addition to the natural gas fuel. This is in part a direct result of hydrogen's higher flame temperature, as discussed in Appendix 2. However,

these increases in NO_x are substantially greater than would be expected simply on account of the increase in adiabatic flame temperature [7]. NO_x emissions are most likely being further increased by interactions between changes in the in-flame stoichiometry, enhanced diffusion of hydrogen in the reaction zone, and reactive radical concentrations affecting both the reaction zone temperature and the post-flame NO-forming reactions. NO_x emissions are consistently reduced by later combustion timings at both loads and all hydrogen concentrations; the effects of timing on NO_x, while consistent, are particularly strong owing to the large effect of timing on the combustion temperature [6].

Emissions of particulate matter, shown in Fig. 7, are also influenced by reaction zone conditions and increasing concentrations of H_2. At the low-load case, because the baseline PM emissions are very low, the influence of H_2 addition is relatively small; only at the latest timing (highest emissions values) is a significant reduction in PM observed. This effect of timing on PM emissions is shown in Fig. 6. Conversely, PM emissions at high load are consistently reduced by H_2 addition to the fuel. Similarly to the low-load case, the hydrogen has a greater influence at the timing conditions where the PM levels are higher, although at high load these are the intermediate timings. The observed reduction in PM at all timings at high load is primarily due to lower BC emissions, also shown in Fig. 7. The effects of timing on BC are not shown, but are equivalent to the effects of timing on PM. The oxidation of PM precursors (which typically lead to black carbon) in the reaction zone will be enhanced by the higher concentrations of radicals such as H and OH, originating from the hydrogen in the fuel, resulting in less PM formation and hence lower emissions of PM and BC. However, volatile concentrations are still significant; these are typically attributed to
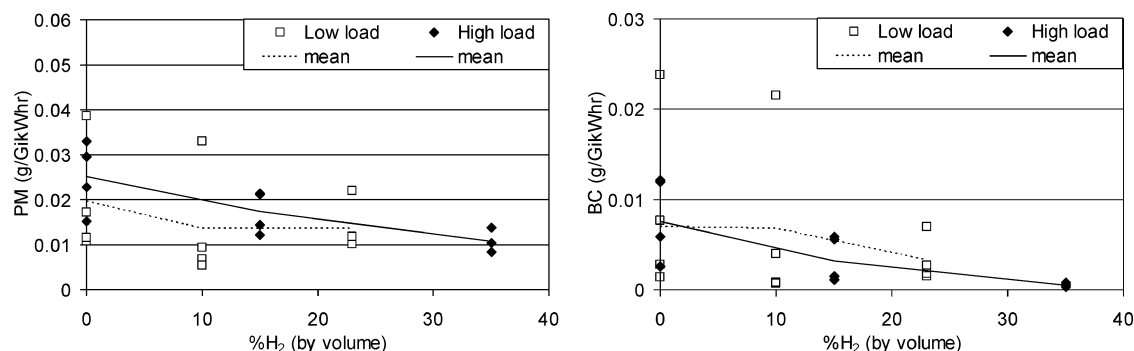
Fig. 7 Effects of hydrogen content on total particulate and black carbon emissions. The trend in PM with timing is shown in Fig. 6; the trend with timing for BC is equivalent to that for PM at both load conditions and at all hydrogen concentrations

evaporated, but unreacted, lubricating oil and diesel pilot fuel which condenses in the exhaust dilution process. The relative contribution of volatiles to the total PM mass increases dramatically with H_2 in the fuel. This indicates that the mechanisms that lead to the virtual elimination of BC emissions with hydrogen in the fuel are not as effective at reducing volatiles.

The reductions in CO_2, CH_4, and BC all contribute to a significant reduction in tailpipe greenhouse gas (GHG) emissions. The total tailpipe GHG reductions can be calculated through the use of 'global warming potentials' for the various emitted species on the basis of their radiative forcing relative to CO_2. Specifically, CH_4 is approximately 23 times more potent a greenhouse gas than CO_2 [29], while black carbon is approximately 500 times more potent [30]. The individual contributions of these three species, averaged over all timings for each load and fuel composition, are shown in Fig. 8. In general, reductions in CO_2 contribute ~70–75 per cent of the total;

CH_4 contributes 15–20 per cent, with the remaining 5–10 per cent from BC. For the low-load, high hydrogen content case, the relative contribution of CH_4 is significantly greater owing to the large reduction in unburned fuel emissions at this condition.

4 DISCUSSION

The addition of hydrogen to the fuel at two different operating conditions has generally similar effects on combustion byproduct emissions, although the effects on the combustion event vary significantly between operating conditions. At low-load conditions, the addition of hydrogen substantially increases the heat release rate, most likely owing to more rapid combustion of the fuel that has premixed prior to ignition of the gaseous fuel. Conversely, at the high-load case, the addition of hydrogen reduces the intensity of the combustion event owing to the lower chemical energy available in the reaction zone immediately after ignition. However, these changes in the average heat release rate do not directly influence the emissions of HC, CO, or PM; these correlate better with improvements in combustion stability at both modes. This indicates that these emissions are more sensitive to combustion variability than to the average combustion progression. These results suggest that the addition of hydrogen to the fuel is improving stability and reducing combustion byproduct emissions, with the greatest effects at lower combustion temperatures.

The varying influences of hydrogen on the average heat release rate suggest that the factors controlling the combustion progression depend on load. The fact that hydrogen addition, which reduces the volumetric energy density of the fuel, reduces the peak heat release rate at high load suggests that this condition is
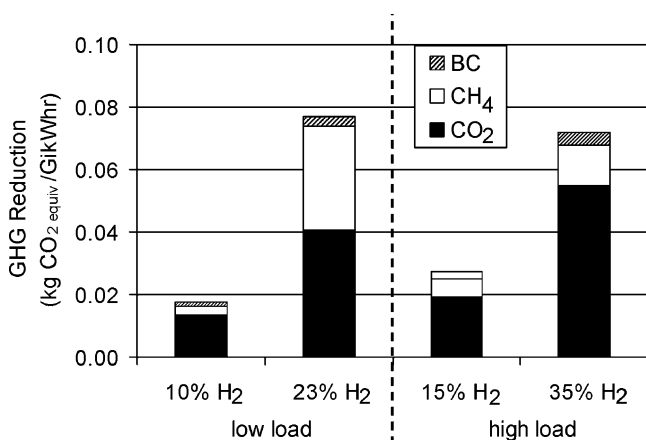
Fig. 8 Tailpipe GHG emission reduction with hydrogen addition to the fuel, averaged for all four timings at each hydrogen concentration

primarily injection rate limited. This suggests that increasing the injection rate, either through increased injection pressures or increased nozzle areas, would tend to increase peak heat release rate, especially with lower energy density fuels. However, such actions could have detrimental effects at low load, where the addition of hydrogen enhances the heat release rate. As a result, adjusting the injection pressure as a function of fuel energy content and load condition to maintain a constant energy injection rate, optimized for a given load, will likely generate the greatest efficiency and pollutant emissions benefits.

Hydrogen-blended natural gas fuelling of a direct injection engine offers significant reductions in tailpipe GHG emissions. The impact on 'well-to-wheel' GHG emissions, however, depends on the emissions associated with the production of the H_2. If the hydrogen is generated using processes that emit significant GHGs (such as electrolysis using fossil-fuel-generated electricity), then the 'well-to-wheel' GHG emissions are substantially greater than for natural gas fuelling. If the hydrogen production is based on energy from nuclear power or from renewable sources such as wind, hydroelectric, or solar, the associated GHG emissions can be reduced to somewhere in the range of 0.5–5 kg CO_2 equiv./kg H_2 produced [31]. Combined with the emissions reductions shown in Fig. 8 and the reduction in natural gas consumption (the production and distribution of which also releases GHGs), this can lead to a reduction in 'well-to-wheel' GHGs of the order of 10 kg CO_2 equiv./kg H_2 consumed. Irrespective of its source, hydrogen addition significantly improves the combustion process and reduces emissions of combustion byproducts.

The application of blended H_2 fuelling for a heavy-duty direct injection vehicle will require some fuel handling system considerations. For engine and emissions control optimization, the capacity to control the H_2 content in the fuel would result in the maximum benefits for the amount of H_2 consumed. However, this would require separate fuelling systems and on-board storage, significantly increasing the fuel system complexity. Blending the H_2 into the natural gas at the fuelling station minimizes fuel system complexity; a sensor to detect H_2 levels in the fuel, and corresponding adaptations to the engine control strategies, would permit engine operation to maximize the potential offered by H_2 addition. This would also allow the engine to operate on varying levels of H_2, depending on the fuel source. The engine control computer could be used to control the combustion process to ensure that the engine was operated at maximum efficiency and with minimum emissions over all H_2 contents, permitting the near-term application of H_2 in heavy-duty transport applications with minimal effects on the engine's fuel system.

5 CONCLUSIONS

1. The ignitability of the fuel mixture is enhanced at all loads by hydrogen addition. This results in shorter ignition delays and in a shorter time for the flame to spread through the early, premixed, stage of the gaseous combustion event.
2. At low load, the hydrogen in the fuel increases the combustion rate during the partially premixed combustion phase. It also reduces the cycle-to-cycle variability in the overall combustion process.
3. At high load, the lower energy density of the hydrogen-blended fuel reduces the peak heat release rate but improves the stability of the combustion event.
4. The overall combustion duration is not significantly affected at either low or high load. This suggests that, while the gaseous fuel ignition event is reaction rate limited and the limitations of the early combustion phases depend on the operating condition, the overall combustion duration is primarily limited by the turbulent mixing rate, which is relatively insensitive to fuel composition.
5. The addition of H_2 reduces CO, HC, and PM emissions. These reductions are more than would result merely from the lower carbon content of the fuel, the magnitude of which is demonstrated by the reduction in CO_2 emissions. The principal cause of the reductions in CO, HC, and PM is most likely enhanced oxidation occurring because of improved combustion stability and higher concentrations of reactive radicals.
6. The NO_x emissions are increased by H_2 addition at both low and high loads. This is in part due to hydrogen's higher flame temperature, but may also be a result of the changes in the fuel–air stoichiometry and direct influences of the H_2 on the NO formation chemistry.
7. The addition of H_2 reduces engine-out GHG emissions. Using a renewably generated hydrogen as an additive in heavy-duty direct injection engines fuelled with natural gas could reduce both local and global air pollutant emissions with only minor modifications to the engine system.

ACKNOWLEDGEMENTS

REFERENCES

1 McTaggart-Cowan, G. P., Reynolds, C. C. O., and **Bushe, W. K.** Natural gas fuelling for heavy-duty on-road use: current trends and future direction. *Int. J. Environ. Stud.*, 2006, **63**(4), 421–440.

2 Cho, H. M. and **He, B. Q.** Spark ignition natural gas engines – a review. *Energy Conversion and Managmt*, 2007, **48**, 608–618.

3 Harrington, J., Munshi, S., Nedelcu, C., Ouellette, P., Thompson, J., and **Whitfield, S.** Direct injection of natural gas in a heavy-duty diesel engine. SAE technical paper 2002-01-1630, 2002.

4 California Air Resources Board, Executive Order A-343-0004r1. Available from: www.arb.ca.gov/msprog/onroad/cert/mdehdehdv/2008/westport_hhdd_a3430004r1_14d9_1d2-0d01_cngplusdiesel.pdf. Effective 30.5.2008. Accessed 18.6.2008.

5 Ladommatos, N., Abdelhalim, S., and **Zhao, H.** The effects of exhaust gas recirculation on diesel combustion and emissions. *Int. J. Engine Res.*, 2000, **1**(1), 107–126.

6 McTaggart-Cowan, G. P., Bushe, W. K., Munshi, S., and **Hill, P. G.** NO_x reduction from a heavy-duty diesel engine with direct injection of natural gas and cooled exhaust gas recirculation. *Int. J. Engine Res.*, 2004, **5**(2), 175–192.

7 Hill, P. G. and **McTaggart-Cowan, G. P.** NO_x formation in a diesel engine fueled by natural gas. SAE technical paper 2005-01-1727, 2005.

8 Sorensen, H. The Boston reformed fuel car – a low pollution gasoline fuel system for internal combustion engines. SAE technical paper 729218, 1972.

9 Houseman, J. and **Hoehn, F. W.** A two-charge engine concept: hydrogen enrichment. SAE technical paper 741169, 1974.

10 Swain, M. R., Yusuf, M. J., Dulger, Z., and **Swain, M. N.** The effects of hydrogen addition on natural gas engine operation. SAE technical paper 932775, 1993.

11 Sierens, R. and **Rosseel, E.** Variable composition hydrogen/natural gas mixtures for increased engine efficiency and decreased emissions. *Trans. ASME, J. Engng for Gas Turbines and Power*, 2000, **122**, 135–140.

12 Munshi, S. R., Nedelcu, C., Harris, J., *et al.* Hydrogen blended natural gas operation of a heavy duty turbocharged lean burn spark ignition engine. SAE technical paper 2004-01-2956, 2004.

13 Allenby, S., Chang, W.-C., Megaritis, A., and **Wyszynski, M. L.** Hydrogen enrichment: a way to maintain combustion stability in a natural gas fuelled engine with exhaust gas recirculation, the potential of fuel reforming. *Proc. Instn Mech. Engrs, Part D: J. Automobile Engineering*, 2001, **215**, 405–418.

14 Collier, K., Mulligan, N., Shin, D., and **Brandon, S.** Emission results from the new development of a dedicated hydrogen-enriched natural gas heavy duty engine. SAE technical paper 2005-01-0235, 2005.

15 Larsen, J. F. and **Wallace, J. S.** Comparison of emissions and efficiency of a turbocharged lean-burn natural gas and hythane-fueled engine. *Trans. ASME, J. Engng for Gas Turbines and Power*, 1997, **119**, 218–226.

16 Bauer, C. G. and **Forest, T. W.** Effect of hydrogen addition on the performance of methane-fueled vehicles. Part I: effect on S.I. engine performance. *Int. J. Hydrogen Energy*, 2001, **26**, 55–70.

17 Kido, H., Nakahara, M., Hashimot, J., and **Barat, D.** Turbulent burning velocities of two-component fuel mixtures of methane, propane, and hydrogen. *JSME Int. J. Ser. B*, 2002, **45**, 355–362.

18 Gaudicheau, J. L., Denet, B., and **Searby, G.** A numerical study of lean CH_4/H_2/air premixed flames at high pressure. *Combust. Sci. and Technol.*, 1998, **137**, 81–99.

19 Schefer, R. W. Hydrogen enrichment for improved lean flame stability. *Int. J. Hydrogen Energy*, 2003, **28**, 1131–1141.

20 Akansu, S. O., Dulger, Z., Kahraman, N., and **Veziroglu, T.** Internal combustion engines fueled by natural gas–hydrogen mixtures. *Int. J. Hydrogen Energy*, 2004, **29**, 1527–1539.

21 Huang, Z., Wang, J., Liu, B., Zeng, K., Yu, J., and **Jiang, D.** Combustion characteristics of a direct-injection engine fueled with natural gas–hydrogen blends under various injection timings. *Energy and Fuels*, 2006, **20**, 1498–1504.

22 Karbasi, M. and **Wierzba, I.** The effects of hydrogen addition on the stability limits of methane jet diffusion flames. *Int. J. Hydrogen Energy*, 1998, **23**(2), 123–129.

23 Naha, S. and **Aggarwal, S. K.** Fuel effects on NO_x emissions in partially premixed flames. *Combust. and Flame*, 2004, **139**, 90–105.

24 Rortveit, G. J., Zepter, K., Skreiberg, O., Fossum, M., and **Hustad, J. E.** A comparison of low-NO_x burners for combustion of methane and hydrogen mixtures. *Proc. Combust. Inst.*, 2002, **29**, 1123–1129.

25 McTaggart-Cowan, G. P., Jones, H. L., Rogak, S. N., Bushe, W. K., Hill, P. G., and **Munshi, S. R.** Direct-injected hydrogen–methane mixtures in a heavy-duty compression ignition engine. SAE technical paper 2006-01-0653, 2006.

26 McTaggart-Cowan, G. P., Bushe, W. K., Hill, P. G., and **Munshi, S. R.** A supercharged heavy-duty diesel single-cylinder research engine for high-pressure direct injection of natural gas. *Int. J. Engine Res.*, 2004, **4**(4), 315–330.

27 **McTaggart-Cowan, G.** *Pollutant formation in a gaseous-fuelled, direct injection engine*, PhD Thesis, University of British Columbia, 2006.

28 **Heywood, J. B.** *Internal combustion engine fundamentals*, 1988 (McGraw-Hill, New York).

29 Intergovernmental Panel on Climate Change. Climate Change 2001: The Scientific Basis. IPCC Third Assessment Report. 2001. p. 388. Available from: http://www.grida.no/climate/ipcc_tar/. Accessed 18.5.2007.

30 **Hansen, J., Sato, M., Kharecha, P., Russel, G., Lea, D. W.,** and **Siddall, M.** Climate change and trace gases. *Philosophical Trans. R. Soc., Part A*, 2007, **365**, 1925–1954.

31 **McTaggart-Cowan, G. P., Rogak, S. N., Hill, P. G., Munshi, S. R.,** and **Bushe, W. K.** Hydrogen–methane blend fuelling of a heavy-duty, direct-injection engine. In Proceedings of ASME International Mechanical Engineering Congress. ASME technical paper IMECE2007-41373, 2007.

APPENDIX 1

Notation

|	
ATDC	after top dead centre
BC	black carbon
CH_4	methane
CoV	coefficient of variation
CO	carbon monoxide
CO_2	carbon dioxide
EGR	exhaust gas recirculation
GHGs	greenhouse gases
GID	gaseous fuel ignition delay time
H_2	molecular hydrogen
HC	hydrocarbons
HRR	heat release rate
IHR	integrated heat release rate
NO_x	oxides of nitrogen (NO, NO_2)
PM	fine particulate matter
°CA	crank angle degree
50%IHR	mid-point of integrated heat release rate

APPENDIX 2

Effect of hydrogen addition on flame temperature

Hydrogen addition to a natural gas fuel significantly influences the combustion event, through both a direct effect on the chemical kinetics and flame temperature and indirectly (for non-premixed combustion) through differential diffusion. Modelling a non-premixed, transient, turbulent, methane–hydrogen combustion process is a significant challenge beyond the scope of this work. However, a simple thermodynamic calculation of the adiabatic flame temperature can provide an indication of the temperature in the reaction zone [28]. This calculation assumes that the reactions proceed to equilibrium instantaneously at constant pressure and that there is no heat loss during the reaction event. Using the bulk in-cylinder pressure (measured) and temperature (estimated) at the start of combustion can provide a useful indication of the variation in the reaction zone temperature; it has previously provided a useful correlation with NO_x emissions [6].

Although the overall fuel–air stoichiometry is known, the local equivalence ratio in the reaction zone will vary spatially and temporally. The local equivalence ratio has a much greater effect on flame temperature than does hydrogen content, as shown in Fig. 9. However, at all equivalence ratios, there is a small but significant increase in flame temperature, especially for richer regions of the reaction zone. This provides a qualitative indication that at least some increase in NO_x emissions may be expected owing to the higher flame temperature.
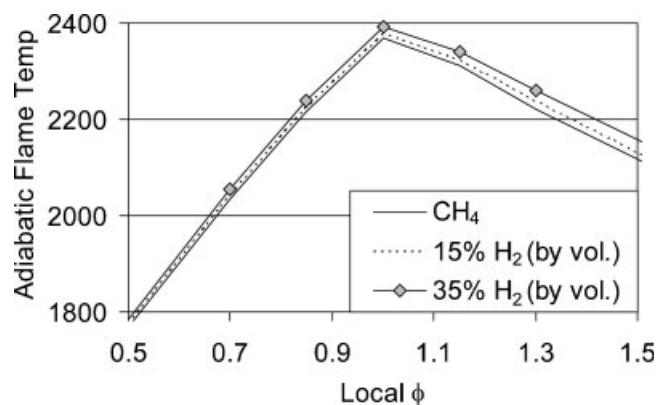
Fig. 9 Effect of hydrogen content on adiabatic flame temperature over a range of equivalence ratios