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A HYDROGEN FUELLED INTERNAL COMBUSTION ENGINE DESIGNED FOR SINGLE SPEED/POWER OPERATION

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Abstract—Sandia National Laboratory is developing from first principles a hydrogen fuelled internal combustion engine for driving an electrical generator that can be utilized either as a stationary power set or the auxiliary power unit in a hybrid vehicle. The intent is to take advantage of hydrogen's unique fuel characteristics and the constant speed characteristics of generator sets to maximize thermal efficiency while minimizing emissions.

The current experiments utilize a flat cylinder combustion chamber shape with two ignition points at high (14:1) compression ratio. Emissions and indicated thermal efficiency measurements with fuels of hydrogen, natural gas and a blend confirm low emissions and high thermal efficiency. CFD modelling done by Los Alamos National Laboratory (Los Alamos, NM) using their KIVA code is helping to further direct variations in the experimental design space. © 1998 International Association for Hydrogen Energy

INTRODUCTION

Sandia National Laboratory, in collaboration with Los Alamos and Lawrence Livermore National Laboratories, is currently developing a hydrogen fuelled engine for a generator set or hybrid vehicle application. The design approach is to utilize first principles to take advantage of hydrogen's unique characteristics of high flame speed, ability to spark ignite homogeneous mixtures at low equivalence ratios and high effective octane to achieve high thermal efficiency while satisfying low emissions requirements without exhaust gas after treatment. The goal is to comply with the California Air Resources Board (CARB) proposed standard for Equivalent Zero Emission Vehicle (EZEV) limits for NO_x [1], the principal emission from hydrogen fuelled engines.

Such an approach, when combined with the energy storage aspects of a hybrid powertrain, will allow operation in the constant power, on/off regime. In such an application, it is anticipated that both the emissions and thermal efficiency will be competitive with fuel cell powertrains, thus offering a low cost interim solution to this transportation need. In addition, the internal combustion engine approach may allow operation on a blend of natural gas/hydrogen, again an attractive option for the transition to a hydrogen infrastructure.

The current experiments at Sandia are being performed in a modified for spark ignition Onan 0.491 litre single cylinder Diesel engine. While this engine has been modified for operation on pure hydrogen fuel, the experiment is carefully instrumented and precise comparison between 100% hydrogen, 30% hydrogen/natural gas by volume and 100% natural gas has been made regarding indicated efficiency, as calculated by integration of the pressure–volume relationships, and NO_x, the principal emission from hydrogen fuelled engines.

ONAN ENGINE

The Onan Engine employed is a modification of the Diesel engine used to power the cooling system of many refrigerated truck trailers. The existing iron cylinder head was discarded and replaced with a custom machined aluminum unit designed and built by Lawrence Livermore National Laboratory (Livermore, CA) with provision for pressure measurement and two point ignition. Both the piston top and the cylinder head are flat, resulting in a right cylinder combustion chamber shape at top dead centre. The spark plugs are located on a common diameter line equidistant from the wall and the cylinder centre. The inlet runner is configured for low swirl.

The compression ratio of the Onan engine was determined by measuring the volume of the cylinder head when bolted to a flat plate using a precision gas volume measurement system at Sandia. In addition, the leakage of the engine through valves and past the piston was carefully quantified and found to be representative of a well functioning modern engine. No exhaust gas recirculation was employed.

Relevant engine characteristics are:

Bore	82.55 mm
Stroke	92.08 mm
Displacement	0.4928 l
Compression ratio	14.04 : 1
Valve timing	Stock
Spark plugs	Champion 53R
Ignition system	Mallory HyFire 667 CDI (2 systems)
	Mallory ProMaster 28880 coils
Inlet system	Pressurized, unthrottled

Modelling of the in cylinder flow was performed by Los Alamos using the KIVA three dimensional CFD code [2]. The calculations suggested that a tumbling motion was developing in the cylinder during induction of fresh charge and that the chamber was not quiescent.

In an attempt to modify this predicted tumbling motion, shrouded valves of various degree were tested in the engine as well as characterized on a steady flow swirl test rig. This flow test, performed by Mike Swain at the University of Miami, utilizes an on axis spinner in a transparent cylinder. The speed was measured stroboscopically. Figure 1 shows the results of these tests for three different shroud configurations (1.6, 3.2, 6.3 mm high, 180° coverage) and an unshrouded valve. Included for reference, are the results from a helically-ported, 2.0 L Chevrolet head. Throughout these swirl tests, a constant pressure drop of 71 mm H_2O was maintained across the valves.

When tested in the engine, the highest swirl valve and the unshrouded valve gave comparable efficiency, while the two intermediate swirl level valves gave 4% higher indicated thermal efficiency. Since the bulk of our work has been conducted with the 1.6 mm shroud, the results presented here are for this valve configuration.

Experimental setup

The intent of the experiments with hydrogen was to operate with a homogeneous charge, and thus great care is taken to assure this condition. The hydrogen and air to the engine were both supplied at 800 kPa. The hydrogen is controlled by a MKS model 1559A mass flow controller specifically calibrated by MKS for hydrogen. Air and hydrogen were then mixed in a stagnation chamber, passed through a sonic orifice and conveyed through the inlet manifold to the engine. Stagnation chamber pressure was measured with a Heise model 901 pressure transducer (1400 kPa full scale). The orifice was machined with a 2 : 1 elliptical contour on the upstream side and the diameter was measured to within 0.0025 mm. The mass flow rate of the mixed gases was calculated using standard sonic orifice relationships and a calibrated discharge coefficient (0.98). The air flow rate was obtained by subtracting off the hydrogen flow.

This inlet system has several attractive features. First, the pressure upstream of the orifice is steady, resulting in continuous mixing of the fuel and air devoid of the flow fluctuations characteristic of single cylinder engines. Second, the passage of the gases through a critical orifice develops a violent mixing environment, ensuring a homogeneous charge. Third, the mass flow rate into the engine was precisely controlled and thus, the engine can be supercharged or run at sub atmospheric pressure by adjusting the upstream air pressure.

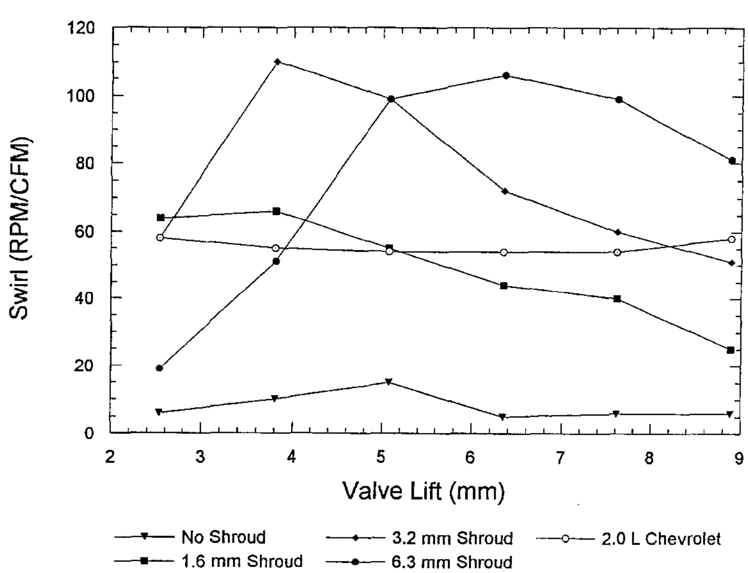

Fig. 1. Effect of valve lift on swirl.

Pressure measurement

The engine is equipped with an AVL model QC42D-X quartz pressure transducer for in cylinder pressure measurement. The charge output of the transducer is converted to voltage by a Kistler type 5010A dual mode amplifier and fed to a Data Translation DT2821-F-16SE 12 bit analog to digital translation board. A Pentium-based 90 MHz PC controls the system and records crankshaft position and inlet manifold pressure at two locations from Teledyne-Taber model 254 strain gage transducers, each amplified by Daytronic model 3270 signal conditioners. The AVL quartz transducer is set by the PC to a defined value at 150° into the cycle (end of inlet stroke) during each cycle as a reference point.

NO_x measurements

The exhaust gases are sampled 8 cm from the cylinder head. The sample is transferred through a heated sample line to a Rosemont Analytical Inc. model 951 A Chemiluminescent NO/NO_x Analyzer. Calibration of this instrument with both zero emissions and 12 ppm NO calibration gases was performed both before and after each test series. This analyzer has a minimum full scale readout of 10 ppm with a sensitivity of 0.1 ppm on this scale, and was found to be stable within 1%.

Fuels and air

The hydrogen used was at least 99.9% pure, supplied from pressurized cylinders. The natural gas was purchased as Tennessee natural gas, consisting of 93.6% methane, 3.6% ethane, 1% propane, 0.7% carbon dioxide, 0.5% nitrogen and 0.4% butane. The 30% hydrogen blend was made up containing 67.5% methane, 2.5% ethane and 30% hydrogen. The air was supplied from an 11 kW air compressor at 800 kPa, run through a desiccator and filtered.

Test procedure

The engine was operated at a comfortable operating point until water temperatures had stabilized at 45–50°C (typically 45 min). During that time the NO_x analyzer was calibrated with both zero emission gas and span gas. Confirmation of spark plug operation was checked by operating each plug separately and noting the torque drop off in each case. After the initial warm up the test conditions were set and the engine was allowed to reach steady state before any data was taken. Pressure data was recorded for 100 cycles and stored as the average at predetermined crank angle degree sample points.

EXPERIMENTAL RESULTS

Tests were conducted with all three fuels (hydrogen, 30% hydrogen/70% natural gas by volume, and natural gas) at 1800 rpm for equivalence ratios which would produce NO_x levels in the CARB proposed EZEV standard range. Since the proposed standards are specified as emissions in grams per mile, the vehicle efficiency affects the results. Equivalence ratio (ϕ), defined as the ratio of the actual fuel/air ratio to the stoichiometric ratio, is also a factor when emissions are measured in parts per million (ppm). For our case we have assumed a 60 miles per gallon (mpg) gasoline equivalent performance level ($\sim$25 km/l), which we believe is reasonable for a hybrid vehicle [3]. Figure 2 portrays the NO_x level in ppm that must not be exceeded as a function of equivalence ratio to meet the proposed EZEV standard.

Additional tests were conducted to determine the minimum ϕ at which the engine would operate. This turned out to be $\phi = 0.2$ for hydrogen, $\phi = 0.48$ for the 30% hydrogen fuel and $\phi = 0.62$ for natural gas. These tests were conducted at volumetric efficiencies of approximately 100%, with indicated efficiency calculated through 2 revolutions (a full 4 stroke cycle) of the crankshaft. The average pressure from 100 cycles is integrated as a function of volume to determine the work:

$$\text{Work} = \int P\,dV.$$

The indicated efficiency was calculated using the lower heating value of the appropriate fuel. In addition, a test series was conducted with hydrogen fuel to determine the effect of various volumetric efficiencies, principally to determine if the engine indicated mean effective pressure (IMEP) can be increased while maintaining the required NO_x levels. These specific results are presented with indicated efficiency calculated from the compression and expansion strokes only (one revolution of the crankshaft), thus removing from the calculation the work produced or absorbed by the inlet process.

Figure 3 presents NO_x and indicated efficiency results as a function of spark advance before top dead centre (TDC) for hydrogen fuel at three equivalence ratios. The EZEV standard is included. It can be seen that operation above $\phi = 0.4$ is not possible in compliance with this NO_x limit. A similar presentation of results for natural gas is contained in Fig. 4. From this plot it can be seen that the EZEV levels can not be met with this fuel, for operation at $\phi = 0.62$ was erratic. When ϕ was increased to 0.64 the engine running is smoothed and torque improved markedly.

The same type of plot for the 30% hydrogen fuel is shown in Fig. 5. Operation at less than 0.52 incurs a large efficiency penalty but at $\phi = 0.52$ operation seems reasonable. From these series of plots it can be inferred that operation with hydrogen at $\phi = 0.40$ and with 30% hydrogen at $\phi = 0.52$ give similar NO_x results. These two conditions are plotted together in Fig. 6.

The next plot of this series, Fig. 7 displays hydrogen fuelled operation at a range of IMEP's (obtained by varying the volumetric efficiency). The equivalence ratio is fixed at $\phi = 0.38$ and each point is taken at the spark advance for maximum torque. Also included on this plot are two points from the mixed gas series.

To investigate and compare more closely the two operating conditions of $\phi = 0.4$ for hydrogen and

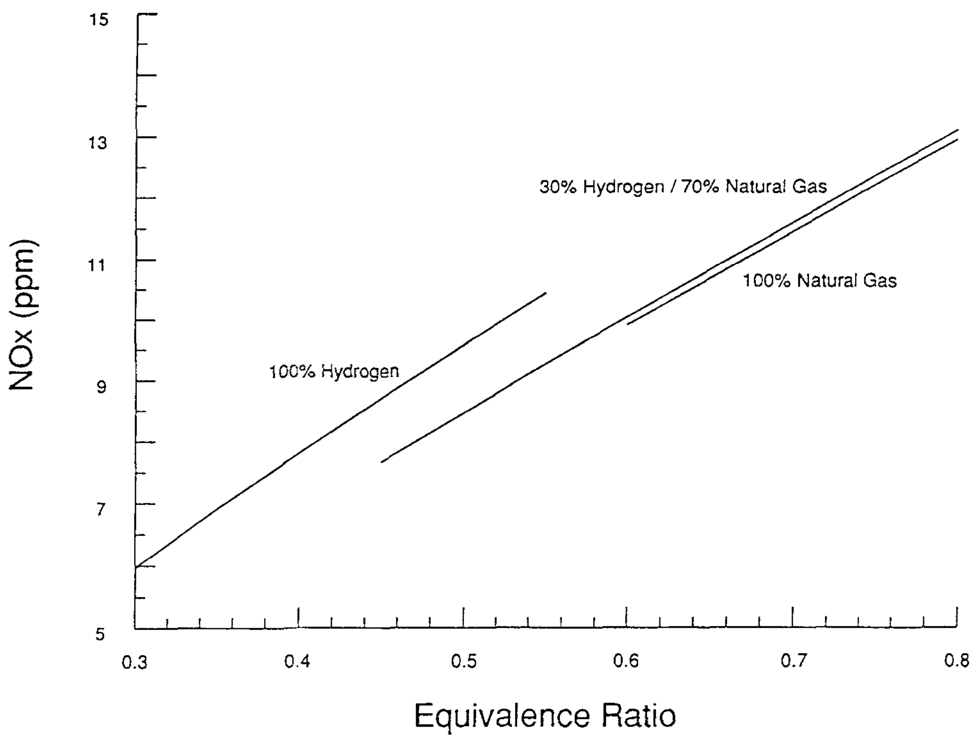

Fig. 2. NO_x limit vs equivalence ratio to meet proposed CARB EZEV Standard for vehicle attaining 60 mpg.

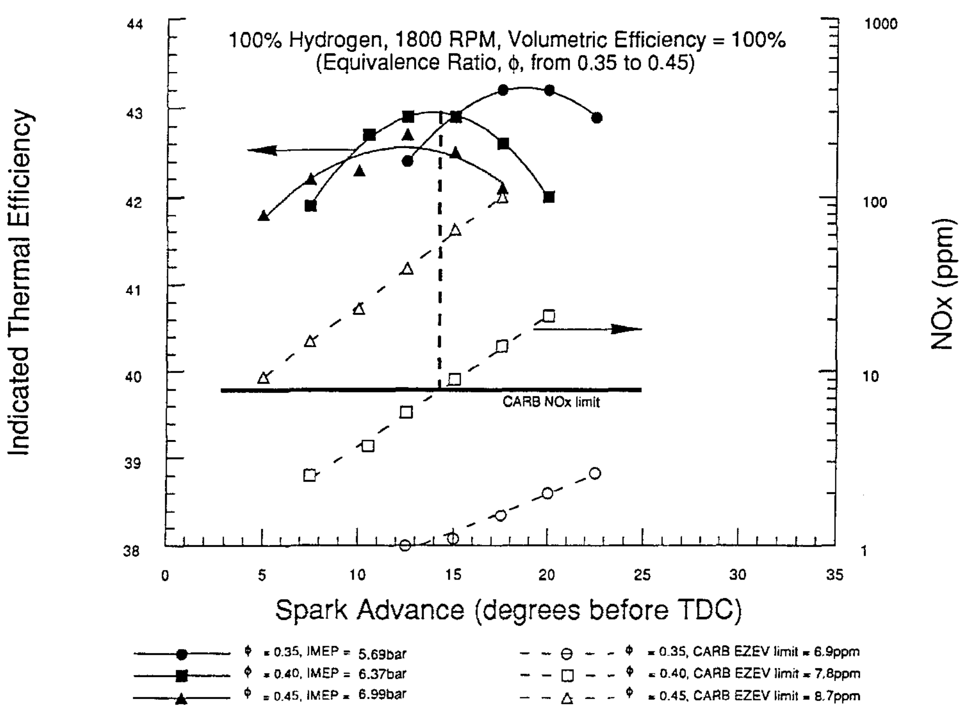

Fig. 3. Indicated thermal efficiency and NO_x vs spark advance. 100% Hydrogen, 1800 rpm, volumetric efficiency = 100%.

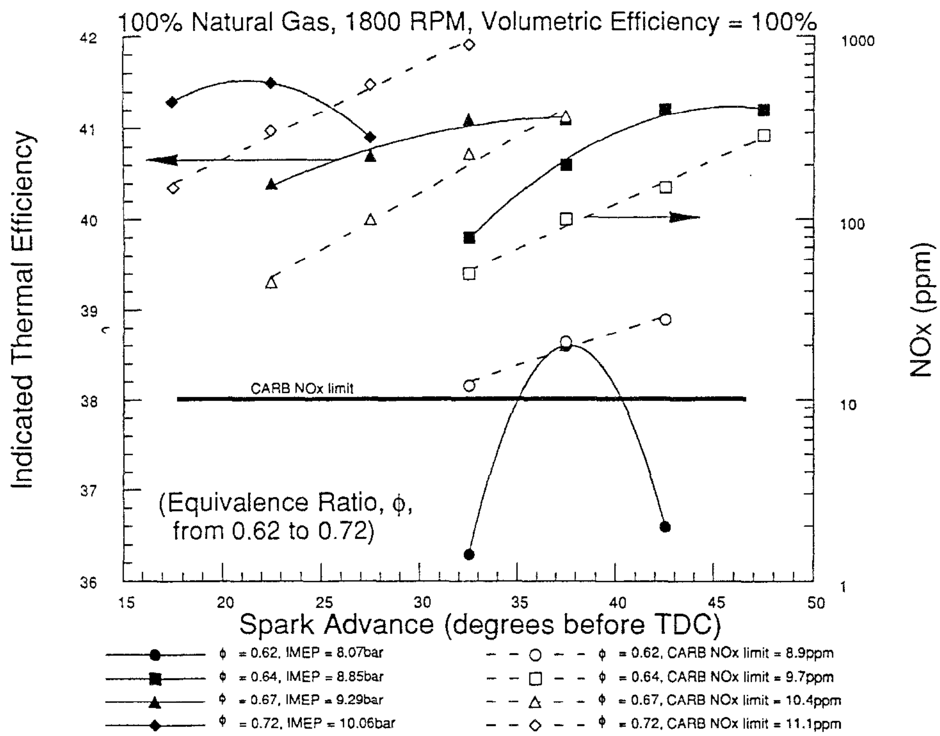

Fig. 4. Indicated thermal efficiency and NO_x vs spark advance. 100% Natural gas, 1800 rpm, volumetric efficiency = 100%.

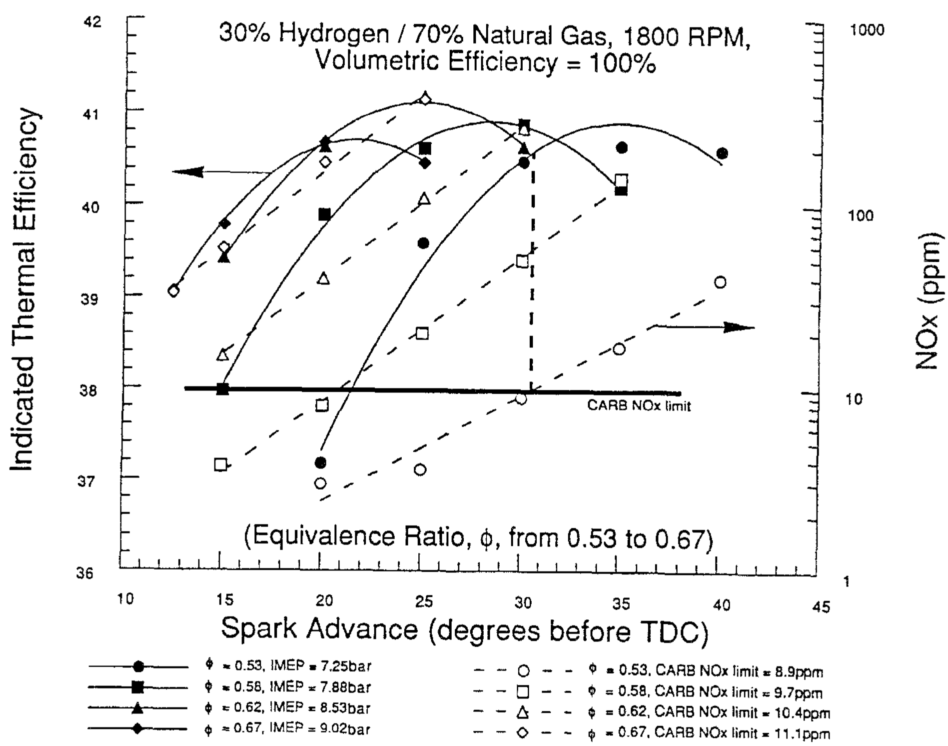

Fig. 5. Indicated thermal efficiency and NO_x vs spark advance. 30% Hydrogen/70% natural gas, 1800 rpm, volumetric efficiency = 100%.

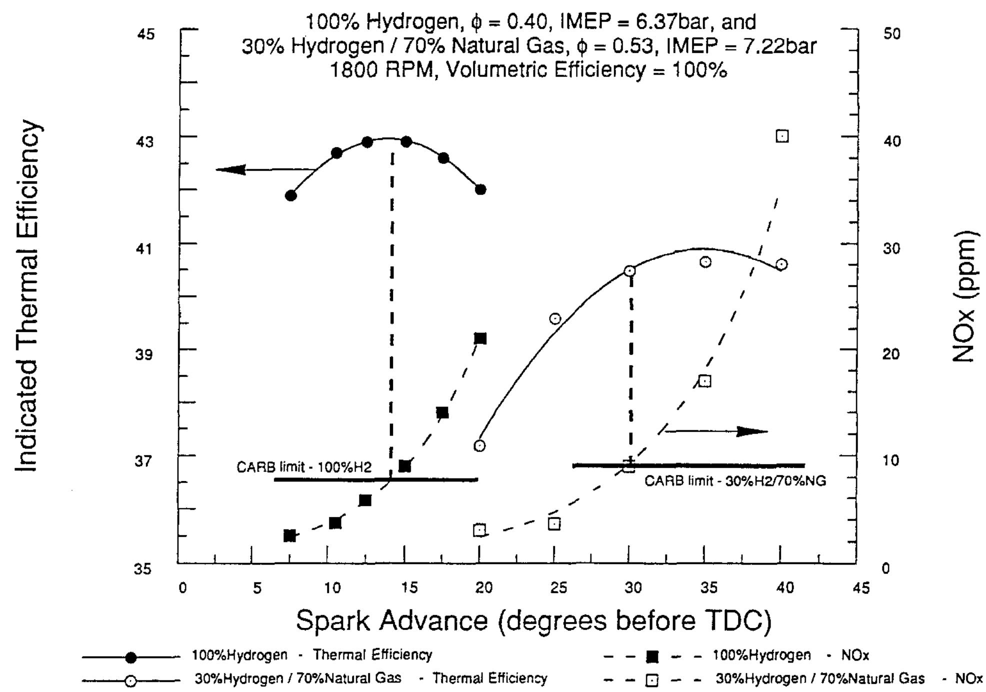

Fig. 6. Indicated thermal efficiency and NO_x vs spark advance. 100% Hydrogen, $\phi = 0.4$, IMEP = 6.37 bar, and 30% hydrogen/70% natural gas, $\phi = 0.53$, IMEP = 7.22 bar, 1800 rpm, volumetric efficiency = 100%.

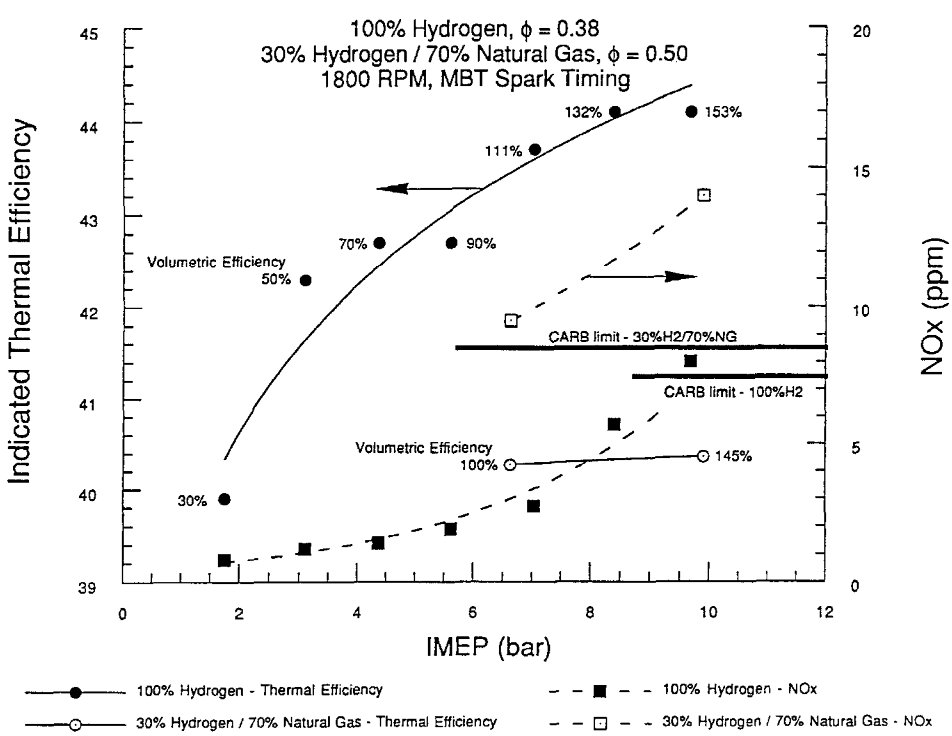

Fig. 7. Indicated thermal efficiency and NO_x vs indicated mean effective pressure. 100% Hydrogen, $\phi = 0.38$, 30% hydrogen/70% natural gas, $\phi = 0.50$, 1800 rpm, MBT spark timing.

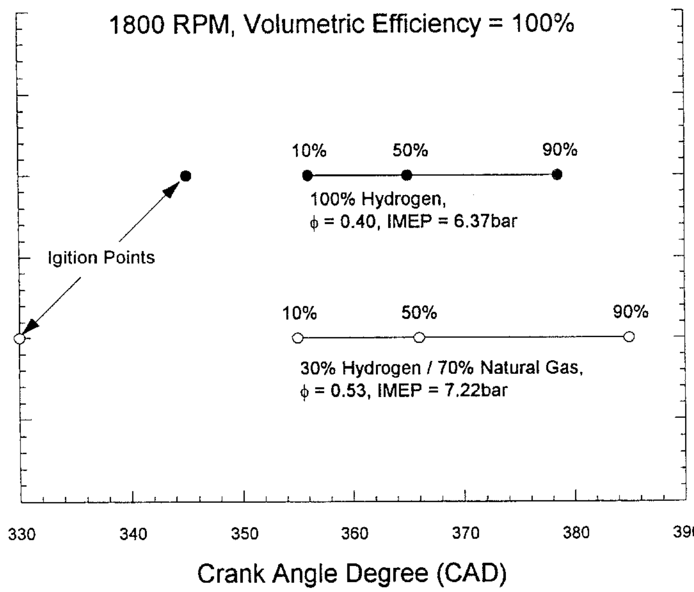

Fig. 8. Burn duration at specific crank angle degrees.

$\phi = 0.52$ for the 30% hydrogen fuel, one more plot is included. Figure 8 shows the burn duration of the two fuels at the proposed EZEV NO_x operating point. The burn duration for each is calculated from the respective cylinder volume–pressure data, according to the method of Rassweiler and Withrow [4].

DISCUSSION

It should be noted that this engine is being optimized for operation on hydrogen, not a mixed fuel. Thus the reaction rate and burn duration are not optimized for these slower burning mixtures. This effect may be causing the falloff in efficiency with reduced ϕ for the mixed fuel and the natural gas relative to hydrogen.

What does appear to be of interest is the possibility of operating a hybrid vehicle engine on a 30% hydrogen blend at equivalent zero emission vehicle specifications with no exhaust gas after treatment and at higher equivalence ratio than is possible with hydrogen. The 30% hydrogen fuel is an attractive option (compared with hydrogen) due to both the higher output of the engine and the storage density advantage when stored as a pressurized gas.

The indicated efficiency penalty measured in this test series (6% less for the 30% hydrogen fuel relative to hydrogen) is both small and, possibly, correctable. Figure 8 shows that the burn duration of the 30% fuel is longer, perhaps too long for peak efficiency.

The low equivalence ratios required to meet these stringent emission standards do not produce attractive power densities in naturally aspirated engines. Super- or turbocharging would increase this power density to a more useful range, and the effect on NO_x emissions and indicated thermal efficiency is shown in Fig. 7. It can be seen that while there is an increase of NO_x, the sensitivity is small. In addition, indicated thermal efficiency is improved to a peak value of 44%, the highest recorded on this engine. Note that the two points for the 30% hydrogen fuel appear to follow the hydrogen performance (note, this data was recorded at peak torque values): it is possible to reduce these NO_x values significantly with slight retardation of the spark advance.

CONCLUSION

Results from the Onan single cylinder research engine demonstrate that it is possible to build a high efficiency, equivalent zero emissions auxiliary power unit for hybrid vehicles fuelled by hydrogen or 30% hydrogen/70% natural gas blends.

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