

A Comparative Study of a Spark Ignition Engine Running on Hydrogen, Synthesis Gas and Natural Gas

Pedro Orbaiz, Michael J. Brear, Payman Abbasi and Peter A. Dennis
University of Melbourne

ABSTRACT

This paper presents an experimental, numerical and theoretical study of the performance of the same spark ignition engine running on four different gaseous fuels: hydrogen, two synthesis gases and natural gas. Measurements of the brake thermal efficiency, the combustion variability, the engine out emissions and the indicated, pumping and friction mean effective pressures are first presented, with particular interest placed on the lean burn performance. Combustion analysis is then undertaken, with the crank angle resolved in-cylinder turbulence and the flame propagation plotted on the so-called 'Bradley diagram' for turbulent premixed combustion. The loci of the combustion events on the Bradley diagram are then used to explain the observed, relative performance of the engine running on these four fuels. In particular, turbulent flame stretch is argued to play a significant role in determining variations in engine performance on these four fuels, and also appears to explain the ability of hydrogen-rich fuels to perform well over a relatively wide range of lean engine operating conditions.

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INTRODUCTION

As the world's energy demand and the need to reduce anthropogenic greenhouse gas emissions both grow, the use of alternative fuels is of increasing interest. 'Synthesis gas' or 'syngas' is a common alternative fuel, and is a gas mixture of varying composition but always containing carbon monoxide and hydrogen. It can be produced from a variety of sources such as coal, biomass, organic waste and natural gas, amongst others, and is already widely used in a number of industrial applications, particularly power generation [1,2,3] and liquid fuel synthesis via Fischer-Tropsch conversion of coal or natural gas [4, 5]. Syngas production from solid fuels, such as coal or biomass, normally involves the partial oxidation and gasification of the fuel source. When using gaseous fuels as feedstocks, such as biogas or natural gas, syngas is normally produced by steam reforming. Indeed, steam reforming of natural gas is the process by which around 85% of the world's hydrogen is produced [6,7,8]. Further, the integration of coal gasification or natural gas reforming with various power generators is a proposed means of achieving so-called 'pre-combustion carbon capture', and is thus a key part of many proposed, future carbon-capture and storage (CCS) power generation systems [9]. As such, the combustion of syngas from coal, natural gas or renewable resources is a problem of growing interest, particularly given the recent large growth in

conventional and unconventional natural gas production in Australia, the United States and elsewhere [10, 11].

Since the composition of these syngases can range widely in terms of hydrogen, carbon monoxide, methane and diluent content, it is difficult to make general conclusions about how these fuels perform in reciprocating engines. Nonetheless, a significant literature has been published on the combustion of mixtures of hydrogen-methane and different syngas compositions in reciprocating engines e.g. [2, 12,13,14,15,16]. Hydrogen and methane have proved to be compatible fuels in several respects. Both have high octane numbers, the higher volumetric energy density of natural gas makes it a more effective energy carrier, and hydrogen's broader flammability limits and high flame velocity allow it to burn in more dilute conditions. The inclusion of carbon monoxide to make a syngas mixture appears to improve the knock characteristics without affecting the thermal efficiency of the engine [16, 17].

These trends can be explained by the fundamental physics involved. The laminar flame speed of hydrogen-methane-air and syngas-air mixtures increases significantly with hydrogen content [17,18,19,20,21,22,23]. Moreover, adding hydrogen to methane results in a fuel less prone to quenching by flame stretch [19], and accelerates the initiation of flame propagation [19, 24]. Also, after the flame kernel is initiated,

the rate of turbulent flame propagation is also increased significantly [19, 24]. In hydrogen-methane fuelled engines, this is known to result in a shorter combustion duration, more retarded minimum spark advance for best torque (MBT) and thus, potentially, higher engine efficiency [15, 25, 26, 27, 28].

Engine out emissions have also been extensively reviewed. Although most of the literature shows that the NO_x emissions rise as the percentage of hydrogen in the fuel is increased, there are some studies that show either no change or the opposite effect [12, 29]. The increase of NO_x emissions with hydrogen content is attributed to higher peak combustion temperatures [13, 14, 15, 27, 30]. However, the extension of the lean limit allows stable engine operation at leaner conditions which then results in very low NO_x emissions [12, 13, 14, 25, 30, 31]. Furthermore, the higher tolerance of hydrogen to exhaust gas recirculation (EGR) allows the use of dilute stoichiometric air-fuel mixtures, and thus reduced NO_x emissions [29, 32, 33]. Work done on syngas combustion shows that NO_x emissions are not only affected by the percentage of hydrogen in the fuel, but also by that of CO [15, 30]. This is again attributed to CO increasing the mixture's combustion temperature.

The effects of hydrogen addition on the unburned hydrocarbon emissions (UHC) emissions are consistent throughout the literature [12, 13, 14, 15, 25, 27, 30, 31]. Due to the increased burning velocity of the fuel charge and to the inevitable reduction in hydrocarbon content in the fuel as hydrogen content increases, increased hydrogen content always results in a reduction of the UHC engine out emissions for both hydrogen-methane and different syngas compositions. However, as the lean limit of the engine is approached, slow combustion and misfire can cause the UHC emissions to rise.

Similarly, increasing the hydrogen content in the fuel causes CO emissions to fall. Again, as the lean limit of the engine is approached, the slow and unstable combustion process can result in increased CO emissions [12, 13, 14, 15, 25, 27, 30, 31]. In the case of hydrogen-methane mixtures this is only due to the partial oxidation of methane. However, in the case of syngas, unburned CO in the fuel can also be an important source of emissions [1, 26, 30].

This paper intends to relate such observed behaviour to existing, turbulent combustion theory. This involves a systematic study of the same spark ignition engine running on four different gaseous fuels: hydrogen, two synthesis gases and natural gas. Of particular interest is the identification of the fundamental parameters determining how turbulent flame propagation affects engine performance on all of these fuels. This is undertaken by comparing the observed engine performance to combustion analysis from in-cylinder pressure measurements.

EXPERIMENTAL METHODS

All experiments were conducted on a six cylinder, four litre, naturally aspirated Ford BF engine with a 10.3:1

compression ratio. The combustion chamber has a pent roof shape with the sparkplug located in its centre. The engine is equipped with four valves per cylinder actuated by double over head cam shafts, both of which have independent variable cam timing (VCT). Fuel is supplied through a multipoint port injection system. Table 1 shows the engine specifications.

The engine was fitted with a Prins Compressed Natural Gas Vapour Sequential Injection Kit, VSI-CNG. Although designed for natural gas, this unit is able to work with other gases. Engine torque was measured using a Heenan-Dynamatic M K-II eddy current dynamometer, with a maximum power rating of 220kW at 2400 to 6000rpm.

Table 1. Engine specifications

Parameter	Value
Bore	92.25 mm
Stroke	99.31 mm
Compression ratio	10.3:1
Cylinders	6, in line
Capacity	3984 cm ³
Intake valve open	-27.5 to 32.5 CAD ATDC
Intake valve close	48.5 to 108.5 CAD BTDC
Exhaust valve open	78.5 to 18.5 BBDC
Exhaust valve close	-2.5 to 57.5 CAD ATDC
Intake centre - line	99.5 to 159.5 CAD ATDC
Exhaust centre - line	131.5 to 71.5 CAD BTDC
Intake duration	256 CAD
Exhaust duration	256 CAD

An Auto-diagnostics ADS9000 gas analyser was used to measure O_2 , CO , CO_2 , NO and hydrocarbons (HC). This device uses a NDIR sensors to measure the concentrations of CO , CO_2 and HC and an electro-chemical cell for NO and O_2 . The O_2 , CO , CO_2 and NO calibration was performed using a conventional span gas and gas divider. The HC measurements, on the other hand, were calibrated by direct comparison of the ADS9000 measurements with those of a fast FID analyser whilst running the same natural gas engine. Previous work done by the group also showed that the ADS9000 measurements show good agreement with a Horiba certification quality analyser [34].

The in-cylinder pressure traces were measured using a Kistler piezoelectric pressure transducer, which was installed in the side of the 6th cylinder combustion chamber. A charge amplifier was used to convert the transducers signal into a potential difference which was consequently read by a Labview card installed in a desktop PC. The final signal was then analysed by an in-house developed software. A rotary incremental encoder with a 0.5° resolution, mounted on the end of the engine's crank shaft, was used to synchronize the pressures transducer signal with the instantaneous volume of the combustion chamber.

In order to have a real-time λ measurement of the exhaust gases, a Bosch LSU 4.9 heated wide band oxygen sensor (UEGO) was installed on the engine's exhaust system directly after the manifold. Equipped with a 'NERSET' oxygen sensor

measuring cell and an oxygen ion pump, the UEGO sensor has a measuring range of $0.7 < \lambda < 5$. This makes it adequate when working with hydrogen based fuels.

Integrated into the engine's ECU, an ATI Vision system was used to change, read and record the engine's operating parameters on real-time. This allowed for the optimization of parameters such as spark timing, injection timing and injection duration. Also, by integrating a data acquisition board, the ATI system worked as a logging tool for the different external sensors: UEGO sensor; Coriolis flow meter; dynamometer load cell; ambient air, cooling system and exhaust gas thermocouples; and the pressure transducers.

A gas mixing system was designed and installed outside the engine test cell. The system was designed around the engine fuel line in such a way that the test cell instrumentation could be used throughout the mixing process. The gases were passed through a CMF series Micro motion Coriolis mass flow meter and stored in the syngas tanks. The amount of each constituent was calculated based on their molar fractions and the law of ideal gases. In order to corroborate the flow meter measurements, partial pressure verifications were made for each species.

All fuels were tested under steady state conditions at three different combinations of speed and load and varying λ values. Spark sweeps were performed to obtain the minimum advance for best torque (MBT). However, no CAM timing sweeps were made, meaning that ECU CAM timings programmed for gasoline were used.

Natural gas was first tested to establish a baseline. The two potential reformates were then tested, 'syngas 900' and 'syngas 1100'. Their composition emulates that of the equilibrium dry products of steam-methane reforming, with a steam-methane ratio of 3 and reforming temperatures of 900K and 1100K respectively. Finally, 99% pure hydrogen was used to analyse the benefits of complete methane conversion and CO_2 capture before combustion. It is noted that natural gas was used to mix the syngases instead of pure methane. The composition of these four fuels is shown on Table 2.

Table 2. Molar composition of the examined fuels (natural gas composition [35]).

Species	Natural Gas	Syngas 900	Syngas 1100	H_2
CH_4	91.5	14	0.6	0
C_2H_6	4.7	0.7	0.03	0
C_3H_8	0.8	0.2	0 (approx)	0
C_4H_{10}	0.1	0.03	0 (approx)	0
C_5H_{12}	0.03	0 (approx)	0 (approx)	0
C_6H_{14}	0.07	0.02	0 (approx)	0
N_2	1.1	0.03	0.01	0
H_2	0	66.7	76.3	99.9
CO	0	5.4	15.1	0
CO_2	1.8	13	7.8	0
Others	0.005	0.05	0.07	0.1

The combustion properties of these fuels are calculated based on the properties of their constituents. Table 3 shows the fuels' lower heating values (LHV), stoichiometric air-fuel ratios by mass (SAFR) and stoichiometric laminar flame speeds (SLFS), which in the case of both syngases was estimated using approximations [21, 36].

Table 3. Fuel combustion properties [21, 36, 37]. (The SLFS is calculated at 1 bar and 298K).

	Natural Gas	Syngas 900	Syngas 1100	H_2
LHV [MJ/kg]	46.3	26.8	25.2	121
SAFR (mass)	16.2	8.44	6.98	34
SLFS [cm/sec]	~40	~150	~170	~210

NUMERICAL METHODS

The simulation tool GT-Power is used to analyse the engine combustion process. The so-called 'reverse run combustion simulation' uses the recorded in-cylinder pressure traces as well as other additional data including cylinder geometry, spark timing, fuel mass and composition, the engine volumetric efficiency and the exhaust composition, to calculate the crank angle resolved mass fraction burned (MFB). It does this using a two zone model of the combustion process [38]:

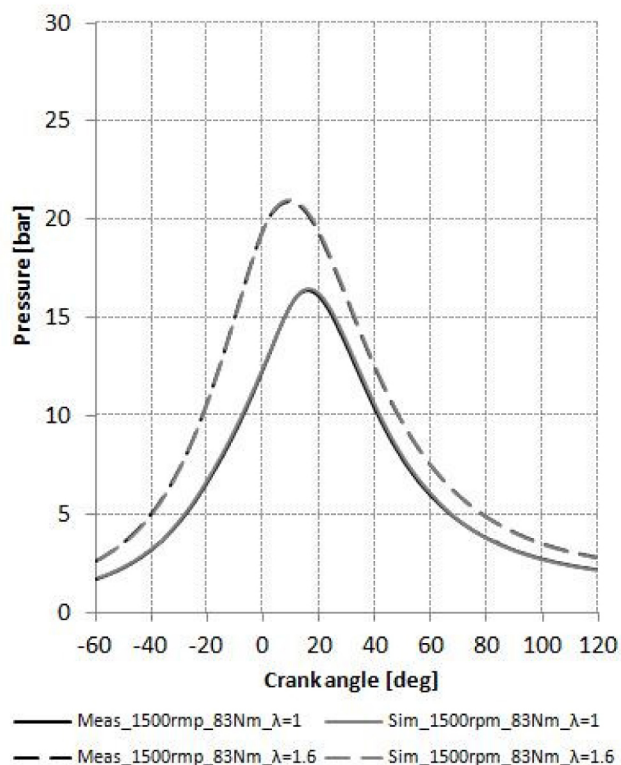
1. At the start of the combustion process (spark discharge), the entire mass inside the combustion chamber is in the unburned zone. This includes the air-fuel mixture and the residual gases of the previous cycles.

2. As the combustion process evolves, a certain amount of unburned mass is transferred to the burned zone at each time step. The mass transfer is calculated based on conservation of energy, and aims to match the recorded pressure trace, given the combustion chamber geometry and the modelled heat losses. These are calculated based on Woschni's heat transfer correlation. Literature shows that while this correlation has a poor predictive accuracy when used for gaseous fuels, it provides reasonable results when calibrated at each engine operating condition [39,40,41,42,43].

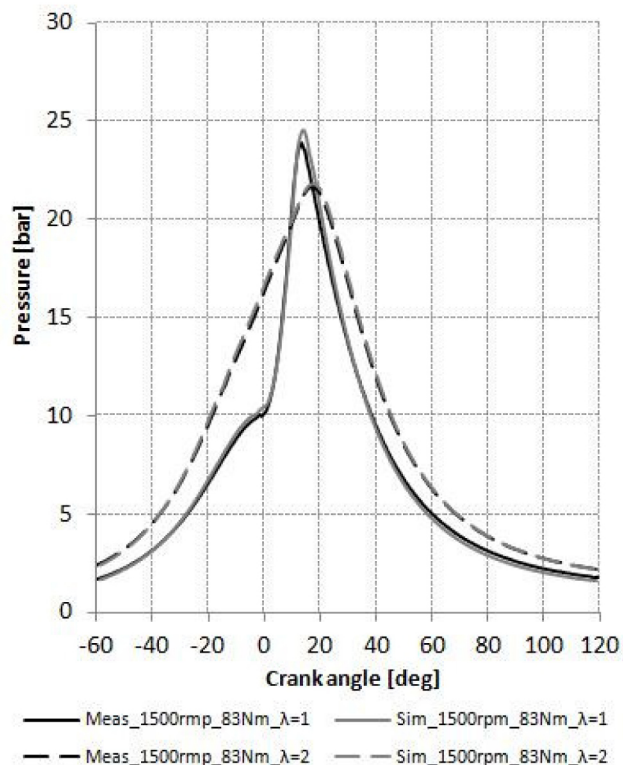
3. At the end of each time step, the unburned and burned zone temperatures are calculated and a chemical equilibrium calculation of the burned zone is done to determine its composition.

Figure 1 shows the pressure traces of both the measured and the reverse run combustion simulations for the different fuels at some of the evaluated λ . Due to the relatively close agreement between the measured data and the simulation results, it can be argued that the combustion model appears to replicate reasonably the combustion events of the different fuels at the different engine operating conditions.

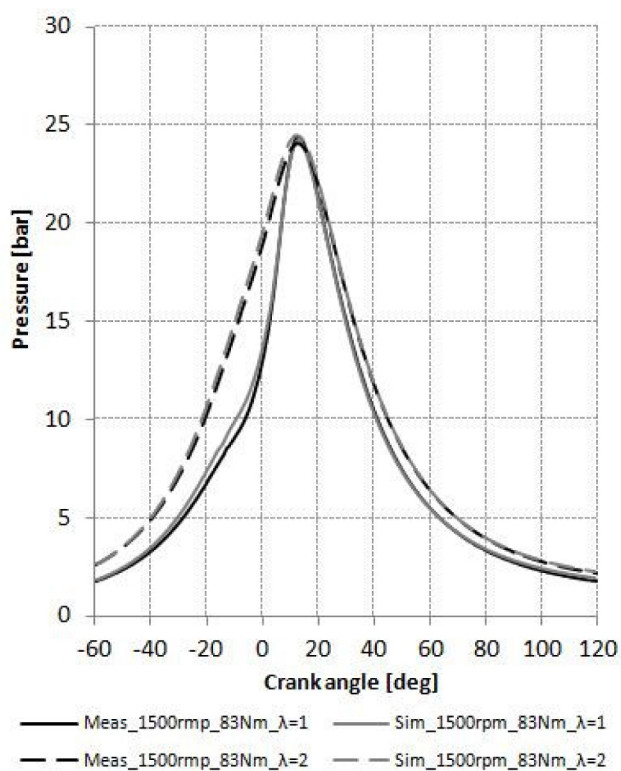
GT-Power's predictive turbulent flame model is then used to further understand the premixed combustion of the different fuels at the varying engine operating conditions [38].



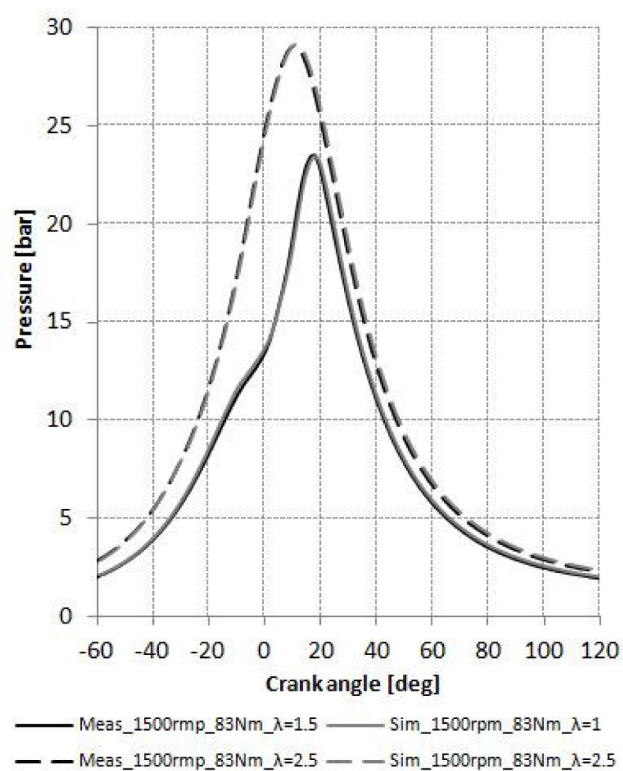
a. Natural gas



c. Syngas 1100

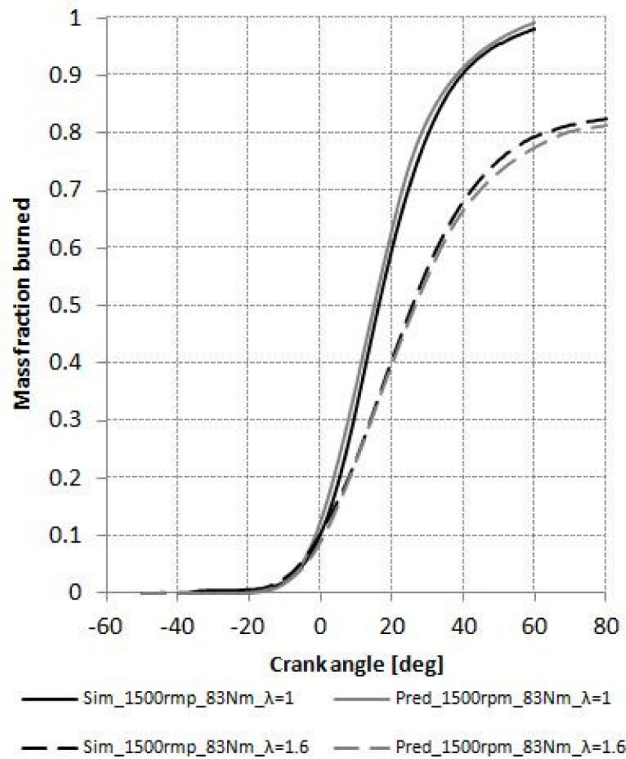


b. Syngas 900

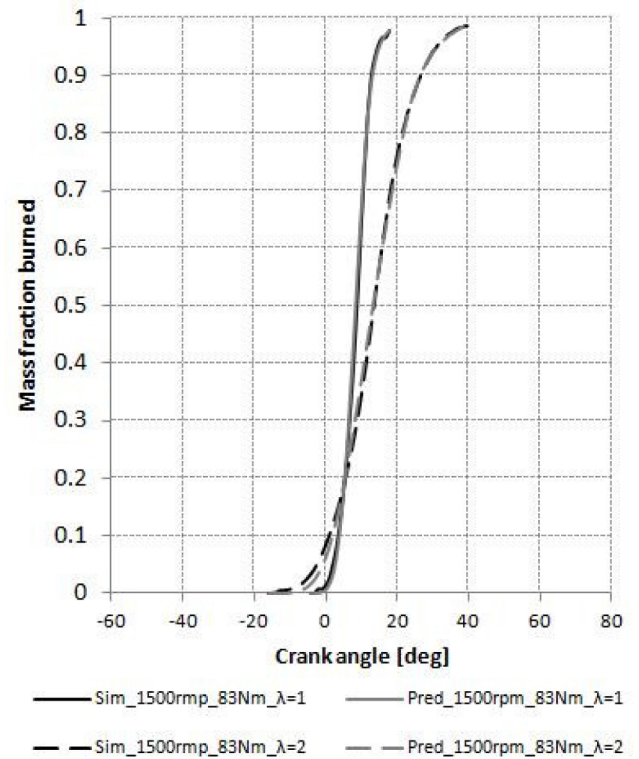


d. Hydrogen

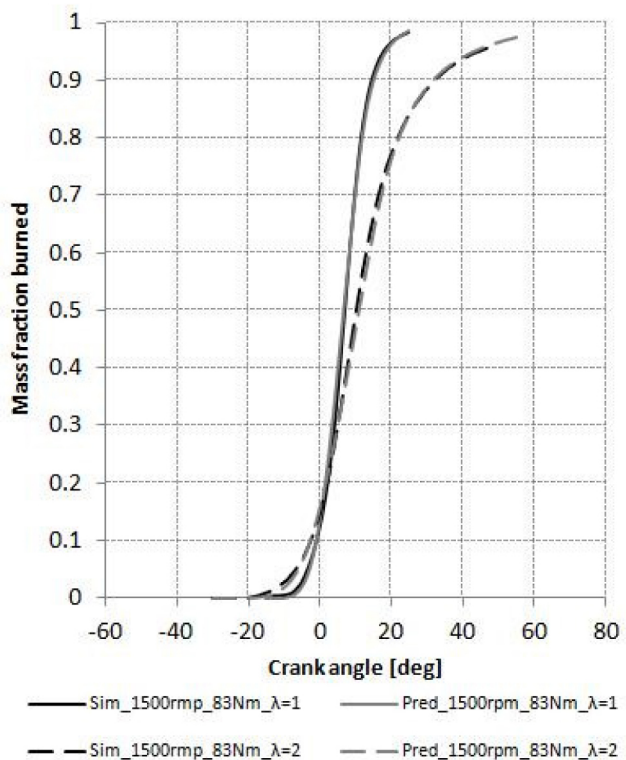
Figure 1. Measured and 'reverse run' in-cylinder pressure versus crank angle degree for the different λ , at 1500rpm and 83Nm.



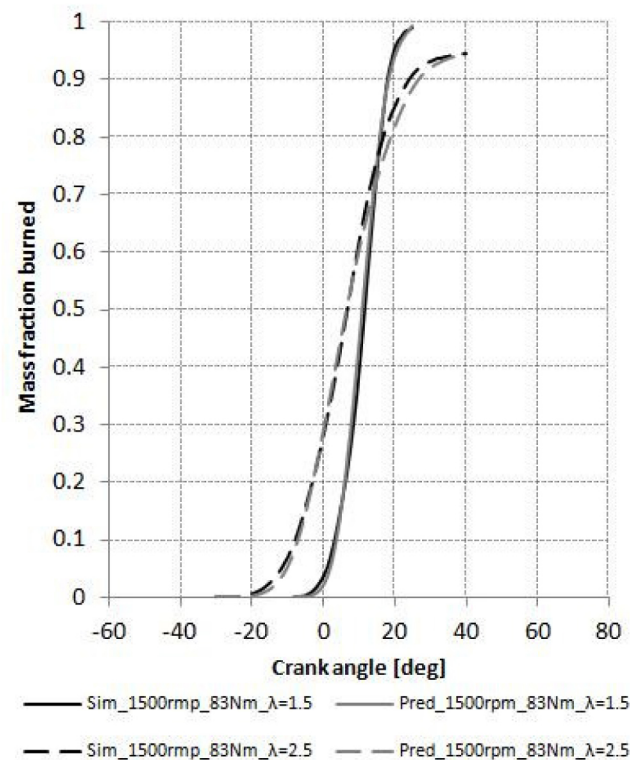
a. Natural gas



c. Syngas 1100



b. Syngas 900



d. Hydrogen

Figure 2. Mass fraction burned versus crank angle for the ‘reverse run’ and ‘predicted run’ for the different fuels at 1500rpm, 83Nm and different λ .

44,45,46]. The effects of turbulence on the flame propagation are adjusted by matching the calculated reverse run MFB curves to those generated by the predictive model. This is done by calibrating parameters which independently affect the flame kernel development time, the turbulent flame speed and the Taylor micro scale [38]. Figure 2 shows the MFB of both the predictive and non-predictive runs for some cases of each of the fuels. Again, the close agreement indicates that the predictive flame model is performing reasonably.

RESULTS AND DISCUSSION

Engine Brake Thermal Efficiency

The brake thermal efficiency (BTE) of the engine is defined as

$$\eta_{bte} = \frac{T_b \Omega}{m_f LHV_x'}$$

where T_b is the engine's brake torque [Nm], Ω is its angular speed [rad/sec], m_f is the fuel mass flow [kg/sec] and LHV_x is the lower heating value of fuel x [MJ/kg]. Figure 3 shows the engine's BTE versus λ for the different fuels at three different engine operating conditions.

Effects of Fuel Composition on Engine BTE

Due to the presence of consistent backfire, stoichiometric engine operation was not possible with pure hydrogen. However, this was not the case when running at $\lambda = 1$ on either of the syngases, which despite their high hydrogen content did not present any combustion abnormalities. This is likely due to the dilution of hydrogen in the other fuel constituents [16, 17]. Figure 3 shows that all fuels achieve similar peak brake thermal efficiencies, with a progressive trend of increasing for peak efficiency with increasing hydrogen content. Syngas 1100 shows very similar results to hydrogen along the entire λ sweep and at all engine operating conditions. This is perhaps expected due to hydrogen's broader flammability limits, lower activation energy and higher laminar flame speed [13, 14, 47,48,49].

Effects of Fuel Composition on Combustion Variability

The coefficient of variability of the indicated mean effective pressure (CoV of IMEP) is a measure of the cycle to cycle variability [50]. It is defined as

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}}$$

where σ_{IMEP} and $\overline{IMEP}$ are the standard deviation and mean value of the indicated mean effective pressure (IMEP) respectively. They are both calculated using in-cylinder pressure traces. Increasing values of the CoV of IMEP are an indication of less stable combustion and therefore can be used to determine the lean limit of the engine. Figure 4 shows the

CoV of IMEP of the different fuels versus λ at varying engine operating conditions. It shows a growth of combustion variability past λ for peak efficiency. This indicates that at leaner air-fuel mixtures the deterioration of the combustion process offsets other benefits of lean engine operation. It also shows that natural gas has a much higher CoV of IMEP than the hydrogen bearing fuels and that increasing both engine speed and load causes the CoV of IMEP of all fuels to drop. All of this is strongly related to the combustion characteristics of the different fuels and is discussed further in the following section.

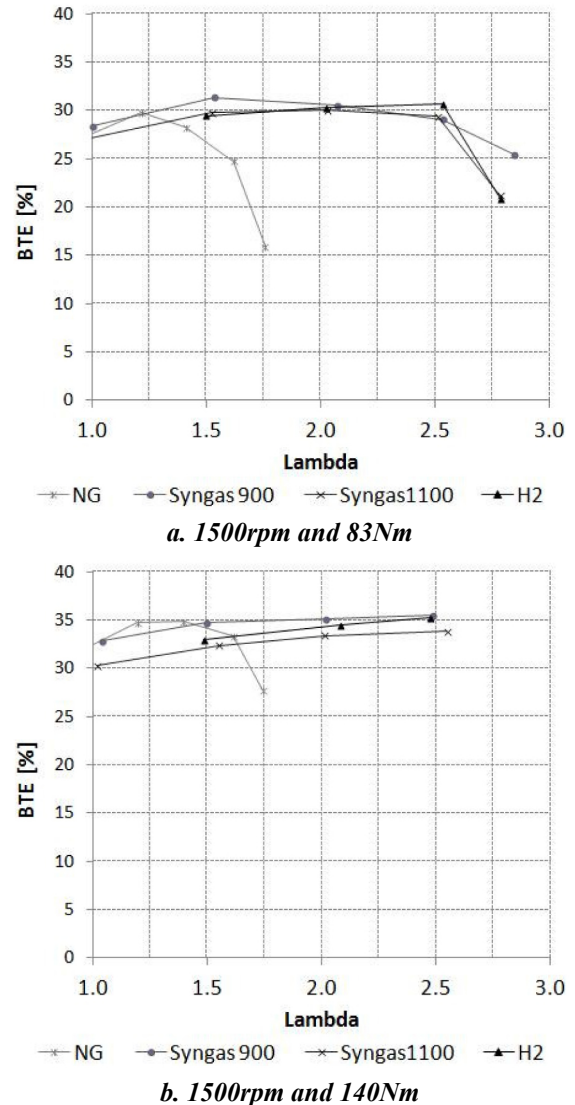


Figure 3. Engine BTE versus λ for the different fuels and different engine operating conditions.

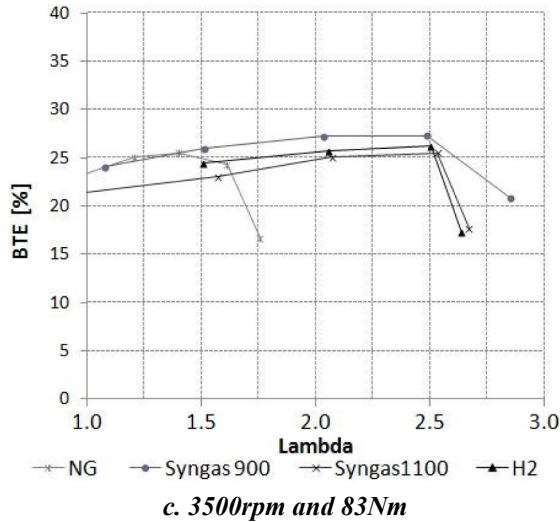


Figure 3 (cont). Engine BTE versus λ for the different fuels and different engine operating conditions.

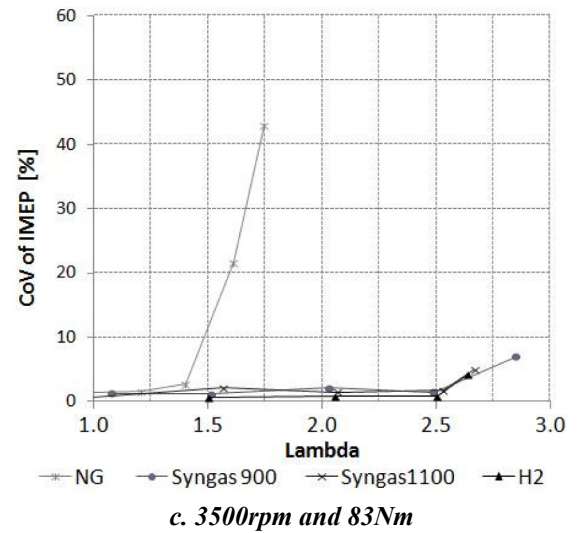
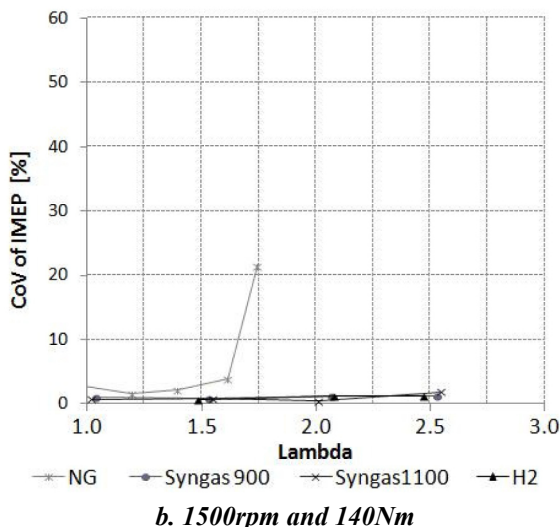
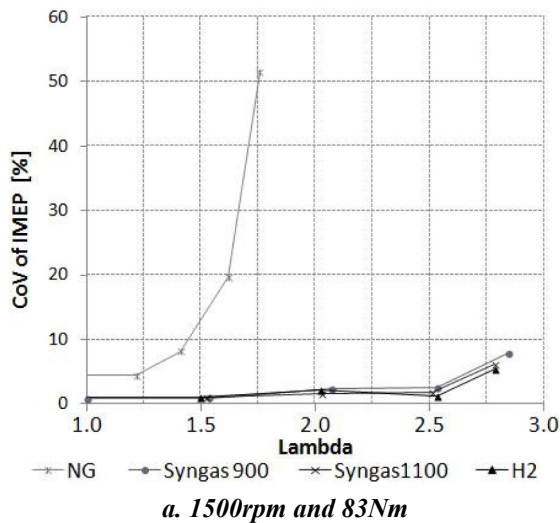


Figure 4. CoV of IMEP versus λ for the different fuels and engine operating conditions.



Some Effects of Speed and Load on Engine BTE

Engine Friction work can be calculated based on the engine's friction mean effective pressure (FMEP)

$$\overline{FMEP} = \overline{IMEP} - \overline{BMEP},$$

where the $\overline{IMEP}$ is calculated over the entire engine cycle using the recorded in-cylinder pressure trace and $\overline{BMEP}$ is the brake mean indicated pressure, which can be written as

$$\overline{BMEP} = \frac{4\pi T_b}{V_d},$$

where V_d is the engine's displacement [L]. If expressed relative to the fuel energy, then the friction losses are

$$W_F[\%] = \frac{\overline{FMEP} V_d \Omega}{4\pi m_f LHV_x}.$$

Losses due to pumping work can be calculated based on the pumping mean effective pressure (PMEP), which in turn is calculated over the admission and exhaust strokes of the engine cycle using the in-cylinder pressure trace. Relative to the fuel energy, pumping losses can be written as

$$W_p[\%] = \frac{\overline{PMEP} V_d \Omega}{4\pi m_f LHV_x}.$$

Figure 5 shows the averaged friction losses for the different engine operating conditions and pumping losses, averaged across the λ sweep, for the different fuels and engine operating conditions. It shows that while increasing load has relatively little effect on friction losses, it considerably reduces the pumping losses. The later is mainly due to reduced throttling. On the other hand increasing engine speed intensified both friction and pumping losses. Both this