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ACHIEVEMENTS AND STATE OF THE ART OF HYDROGEN FUELLED IC ENGINES AFTER TWENTY YEARS OF RESEARCH AT GHENT UNIVERSITY

Sierens R.* ,Verhelst S., Demuynck J. and Vancoillie J.
*Author for correspondence
Laboratory of Transport Technology,
Ghent University,
Sint Pietersnieuwstraat 41,
B9000, Gent, Belgium,
E-mail: Roger.Sierens@ugent.be

ABSTRACT

Hydrogen could be “the” fuel for the future, not only for fuel cells but certainly for internal combustion engines.

The research on hydrogen started at Ghent University in 1990 with the adaptation of a Valmet diesel engine to hydrogen operation (atmospheric, carbureted version) to prove the capability of hydrogen as a fuel for IC engines. Since then several engines were modified for hydrogen use with the state of the art technologies (sequential injection, electronic management units). With European (Craft, Brite) and Belgian grants three buses demonstrated on several levels the application of hydrogen IC engines. At the moment the laboratory test proves an operation with a power output higher than the gasoline engine, with an equal efficiency of the diesel engine and with very low emissions (NO_x less than 100 ppm).

The interests of the research group of Ghent University was not only for the experimental work, but also the combustion process is simulated (GUEST code). The estimated formula of the laminar flame speed of hydrogen by Verhelst is worldwide used in other research studies. At the moment a doctoral study examines the heat transfer in hydrogen engines, which is so different from the already not very accurate heat transfer models in gasoline and diesel engines.

In our laboratory tests, the hydrogen engine is ready for mass production (backfire safe, high power output, high efficiency, very low emissions). But storage on the vehicle recently and infrastructure of the fuel delivery are the bottle-necks for a real implementation of the hydrogen economy. From hydrogen, methanol can be produced on a sustainable way. Methanol is a liquid (no storage problem on het vehicle) and with minor modifications the same infrastructure can be used as for gasoline. Methanol has very good engine characteristics. Will methanol based on hydrogen be then “the” fuel of the future?

KEYWORDS

Hydrogen, combustion engines, power output, efficiency, emissions

INTRODUCTION

For improving the quality of live, it is essential to have access to economically priced energy and to make its use clean and efficient. Certainly the transportation sector mainly depends on petroleum products (fossil fuels). The combustion of these fossil fuels generates useful energy but also results in emissions of pollutants and greenhouse gases to the atmosphere.

Hydrogen is an alternative. The potential of hydrogen as a fuel for the future will benefit not only air quality and climate change, but also offers a solution to national energy sustainability and security.

Research on hydrogen fuelled internal combustion engines is going on at many places (car companies, research centers, universities) and for a long time. This paper is concentrated on the research at Ghent University, with the start of literature study on hydrogen in 1990, the first engine set up and tests in 1991 and the first paper in 1992 [1].

Different engines have been adapted for hydrogen use. A Valmet 4-cyl diesel engine (carburetted), a GM 8-cyl natural gas engine (carburetted, later with port-fuel injectors and electronic control unit) which can run on natural gas, hydrogen and mixture of natural gas and hydrogen. All further (adapted) engines are port-fuel injected (and with ECU): a one cylinder CFR engine, a one cylinder research Audi-NSU engine, a 4-cyl Volvo engine (gasoline or hydrogen). The last two engines are modified recently to run also on methanol.

At the moment all engines are with port-fuel injection, where two strategies are mainly used (or the combination of the two). First, lean operation, at or leaner than of $\lambda = 2$ with WOT (wide open throttle). Second, stoichiometric operation (in fact just rich, $\lambda \cong 0.97$), exhaust gas recirculation (EGR) and the use of a three way catalyst (TWC). The rich condition, with hydrogen in surplus, acts as a reducer in the TWC for the NO_X reaction to N_2. Both strategies can also be combined with supercharging.

The use of hydrogen in a spark ignition IC engine is being developed as a low-cost solution. In spite of the advantages of hydrogen IC engines, some issues remain. There is also a trade-off between power output and NO_X emissions. Also the occurrence of abnormal combustion phenomena like knock, pre-ignition and backfire have to be considered [2].

Advantages of hydrogen for use in internal combustion engines are its wide flammability limits and its fast burning velocities, even at lean conditions. Therefore a hydrogen fuelled IC engine can operate at a wide range of the air-fuel ratio λ ($\lambda = 1/\Phi$). This makes it possible to run the engine from stoichiometric ($\lambda = 1$) to very lean ($\lambda = 4$) with wide open throttle (WOT). In this way the power output of the engine (load condition) is regulated by the air-fuel ratio, without throttle losses and therefore increasing the efficiency. With the high flame speed (fast burning velocities), even at lean conditions, the thermodynamic efficiency will also increase.

But there are also some disadvantages. At lean condition, between $1 < \lambda < 2$ the NO_X emissions are too high, and difficult to reduce in a catalyst because of the oxidizing environment. Only leaner than $\lambda \cong 2$ the NO_X emissions are very low. The operating range of $1 < \lambda < 2$ therefore has to be avoided.

Secondly, hydrogen has a very low density (high specific volume), so that at stoichiometric operation, one third of the combustion chamber is filled with hydrogen. This results theoretically in a 18% power loss against a gasoline engine (for the same volume of the combustion chamber, atmospheric).

In the next chapter some results of the experiments at Ghent University will be given. In appendix three demonstration projects, the Greenbus, the ZEMBUS and the CTPT bus are summarised (year of presentation, partners, sponsoring, engine type, storage system of hydrogen).

EXPERIMENTAL RESEARCH

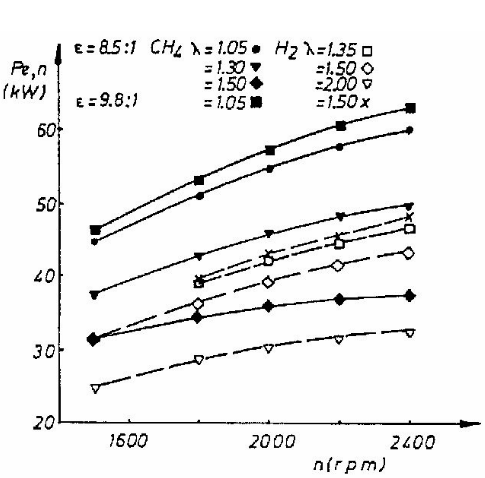

Figure 1 Power output of the Valmet engine fuelled with natural gas or hydrogen

The first tests on a modified diesel Valmet engine at Ghent University started in 1991. The aim of these test was to study the possibilities of the hydrogen fuelled engine: power output, emissions and stable behaviour of the engine (no backfire). At the same time tests with natural gas were carried out for comparison [3].

The original Valmet diesel engine has a power output of 64 kW, which can be reached also with natural gas (CH_4) but not at all with hydrogen (due to the lean conditions to avoid backfire for the carbureted fuel supply system), see Fig. 1. Figure 2 shows the NO_X emissions again for natural gas and hydrogen. At a certain air-to-fuel ratio the NO_X emissions for hydrogen are higher than for natural gas (and gasoline). Only at very lean mixtures ($\lambda \geq 2$), the level becomes acceptably low.

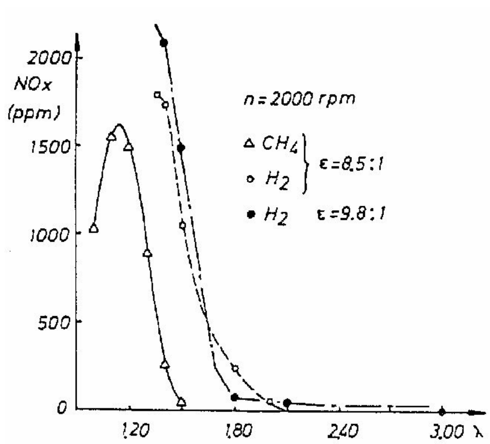

Figure 2 NO_X emissions of the Valmet engine fuelled with natural gas or hydrogen

The tests were done for two compression ratios. These tests show clearly that carburetted hydrogen engines never will obtain a comparable power output against a gasoline, natural gas or diesel engine. Also the higher the compression ratio, the leaner the mixture has to be to avoid backfire.

After these tests it was clear that the state of the art regulation systems should be used: port-fuel injection and electronic managements systems (direct injection has never been a point of research at Ghent University). In a next step numerous tests were carried out on a GM type 454 (Crusader) engine. An operating strategy using a variable air-fuel ratio as a function of engine load was evaluated both for naturally aspirated as well as supercharged conditions. Implementation of this variable air-fuel ratio strategy in a range from $2 < \lambda < 5/0.2 < \varphi < 0.5$ showed a more than 20% increase in engine power compared to a carbureted version without increasing the danger of backfiring [4]. The conclusions of this study also include that as a consequence of the wide range of applied mixture composition, the range of ignition timings is also wide.

As an example Fig. 3 shows the MBT ignition timing for a single cylinder research engine as a function of the air-fuel equivalence ratio, for different engine speeds, at wide open throttle. The large range in equivalence ratio leads to a large range in burning velocities, with lean, slow burning mixtures requiring more spark advance. The effect of engine speed is most obvious for the lean mixtures, where more spark advance is needed when less time is available (higher engine speed) and burning velocities are relatively low. Near stoichiometric, the burning velocities of hydrogen are so high that there is hardly any influence of the engine speed on MBT timing.

On this engine also a comparison is made between the use of hydrogen, natural gas or a mixture of hydrogen and natural gas [5]. After this all research was concentrated to increase the power output and efficiency, at low NO_X emissions and backfire safe operation.

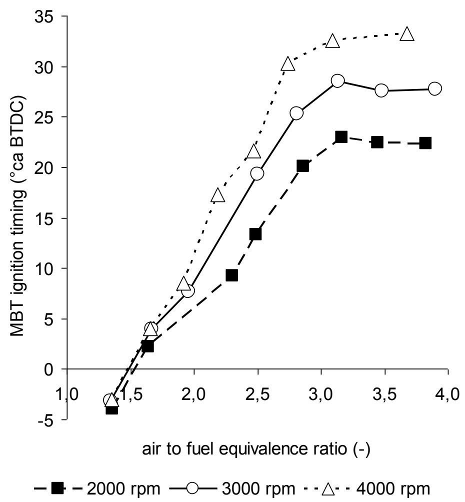

Figure 3 MBT ignition timing as a function of air-fuel equivalence ratio, for different engine speeds, WOT.

There are two options for hydrogen IC engines to obtain power outputs similar to or exceeding the power output of an equivalent, naturally aspirated, gasoline engine, without excessive tailpipe NO_X emissions. The first option is to stay lean of the ‘threshold equivalence ratio’, the air-to-fuel equivalence ratio below which NO_X emissions rise exponentially, and make up for the power loss caused by the lean mixtures through supercharging. The second option is supercharging at stoichiometric mixtures, or in practice at slightly rich of stoichiometric so that a small amount of unburned hydrogen is present in the exhaust which is an effective reducing agent for NO_X, using a three way catalyst.

The second option, running at stoichiometric, is not always possible without occurrence of abnormal combustion phenomena. Using exhaust gas recirculation (EGR) is a means to allow reliable stoichiometric operation. Furthermore, varying the EGR rate can be used to control the power output (as opposed to throttling) which benefits the engine efficiency, and NO_X emissions decrease because of the thermal inertia of the EGR gases. In [6] a comparison is made for the two strategies on a single cylinder research engine for all load and speed conditions.

Figure 4 shows the net BMEP and brake thermal efficiency versus engine speed as a function of the charging pressure. Above 1.2 barg charging pressure, there is no benefit in power output or efficiency. Maximum efficiency is obtained between 0.5 and 1 barg and the maximum brake power is reached for 1.2 barg. Thus, there is a trade-off between efficiency and power output.

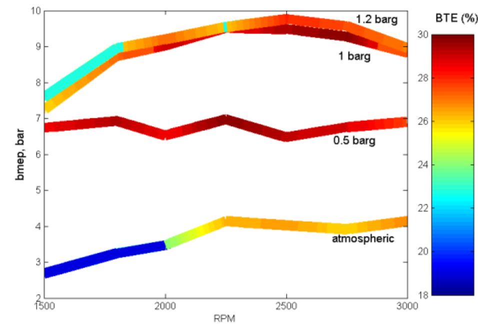

Figure 4 Net BMEP and brake thermal efficiency (BTE) versus engine speed, as a function of charging pressure, for lean-burn operation (engine-out $NO_X \leq 100$ ppm)

Supercharged lean-burn strategy offers the possibility to control the power output as a function of the charging pressure, with high efficiencies due to WOT operation. A TWC cannot be used to control excessive NO_X-emissions at high pressure and engine speed resulting in the necessity for leaner mixtures and thus in a power penalty. An alternative to decrease the excessive NO_X-emissions at high loads could be another catalyst such as a NO_X storage reduction catalyst (NSR) [7].

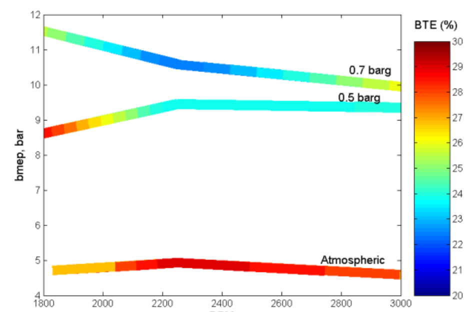

Figure 5 Net BMEP and brake thermal efficiency (BTE) versus engine speed, as a function of charging pressure, for stoichiometric operation + TWC + EGR (tailpipe $NO_X \leq 100$ ppm)

Figure 5 shows the net BMEP versus engine speed with the obtained brake thermal efficiency as a function of the charging pressure. A higher power output than with supercharged lean-burn operation is possible but a lower efficiency is noticed.

The brake thermal efficiencies are lower than for the supercharged lean-burn strategy, which has several causes. First of all, the higher heat-loss at stoichiometric operation leads to lower efficiency. In addition, to ensure the recirculation flow of exhaust gases, the inlet had to be throttled slightly (-0.1 barg), so operation at WOT was not possible. This causes additional pumping losses which decreases the efficiency. Another reason is the dilution of the mixture with EGR which decelerates the combustion resulting in lower efficiencies. On the other hand, the dilution of EGR decreases the cylinder temperatures resulting in lower heat loss. Therefore, it is difficult to define a trend in brake thermal efficiency. It might be concluded that supercharged stoichiometric operation with EGR leads to lower efficiencies. High supercharging needs more compressor power which results in a lower net power output. The main difference is the lower brake thermal efficiency that was observed, compared to supercharged lean-burn operation. Diluting the mixture with EGR implies a longer combustion duration, lower peak temperature and pressure and thus lower power output.

At the moment experiments are going on to compare efficiencies of hydrogen with gasoline and methanol on different engines. For the first tests an efficiency comparison with gasoline was done on a four cylinder sixteen valve gasoline engine with variable inlet valve opening with a total swept volume of 1783cc and a compression ratio of 10.3:1 converted to bi-fuel operation by mounting an additional fuel rail supplying gaseous fuel (in this case, hydrogen) to 8 Teleflex GSI gas injectors (2 per cylinder), mounted on the intake manifold. [8] gives an efficiency comparison between hydrogen and gasoline on this bi-fuel hydrogen/gasoline engine.

The efficiencies while using gasoline and hydrogen were compared at different torque settings (20, 40 and 80 Nm - equivalent to 1.41, 2.82 and 5.64 bar BMEP) and different engine speeds. At each point, MBT-timing and a fixed Intake Valve Opening (IVO) advance of 4°ca BTDC was used. The influence of the IVO advance on the brake torque is fairly limited (up to 3% torque rise), this justifies using a fixed IVO advance.

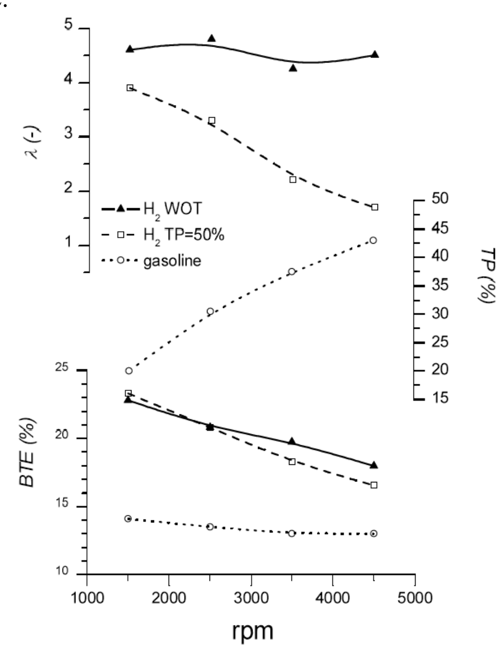

Figure 6 Brake thermal efficiency and respective throttle position or air-to-fuel equivalence ratio as a function of engine speed, for a fixed brake torque of 20 Nm

Figures 6 and 7 show the brake thermal efficiencies as a function of engine speed, for fixed torque outputs of 20 and 40 Nm respectively. Three BTE curves are shown, one is for gasoline (throttled, stoichiometric) operation for which the corresponding throttle position (TP) is shown in the middle part of the graph. The other two are for hydrogen with wide open throttle and with a TP=50% respectively, for these two cases the corresponding air-to-fuel equivalence ratios are given in the top part of the graph.

From Figs. 6 and 7 it is clear that at these low loads, the brake thermal efficiency on hydrogen is (much) higher than on gasoline, the hydrogen BTEs are 40 to 60% higher relative to the gasoline BTEs. This difference is due to the absence of throttling losses (or much lower throttle losses in the H_2 TP=50% case) and the lean mixtures for hydrogen. The higher burning velocity of hydrogen is also a contributing factor, as this leads to a more isochoric combustion. The influence of this factor can be seen directly in Fig. 6 for the 4500 rpm point: the throttle position for gasoline is about 50% there, so the efficiency can be compared to the H_2 TP=50% case. The BTE on hydrogen for this condition is about 18% higher relative to gasoline. This difference is not entirely due to a difference in burning velocity however: as hydrogen displaces more air due to its low density, throttling losses are lower even though the throttle position is identical, as the lower air flow results in lower flow losses.

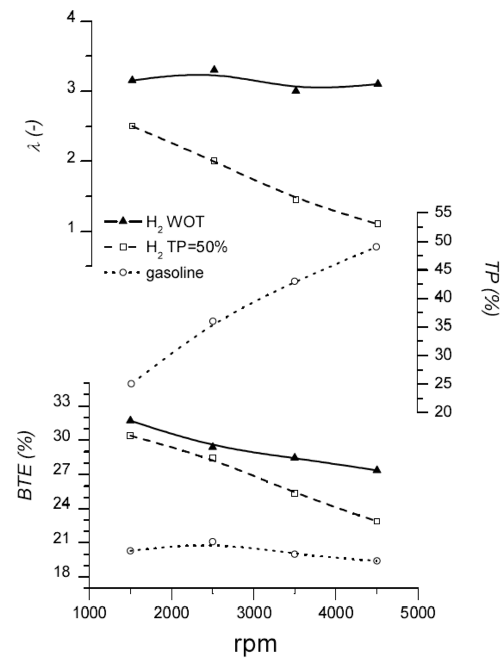

Figure 7 Brake thermal efficiency and respective throttle position or air-to-fuel equivalence ratio as a function of engine speed, for a fixed brake torque of 40 Nm

For hydrogen, the BTE decreases with engine speed, although the decrease is less pronounced for the WOT case. Two effects explain this behavior: first, due to the lean-burn operation and large throttle openings, the air flow is much higher in the hydrogen case than for gasoline. This leads to higher flow losses in the intake manifold. To keep the torque output fixed, this means more hydrogen needs to be injected at higher engine speeds to compensate for these higher losses, with a reduced efficiency as a consequence. For the throttled case, the flow losses include pumping losses so for this case, the BTE is lower and decreases more strongly with engine speed. The increasing hydrogen flow can be seen in the λ curves for the throttled case in Figs. 6 and 7, which show a decreasing air-to-fuel equivalence ratio with increasing engine speed. For the WOT case the air-to-fuel equivalence ratio is more or less constant: the air flow also increases due to an increasing volumetric efficiency. The increasing hydrogen flow results in a second effect, of an increasing air displacement (i.e. air flow decreases) and thus lower flow losses. However, the net air flow increases with engine speed so the intake flow losses increase with engine speed.

In Figs. 6 and 7, the air to fuel equivalence ratio for the WOT case is always higher than 2, as a result NO_X emissions are very low. All these points can thus be part of a practical load control strategy.

With a throttle position at 50%, it was not possible to reach 80 Nm with hydrogen. Instead, for the throttled hydrogen case, the TP was set so that a stoichiometric mixture was obtained. In that case, NO_X emissions can be treated in a TWC with high conversion efficiencies.

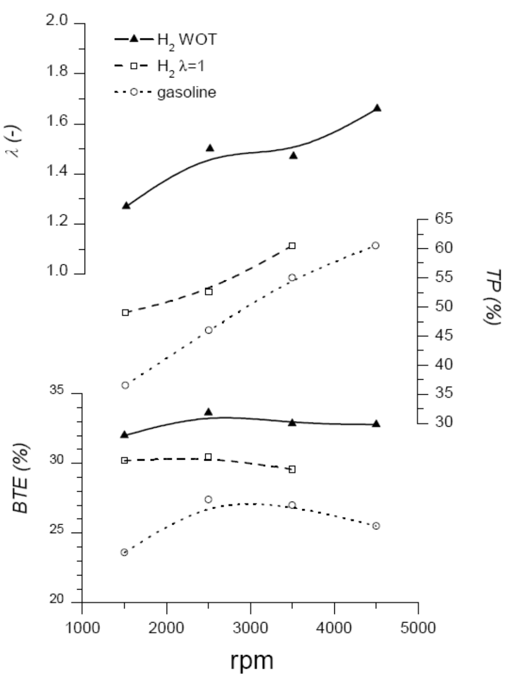

Figure 8 Brake thermal efficiency and respective throttle position or air-to-fuel equivalence ratio as a function of engine speed, for a fixed brake torque of 80 Nm

First, comparing Figs. 6, 7 and 8, the efficiencies of both gasoline and hydrogen can be seen to increase as the delivered torque increases. The explanation differs slightly for gasoline and hydrogen. In the three cases, as a result of the increasing torque, the mechanical efficiency increases strongly. For gasoline, the flow losses across the throttle valve increase because of the larger flow, although this is slightly compensated by a larger TP. The increase in mechanical efficiency is clearly the dominating effect. In the case of hydrogen, the flow losses decrease because of a smaller air flow since more air is displaced by hydrogen as a result of the richer mixture. This also leads to a decreased influence of engine speed on the hydrogen BTEs: from the figures it can be seen that as the load increases, the BTE decreases less strongly with engine speed.

Secondly, Fig. 8 shows that hydrogen WOT measurements at 80 Nm (for all engine speeds) have an air-to-fuel equivalence ratio between 1 and 2. This is below the ‘threshold’ equivalence ratio, taking a NO_X emission of 100 ppm as the threshold. As explained above, for an equivalence ratio between 1 and 2 it is impossible to reduce the NO_X emissions with sufficient efficiency using a TWC since the exhaust oxygen concentration is too high. As a consequence, these points are useless for an automotive application. The efficiency penalty caused by the NO_X boundary condition can be seen by comparing the H_2 WOT BTE curve to the H_2 $\lambda = 1$ BTE curve, showing a relative decrease in BTE of 5 to 10%.

MODELLING OF HYDROGEN COMBUSTION

An engine simulation code based on a two-zone thermodynamic model is developed for hydrogen spark-ignition engines, the GUEST code (Ghent University Engine Simulation Tool) [9]. This code is a compromise between non-predictive zero-dimensional models (type Wiebe law) and complex multidimensional models (type CFD) and is best situated for evaluating engine performance and for predicting the influence of engine settings.

The turbulent combustion model is based on the mass burning rate. For this the rate of entrainment of unburned gas into the (spherical) flame front is calculated from the turbulent entrainment velocity. The mass entrained into the flame front is then supposed to burn with a rate proportional to the amount of entrained burned gas, with a time constant.

Different turbulent burning velocity models were selected for calculation of the turbulent entrainment velocity (Danköhler, Gülder, Leeds, Fractals, Zimont, Peters). All these models need stretched laminar burning velocity data of the air/fuel/residual mixture at the instantaneous pressure and temperature.

The following laminar burning velocity correlation has been determined, based on measurements of cellular flames [10]:

$$u_n(\phi, p, T, f) = u_{n0}(\phi)(T/T_0)^{\alpha T}(p/p_0)^{\beta p}(1-\gamma f)$$

Here, the reference conditions T_0 and p_0 are 365K and 5 bar respectively. The influence of the equivalence ratio at these reference conditions is embodied in u_{n0} and is estimated at:

$$u_{n0} = -4.77\,\phi^3 + 8.65\,\phi^2 - 0.394\,\phi - 0.296$$

The values for α_T, β_p and γ are the following:

- The temperature exponent α_T has a mean value of 1.232.
- The pressure exponent β_p is dependent on the equivalence ratio,
  $\phi < 0.6$:
  $\beta_p = 2.90\,\phi^3 - 6.69\,\phi^2 + 5.06\,\phi - 1.16$

$\phi \geq 0.6$:
$\beta_p = 0.0246\ \phi + 0.0781$

- The parameter γ expressing the effect of residual gases, is given by:
$$\gamma = 2.715 - 0.5\ \phi$$

Next the model was calibrated with series of measurements on a carburetted hydrogen fuelled CFR engine.

The calibration sets the coefficients in the heat transfer model, the flame development model and the turbulent burning velocity model. Once the code has been calibrated for a single measurement, the calibration coefficients are kept constant and the models' predictive capability (simulations at other operating conditions) were be evaluated.

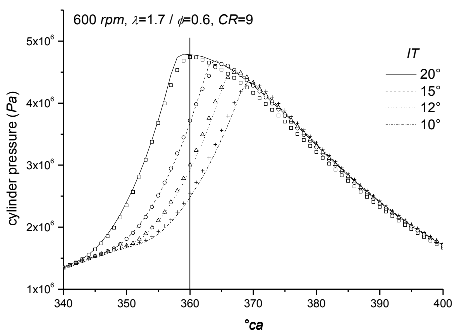

Figure 9 Cylinder pressure versus crank angle for varying ignition timing. Simulations using the 'Damköhler' model are shown in lines, the measured pressure traces are shown in symbols. 600 rpm, $\phi = 0.6$ / $\lambda = 1.7$, CR = 9

As an example Fig. 9 shows the measured and simulated pressure traces for variable ignition timing and Fig. 10 for variable compression ratio both for the 'Damköhler' model. The pressure traces obtained with the other models are similar. It is clear that the correspondence between measurement and calculation is very good.

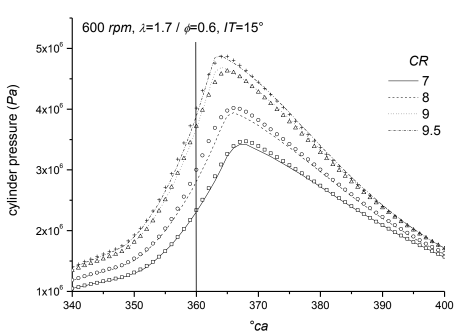

Figure 10 Cylinder pressure versus crank angle for varying compression ratio. Simulations using the 'Damköhler' model are shown in lines, the measured pressure traces are shown in symbols. 600 rpm, $\phi = 0.6$ / $\lambda = 1.7$, IT = 15°

The code and the correlation of the laminar burning velocity of hydrogen mixtures have been used by researchers at other research centers [11-13].

HEAT TRANSFER IN HYDROGEN ENGINES

The heat transfer process in an engine is an important boundary condition for the optimization of the engine and a good heat transfer model is needed in order to accurately simulate the NO_X emissions, which are temperature dependent.

Several heat transfer models for internal combustion engines exist in the literature of which the models of Annand and Woschni are mostly used. They have however been developed for fossil-fuelled engines and they have been cited to be inaccurate for hydrogen engines. The shorter quenching distance of hydrogen leads to a thinner boundary layer, the higher flame speed causes an intensified convection and hydrogen has a higher thermal conductivity. So the maximum possible heat transfer in an engine operating on hydrogen is expected to be higher than in a hydrocarbon-fuelled engine.

The transfer rates and wall temperatures in a hydrogen fuelled CFR engine (and methane fuelled as reference) were measured with a commercially available heat flux sensor. Figure 11 very clearly demonstrates the differences in hydrogen heat flux (compared against methane) for different load conditions of the engine [14].

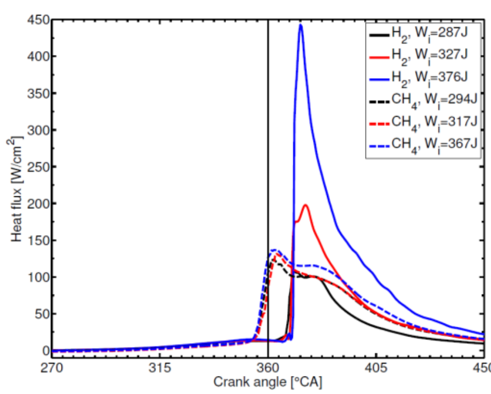

Figure 11 Varying the engine load by 23% results in a variation of 80% in the heat transfer of hydrogen and only 13% in that of methane due to the difference in load control

The results showed that the heat flux of hydrogen decreased substantially if the equivalence ratio (and therefore the engine load) was reduced. In contrast, the heat flux of methane did not decrease that strongly if the in-cylinder mass was reduced to attain lower engine loads. The peak in the heat flux was much higher for hydrogen compared to methane for the highest power output, but it was lower for the lowest power output. The mixture richness clearly has a large influence on the heat transfer process in contrast to the in-cylinder mass which was controlled by a throttle in the intake manifold. Total cycle heat losses have been calculated out of the measured heat flux traces. The trends in the heat flux losses were reflected in the indicated efficiency which was lower for hydrogen compared to methane for the highest power output, but it was higher for the lowest engine load. The extremely high heat losses generated by the combustion of a stoichiometric hydrogen-air mixture will have to be reduced to improve the engine's efficiency at high loads.

Other heat flux sensors will be tested in the near future to gain more confidence on the measurement results and different empirical heat transfer models will be compared and improved to capture the significantly varying heat transfer with changing engine load.

LINK TO METHANOL

Unfortunately hydrogen as a fuel for IC engines in the transport sector has disadvantages. Although it is an excellent fuel in the engine itself for emissions, efficiency and power output as shown before. But the bottle necks to go to an hydrogen-economy in the near future are the storage of the fuel on the vehicle and the building up of the infrastructure for hydrogen supply. One possibility could be that every house has its own production of hydrogen (electrolyzing water by green electricity).

Another solution is the use of sustainable liquid alcohols, such as ethanol and methanol. They are easily stored in a vehicle (as gasoline or diesel giving the same distance that can be driven) and are compatible with existing fuelling and distribution infrastructure.

The application of methanol and ethanol in engines is going on for a long time (for example methanol fuelled engines in California and Canada, ethanol in Brazil). China is now using coal-based methanol as a strategic transportation fuel to ensure the energy independence.

Methanol (and ethanol) have interesting properties with the potential to increase engine performances and efficiencies against gasoline engines (higher heat of vaporization, low stoichiometric air-to-fuel ratio, higher volumetric efficiency, higher power output and efficiency and lower propensity to engine knock) [15-16].

The disadvantages of methanol (and ethanol) as cold start performance and the aldehyde emissions are (nearly) solved by cold start strategies and devices and TWC after treatment.

But most of all, the use of methanol is an interesting approach to decarbonize the transport sector and to secure national energy supply. Methanol can be synthesized from different sources: from fossil fuels (but that is not the option for the future), from 1st and 2nd generation biomass, and from renewably produced hydrogen.

An interesting process for producing methanol in the future is the synthesis of hydrogen and captured CO_2 as is shown in Figure 12.

The CO_2 can be captured from the exhaust gas from power plants, but also from other industries and even from the atmosphere, as it is yet proven [17-18].

From this (possible) link to hydrogen the research on the use of the methanol in IC engines started recently at Ghent University. Two hydrogen engines (Volvo, Audi-NSU) are adapted to the use of methanol and a comparison on the performances for the two fuels are in execution.

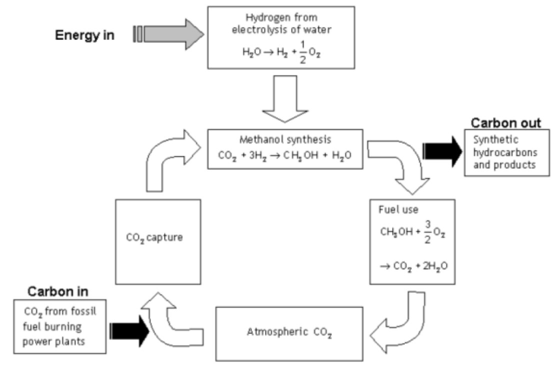

Figure 12 production methanol by captured CO_2

CONCLUSIONS

To run a hydrogen IC fuelled engine is very easy of one is not interested in the power output. (See first tests on the carburetted Valmet engine, lean conditions). But with the actual technologies (port-fuel injection, ECU system) higher power outputs and efficiencies than counterpart gasoline engines and at very low NO_X emissions can be obtained (supercharging lean conditions or stoichiometric with EGR regulation).

Ghent University developed a simulation model for the combustion process of hydrogen IC engines (GUEST code) and the correlation formula of the laminar flame speed of hydrogen is worldwide used. Updating this code is still going on, especially on the heat transfer process. It is shown that heat transfer in hydrogen engines is important for the efficiency of the engine and for the formation (or prediction) of NO_X.

From hydrogen methanol can be synthesized on a sustainable way (CO_2 capture). Methanol is an interesting engine fuel with advantages of storage on the vehicle and use of the existing infrastructure.

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APPENDIX: DEMONSTRATION PROJECTS

July 1994

Greenbus - De Lijn (Flemish public transportation company)

Partners: VCST - Hydrogen Systems NV (Belgium), UGent Laboratory for Transport Technology

Fuel: hydrogen

Sponsoring: Flemish Government and EU

Engine: MAN 12l inline 6cyl

Storage: metal hydrides

October 2000
ZEMBUS (Zero Emissions Bus), resulting from the 'Engine management system for hydrogen fuel' project

Partners: Hydrogen Systems NV (Belgium, project coordinator), Vialle NV (the Netherlands), Trivea Technologies International SA (Luxemburg), Betronic BV (the Netherlands), Continental Energy Systems (Belgium), UGent Laboratory for Transport Technology

Fuel: hydrogen

Sponsoring: European Union Brite-Euram III - CRAFT
Engine: DAF 8.6l inline 6cyl

Storage: liquid

March 2005:
CTPT bus: Clean Technology for Public Transport
Partners: UGent Institute of Sustainable Mobility of which our laboratory is a member and the Karel De Grote Hogeschool in Antwerp, Belgium
Fuel: 'hydrothane', a mixture of hydrogen and natural gas (currently 20/80%)
Sponsoring: the National Lottery of Belgium
Engine: MAN 12l inline 6cyl
Storage: compressed gaseous, 200 bar