Efficiency comparison of hydrogen fuelled IC engines with gasoline and methanol fuelled engines

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In: Conference proceedings of HYPOTHESIS IX, San José, Costa Rica, 12-15/12/2011

EFFICIENCY COMPARISON OF HYDROGEN FUELLED IC ENGINES WITH GASOLINE- AND METHANOL FUELLED ENGINES

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Abstract

One of the advantages of hydrogen as a fuel for IC engines is its high efficiency, certainly at low to mid load conditions, comparable to the efficiencies of diesel engines. Main reason for this is that the engine can run very lean, and that therefore the load can be regulated with the air-fuel ratio. This way the engine runs with wide open throttle for most of the time, avoiding throttle losses. For higher loads, the efficiency starts to decrease due to the increasing heat losses, which are more pronounced than for any other fuel.

Efficiency comparisons are made at two test rigs in the laboratory of Transport Technology. A four cylinder production engine and a one cylinder research engine.

Extensive experiments are done to compare the efficiencies of hydrogen and gasoline, as a function of engine speed and load for stoichiometric and lean conditions (for hydrogen), for variable inlet valve timing, for exhaust gas recirculation and supercharging (also for hydrogen). The most interesting results are shown and the differences for the efficiencies between the different fuels and for the different engine operation settings are explained.

Recently, experiments have started on methanol and the efficiencies are compared as well. Methanol is an interesting alternative fuel, which can be produced in a sustainable way from hydrogen.

Keywords: *hydrogen, efficiency, gasoline, methanol*

Introduction

There is no doubt that we have to find alternative fuels, to replace the derivatives from crude oil such as gasoline and diesel. There is also no doubt that the internal combustion engine will be the main propulsion technology for road transport for a long time.

Hydrogen gas is considered as such a possible fuel of the future, because it is present anywhere on the planet and its combustion does not produce CO2. However, it is always bonded to other chemical elements like oxygen (in water) or carbon (in hydrocarbons) and energy has to be added to release the hydrogen. Therefore, hydrogen is not an energy source but an energy carrier. A huge advantage is that hydrogen can be produced renewably with several methods.

Hydrogen is a good fuel if one considers the combustion in the engine. Several combustion characteristics of hydrogen enable a high engine efficiency over the entire operational range. First, a hydrogen-air mixture can ignite if the volumetric concentration of hydrogen is between 4% and 75% which is a much wider range compared to that of gasoline (1.4% to 7.6%). This allows to control the engine power output by varying the mixture richness like in a classical diesel engine, avoiding the need of a throttle in the intake manifold. Second, a hydrogen flame propagates very fast, resulting in a short combustion duration and consequently a high isochoric efficiency. Third, the theoretical thermal efficiency of the engine cycle can be higher compared to gasoline because of a higher attainable compression ratio and a higher specific heat ratio of a lean hydrogen-air mixture. Moreover, the engine hardly produces emissions. Only oxides of nitrogen (NOx) are emitted at high loads, which can be limited by a correct engine control.

The same properties that make hydrogen such a desirable fuel for internal combustion engines also bear responsibility for abnormal combustion events, including surface ignition, backfiring and auto ignition. Surface ignition is an undesired ignition induced by a hot spot in the combustion chamber. It can lead to backfire which is an uncontrolled ignition of the fuel mixture in the intake manifold. Controlling backfire was a main issue in the development of the first hydrogen engines. The easiest way to avoid it is to use direct injection so that there is no ignitable mixture in the intake manifold.

However, the main disadvantage of hydrogen is its low density. First, this complicates the storage of the fuel on a vehicle. The hydrogen has to be stored in a high pressure tank or as a liquid or as combination of the two to increase the drive range of the vehicle. Second, this limits the power output of an atmospheric hydrogen engine with port fuel injection, because a stoichiometric hydrogen-air mixture contains around 30 vol% of hydrogen, whereas this is only around 1.5 vol% in the case of a gasoline-air mixture. The power output can be increased by charging the intake air or by using direct injection.

Direct injection clearly seems the best option for a dedicated hydrogen engine, but the advantage of port fuel injection is that an existing gasoline (or natural gas) engine can be converted for hydrogen with minor changes. Therefore, this would be the best strategy for the modification of current vehicles. Strictly speaking, only the fuel tank, the injectors and the engine ECU would have to be adapted in this case.

Another possibility is the use of light alcoholics such as methanol and ethanol which have various properties that make them interesting as a transport fuel.

During the oil crises of the 1970s and 1980s, methanol was already the subject of many studies because it can be synthesized from a wide variety of renewable and alternative fossil-based feed stocks. More recently the focus has shifted towards ethanol produced from biomass. However, the biomass substitution for transport energy is limited by the arable land area and competition with food production, i.e. the biomass limit. Methanol production can transcend this biomass limit. It can be produced from coal or gas quite cheaply. For example, the wholesale price of coal-based methanol in China is about one third of that of gasoline. Moreover, methanol can be produced from renewable sources such as biomass, waste and even from atmospheric CO2. A number of researchers have proposed a sustainable methanol cycle in which methanol is synthesized from renewably produced hydrogen and CO2 captured from the air [1]. This closed carbon cycle is an active area of research.

Since methanol and ethanol are liquid fuels, only minor modifications are needed in order to use it in a combustion engine. Nevertheless, alcohol-resistant materials should be used for every part that comes in contact with the fuel. The lower volumetric energy content of methanol usually calls for a bigger fuel tank. Finally, the engine ECU should be recalibrated to ensure optimal engine operation on methanol. In fact, the changes to the engine are so small that methanol is mostly used in flex-fuel spark-ignition engines which are able to operate on any mixture of alcohol and gasoline. Over the years a lot of experience has been gained from extensive fleet trials, so that today most of the technical challenges involving methanol fuelled-vehicles have been solved.

Unlike many other alternative fuels, methanol and ethanol have the potential to increase engine performance and efficiency over that achievable with gasoline thanks to a variety of interesting properties. Their high heats of vaporization, combined with low stoichiometric air-to-fuel ratios, lead to high degrees of intake charge cooling as the fuel evaporates. This is especially true for engines with direct injection. The charge cooling not only leads to increased charge density, and thus higher volumetric efficiency, but also considerably reduces the propensity of the engine to knock.

At Ghent University, hydrogen has been investigated since 1991. The initial research was focused on the efficiency and power optimization of the engine with ultra low emissions of NOx as a constraint [2-3]. The research resulted in engines with a higher power output than that of an equivalent gasoline engine and higher efficiencies than that of an equivalent diesel engine. Over the years, a simulation tool has been developed to facilitate the optimization of a hydrogen engine [4]. Currently, the heat transfer process in the engine is investigated because it is an important boundary condition for the optimization of the engine. A good heat transfer model is needed in order to accurately simulate the NOx emissions, which are temperature dependent [5]. Extensive studies were carried out on the different load strategies of the hydrogen fuelled engines. Results are given for atmospheric and supercharged stoichiometric operation with EGR (and TWC), for lean conditions and for variable valve timing [6,7,8].

A detailed overview of the world wide published efficiencies and the properties of hydrogen that allow these high efficiencies are given in [9].

For methanol higher efficiencies are obtained to gasoline due to the high heat of vaporisation, the higher flame speed and the higher knock resistance. It permits the application of optimal values for spark advance, high compression ratios and opens opportunities for aggressive downsizing without the need for fuel enrichment at high loads. The overall power increase of using alcohol in turbocharged flex-fuel engines with direct injection can mount to 20% [10]. Using methanol in dedicated engines with high compression ratios makes it possible to achieve efficiency levels higher than that of current diesel engines [11].

This paper gives an overview of the efficiencies of hydrogen fuelled engines for different load strategies and some comparison is made with gasoline (as a reference fuel) and with methanol (as another possible alternative fuel).

Test engines

The results on two different engines are discussed further in this paper. The first engine is a production 4 cylinder gasoline Volvo engine. The second is a single cylinder research Audi engine.

The Volvo four cylinder sixteen valve gasoline engine with a total swept volume of 1783cc and a compression ratio of 10.3:1 was converted to tri-fuel operation by mounting an additional fuel rail supplying gaseous fuel to 8 (2 per cylinder) Teleflex GSI gas injectors, mounted on the intake manifold. The intake manifold was modified to avoid any damage if backfire would occur during the hydrogen measurements: a T-type branch pipe was mounted on the intake manifold with the 'straight ahead' branch closed by a foam plug and the other branch leading to the air filter – mass air flow (MAF) sensor – throttle valve assembly. Any pressure rise in the intake manifold due to the occurrence of backfire results in the foam plug being blown out instead of damaging other components such as the MAF sensor.
The engine has continuously variable valve timing (CVVT) on the intake camshaft, allowing up to 40 degrees crank angle (°ca) advance of the intake valve opening and closing time. A MoTeC M800 engine control unit is used to control ignition timing, start of injection, injection duration and intake valve timing.

The single cylinder Audi engine (400cc) is coupled with a DC motor. The operating speed is regulated between 1500 and 4500 rpm and the compression ratio is set to 10.2.
Two Teleflex GFI gas injectors are used for port fuel injection of gaseous fuels. Fuel (hydrogen) is supplied at 2 barg. Fuel mass flow is measured using a Bronkhorst Hi-Tec sensor.
The test bench is equipped with a Busch MM1102BP claw compressor to enable high supercharging pressures (up to 2 barg). Both rotors are fitted with a waterproof coating to stand up to the condensation water coming from the EGR-system. The compressor is driven by a 7 kW electric motor which is fed by an inverter, allowing varying the inlet pressure by adjusting the compressor speed.
It turned out to be necessary to install a choke valve to be able to create vacuum at the inlet of the compressor. This vacuum overcomes the pressure losses in the relatively long EGR-pipe, and causes a flow of exhaust gases. The surplus of energy to create this vacuum is provided by the compressor. In order to protect the compressor against excessive inlet temperatures, an EGR-cooler was installed to keep the inlet temperature around 50°C. At the end of the exhaust system, a TWC was installed. A TWC makes it possible to reduce the NOx emissions at stoichiometric operation. For a more efficient reduction, mixtures slightly rich of stoichiometric should be used so that the small amount of excess hydrogen can act as reducing agent.
A MoTeC M4Pro engine control unit is used to control ignition timing, start of injection and injection duration. A cold rated spark plug with a silver central electrode was used to minimize spark plug hot spots and catalytic reactions.

Experimental results

Volvo engine

As mentioned before this engine has a continuously variable valve timing (CVVT) on the intake camshaft. The influence of this valve timing on the efficiency of the engine has been examined. As an example table 1 show the results for an engine speed of 1500 rpm for hydrogen at stoichiometric conditions for different inlet valve timings. Each time the throttle position of the gas inlet valve is slightly adapted to obtain the same power output.

Measuring point	1	2	3	4	5	6	7
Lambda	2	1	1	1	1	1	1
Throttle position	100	31,5	31	30	29,5	30	28,5
Ignition timing (°ca BTDC)	10	-1	-1	-1	-1	-1	23
Inlet valve timing (°ca BTDC)	0	0	4	16	28	40	28
Brake thermal efficiency	34,1	27,6	27,6	27,6	27,8	28,1	20,8
Fuel	H_2	H_2	H_2	H_2	H_2	H_2	gasoline

Table 1: brake thermal efficiencies for different inlet valve timings

These experiments show that for an inlet valve change from 0 to 16°ca there is no change on the efficiency and only a very small increase for higher valve timing advances. From this indication it was decided to keep the inlet valve timing constant for all further measurements (all measurements then done at MBT timing on a fixed Intake Valve Opening advance of 4°ca BTDC). Table 1 gives also a comparison for hydrogen with wide open throttle WOT (λ = 2) and with gasoline (stoichiometric, λ = 1). The higher efficiency for hydrogen at WOT and the lower efficiency for gasoline will be discussed in the following set of measurements.

In following tests a comparison is made for hydrogen, gasoline and methanol for different load conditions (results shown for 20 and 80 Nm torque, corresponding with 1.41 and 5.64 bar bmep) and this for the whole speed range. For gasoline and methanol this is at stoichiometric conditions (λ = 1) and for hydrogen with WOT (and for the 80 Nm case for hydrogen also stoichiometric and throttled).

Fig. 1 shows the brake thermal efficiencies (BTE) and respective throttle positions or lambda values as a function of engine speed for a fixed brake torque of 20 Nm and Fig. 2 for a brake torque of 80 Nm.

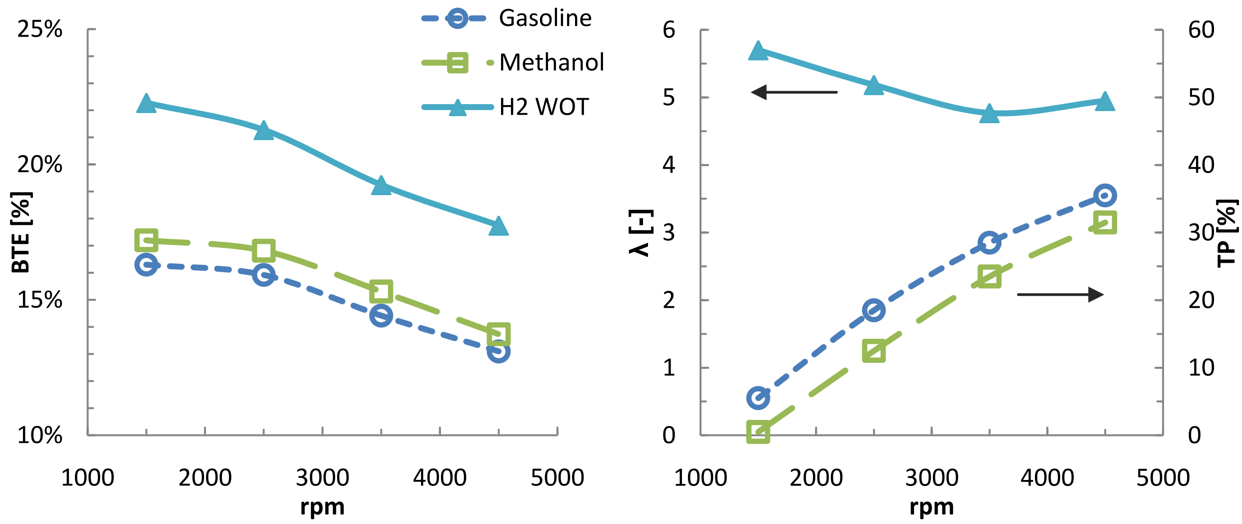

Figure 1 – Brake thermal efficiency and respective throttle position or air-to-fuel equivalence ratio as a function of engine speed, for a fixed brake torque of 20 Nm

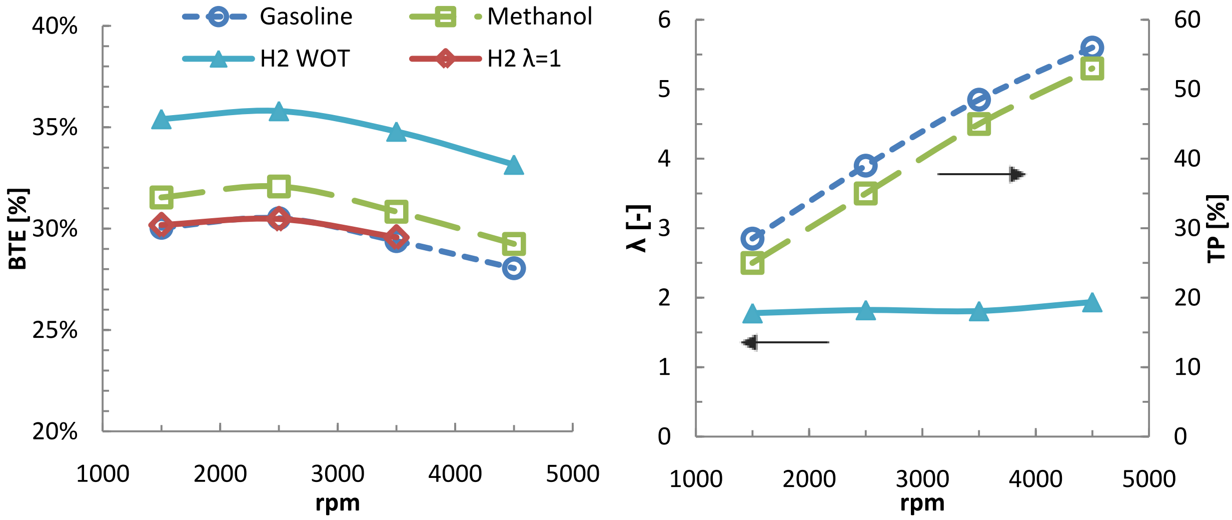

Figure 2 - Brake thermal efficiency and respective throttle position or air-to-fuel equivalence ratio as a function of engine speed, for a fixed brake torque of 80 Nm

Fig. 1 learns that for low load (20 Nm) for hydrogen (WOT, lean mixture) the efficiency is much higher then for gasoline or methanol (see also table 1).
This is mainly due that for gasoline and methanol the throttle has to be closed (throttle position TP = 0 is a closed gas inlet valve), resulting in high pumping losses. The lean mixtures of hydrogen results also in reduced cooling losses and higher theoretical efficiencies.

The efficiency on methanol is slightly higher than on gasoline. This is due to the higher burning velocity of methanol and the reduced losses due to lower airflow (less air is needed for the same power output, stoichiometric air fuel ratio is equal to 6.5 and for gasoline 14.7). This advantage is partly lost because of the smaller throttle opening (see Fig. 1). On the other hand the in-cylinder cooling losses are smaller for methanol (charge cooling by evaporation, reduced flame and exhaust temperatures).

At higher engine speed the efficiencies for all fuels decrease due to higher airflows and an increase in flow losses (in spite of the larger throttle openings for gasoline and methanol).

For a higher load (80 Nm, see Fig. 2) the efficiencies for the different fuels are significantly higher. Here also tests are done with stoichiometric, throttled hydrogen. The main reason for the higher efficiencies at higher torque (increased power output) is the increase of the mechanical efficiency, and for gasoline and methanol the more open throttle (despite a larger flow). For 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 efficiencies.

There is an important difference in the efficiency for hydrogen WOT and hydrogen stoichiometric ($\lambda = 1$) throttled. This is due of course to the closing of the throttle but also to higher heat losses. For low load conditions the heat transfer for hydrogen is simular to the other fuels but for high loads (and $\lambda = 1$) the heat transfer rise strongly [12].

Final remark is 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. For an air-to-fuel 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.

Audi engine

This section reports measurements on a single cylinder hydrogen PFI engine equipped with an exhaust gas recirculation (EGR) system and a supercharging set-up. The measurements were aimed at increasing the power output to gasoline engine levels or higher, while maximizing efficiency and minimizing emissions. Two strategies were tested: one using stoichiometric mixtures, with or without EGR, where a three way catalyst (TWC) was relied upon for aftertreatment of oxides of nitrogen (NO_x); and a second one using lean mixtures limiting engine-out NO_x emissions so that aftertreatment was not needed.

For lean-burn measurements were taken at 1500 and 1800 RPM and from 2000 to 3000 RPM in steps of 250 RPM. Starting from atmospheric operation, the supercharging pressure was then increased up to 1.5 barg in different steps.

Once the engine ran at lean-burn (λ = 2) with a chosen inlet pressure and engine speed, the NO_x emissions were measured. When supercharging, pressures and temperatures increase, so that leaner mixtures (than λ = 2) may be required in order for the NO_X emissions to stay at or below the maximum of 100 ppm. All lean-burn measurements were taken at the minimal spark advance for maximum brake torque (MBT timing).

For stoichiometric measurements at atmospheric operation and with supercharging at 0.5, 0.7 and 1 barg were taken at 1800, 2250 and 3000 RPM. As mentioned before, stoichiometric mixtures offer a higher power output and the possibility to use a conventional TWC but the occurrence of abnormal combustion phenomena rises due to higher cylinder peak temperatures and pressures. An excess of H_2 can be used as a reduction agent in the TWC. All experiments are done with a λ = 0.97 in order to decrease the NO_x emissions. Thus, "stoichiometric" operation is defined as the use of λ = 0.97.

Fig. 3 shows the net bmep and the brake thermal efficiency (BTE) versus engine speed, as a function of charging pressure for lean burn operation (at engine-out $NO_x \leq$ 100 ppm).

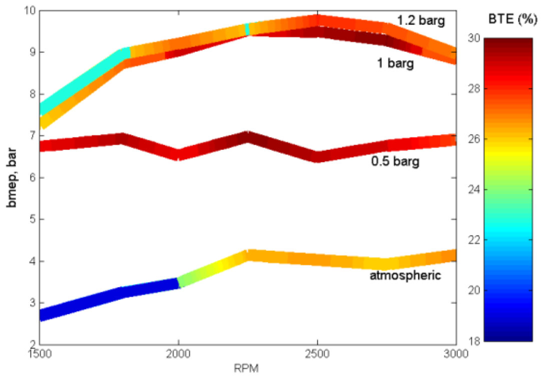

Figure 3. Net bmep and brake thermal efficiency (BTE) versus engine speed, as a function of charging pressure, for lean burn operation (engine-out $NO_x \leq$ 100 ppm)

Fig. 4 shows as well the net bmep and brake thermal efficiency (BTE) versus engine speed, as a function of charging pressure, for stoichiometric operation + TWC + EGR (and tailpipe $NO_x \leq 100$ ppm).

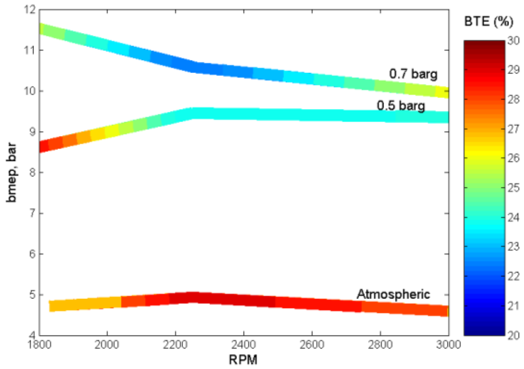

Figure 4. Net bmep and brake thermal efficiency (BTE) versus engine speed, as a function of charging pressure, for stoichiometric operation + TWC + EGR (tailpipe $NO_x \leq 100$ ppm)

These two figures give an overview for the whole speed and load range.

A detailed comparison for the two strategies at 2000 rpm and in function of the charging pressure is given in Fig. 5.

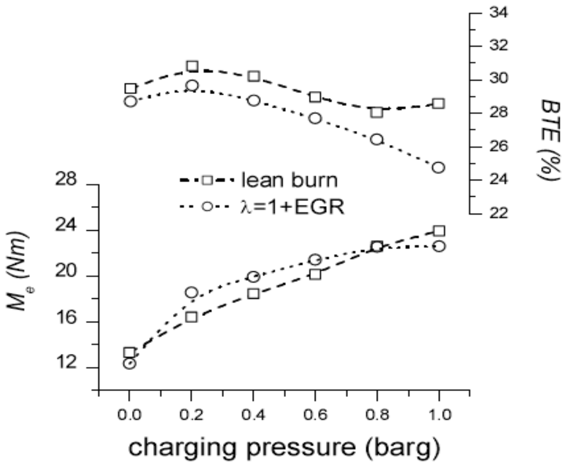

Figure 5: Comparison of brake torque and efficiency as a function of supercharging pressure.

Table 2 gives a comparison at maximum brake torque for the two strategies at 2000 and 3000 rpm.

	Lean burn		EGR	
Engine speed	M_e (Nm)	η_e (%)	M_e (Nm)	η_e (%)
2000 rpm	23.9	28.6	22.6	24.7
3000 rpm	25.6	27.7	21.2	22.5

Table 2. Comparison of maximum brake torque and corresponding efficiency when supercharging, between lean burn operation and stoichiometric + EGR operation, at the NO_x threshold, for 2000 rpm and 3000 rpm.

Fig[s]. 3 and 4 learn that lean-burn supercharged operation offers the possibility to obtain similar and even higher power output than conventional gasoline engines, with higher efficiencies and without any need for aftertreatment. Supercharging at more than 1.2 barg is not effective anymore due to the high required compressor power and the fact that a leaner mixture had to be used in order to control NO_x-formation. Between 0.5 barg and 1 barg the highest brake thermal efficiencies are obtained, up to 30%. The highest power output is reached for 1.2 barg resulting in a brake thermal efficiency around 28%. Lean-burn supercharged operation was investigated at WOT for all the loads and engine speeds, resulting in negligible pumping losses.

Stoichiometric operation offers the possibility to generate even more power when supercharging. The TWC can be activated if the exhaust temperature is above 500 °C but EGR is still needed to control the NO_x formation. At low charging pressures, more EGR is needed than at high charging pressures due to insufficient exhaust temperature (especially at low RPM). At high charging pressures and low RPM, no EGR is required. Increasing the engine speed at high pressure on the other hand necessitates the use of EGR to stay below the tailpipe NO_x limit. The overall obtained brake thermal efficiency is lower than for supercharged lean-burn operation at a similar net power output. Higher power outputs could be reached if the engine withstands the high cylinder pressures, but results in lower efficiencies.

The higher efficiency for the lean burn strategy against the stoichiometric (+ EGR) strategy for the whole load range is also seen in Fig. 5 at an engine speed of n = 2000 rpm. This is due, and this is for all engine speeds (see table 2), to the higher power needed for supercharging of the mixture of EGR and fresh charge at EGR operation, as a result of the higher temperature at the inlet of the compressor.

It is concluded that supercharged lean-burn operation is the best option to obtain similar or even higher power output than a gasoline engine, with the advantage of higher efficiencies, the absence of an EGR-circuit and no need for aftertreatment.

Conclusions

In this paper the efficiencies of two hydrogen fuelled IC engines are measured and discussed. For the first engine, a production four cylinder engine, a comparison is made with gasoline (as a reference fuel) and with methanol (an alternative fuel which can be produced from hydrogen on a sustainable way), for the whole speed and load range of the engine. The highest efficiencies are obtained for hydrogen, running lean and with wide open throttle. As expected and explained the efficiencies for methanol are slightly higher than for gasoline.

For the second engine, a one cylinder research engine, the influence of supercharging is examined, both for lean and stoichiometric hydrogen operation. For stoichiometric condition exhaust gas recirculation is used instead of throttling the engine. The measurements show that the highest efficiencies are obtained for the lean burn strategy.

Measurements on the Audi engine with methanol and gasoline are carried out at the moment and will be presented at the conference.

Definitions/abbreviations

AFR: air-fuel-ratio (kg air per kg of fuel)
BDC: bottom dead center
BTDC: before top dead center
BTE: brake thermal efficiency
CA: crank angle
CVVT: continuously variable valve timing
DI: direct injection
ECU: electronic control unit
EGR: exhaust gas recirculation
IT: ignition timing (°ca BTDC)
MBT: minimum spark advance for best torque
NO_x: oxides of nitrogen (NO and NO_2)
PFI: port fuel injection

TDC: top dead center
TP: throttle position
TWC: three way catalyst
WOT: wide open throttle
λ: air-to-fuel equivalence ratio (actual AFR / stoich. AFR)

Acknowledgements

The authors wish to thank the many master thesis students in the Transport Technology research group for their contribution to the present work. The research is funded by Ph. D. grants (SB-81139 and SB-091221) of the Institute for the Promotion of Innovation through Science and Technology in Flanders (IWT-Vlaanderen) and by Ph. D. grants (FWO09/ASP/030 and FWO11/ASP/056) of the Research Foundation-Flanders (FWO)). These financial supports are gratefully acknowledged.

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