VARIABLE COMPOSITION HYDROGEN/NATURAL GAS MIXTURES FOR INCREASED ENGINE EFFICIENCY AND DECREASED EMMISSIONS

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ABSTRACT

Adding hydrogen to natural gas extends the lean limit of combustion and extremely low emission levels can be obtained. Even the equivalent zero emission vehicle (EZEV) requirements can be reached. The emissions reduction is especially important at light engine loads.

In this study tests have been carried out with natural gas, pure hydrogen and different blends of these two fuels. The fuel supply system provides natural gas/hydrogen mixtures in variable proportion, regulated independently of the engine operating condition.

The influence of the fuel composition on the engine characteristics and on the exhaust emissions has been examined, mainly but not exclusively for 10 and 20 % hydrogen addition.

It is shown that to obtain maximum engine efficiency for the whole load range while taking low exhaust emissions into account, the mixture composition should be varied with respect to engine load.

1. INTRODUCTION

Hydrogen-enriched natural gas has been the subject of several research projects. These mixtures of natural gas and hydrogen are commonly named 'Hythane' (a registered trademark of Hydrogen Consultants Inc.).

Main point of interest is that with the addition of hydrogen, the lean limit of natural gas operation can be extended, without going into the lean misfire region. This results in low and even extremely low NO_x levels with only a slight increase in hydrocarbons. The low exhaust emission levels are obtained without emission control equipment (without a catalytic convertor). The CO and CO_2 values are lower for any gaseous fuel compared to gasoline, and for hydrogen no CO or CO_2 is formed at all (from the fuel itself).

Although hydrogen is an alternative fuel with very clean burning characteristics, a high flame propagation speed and wide flammability limits, it also has disadvantages. The complexity and weight of hydrogen storage, the loss of power associated with the use of pure hydrogen and the backfire phenomenon are the most important ones. The addition of natural gas to hydrogen (also hythane, but with a high percentage of hydrogen) can solve the backfire problem. In many cases backfire restricts the operating region of the air-fuel mixture on the 'rich' side. With natural gas addition stoichiometric mixtures can be run without any other precautions.

The proposition of an Equivalent Zero Emission Vehicle (EZEV) at 10 % of the 1997 ULEV (ultra low emission vehicle) requirements by the California Air Resources Board (CARB) has encouraged the research in lean burn hydrogen or hythane spark ignited engines.

In the literature [1 - 9] similar trends are found using hythane blends. When comparing the results in detail, one has to keep in mind that the composition of natural gas can be quite different (for different countries). The methane content can change from 90 to 98 %, with sometimes high nitrogen concentrations (1 to 8 %). This will of course influence the experimental results.

2. DESCRIPTION OF THE TEST RIG

2.1. Engine

A Crusader T7400 spark ignited engine (based on the GM 454 engine, best known as the Chevrolet Big Block) was adapted for gaseous fuels.

The engine specifications are :

- 8 cylinders in V
- bore : 107.95 mm
- stroke : 101.60 mm
- swept volume : 7.4 l (454 in^3)
- compression ratio : 8.5:1
- engine speed : 1000 - 4500 rpm
- ignition sequence : 18436572.

The engine is connected to a water (Froude) brake.

The gas (natural gas, hydrogen or hythane) is mixed with the air in a gas carburettor (see 2.2). The venturi of the gas carburettor is slightly under-dimensioned, which causes a lower volumetric efficiency and some power loss. For the comparison of the different fuels, this has no importance.

The composition of the natural gas consists of 91 % methane, 6 % ethane, 1.5 % propane, 0.8 % nitrogen and smaller fractions of higher hydrocarbons. The ignition is done by a single-firing system (one spark for each cycle of 720°ca) which is necessary to avoid backfire when using hydrogen. The ignition timing is regulated by a special disc on the distributor.

2.2. The fuel supply system

The layout of the system is shown in Fig. 1. The fuel is delivered to the engine through a gas venturi, which is supplied with the fuel mixture at a slight overpressure. The richness is controlled using a control valve in the supply line.

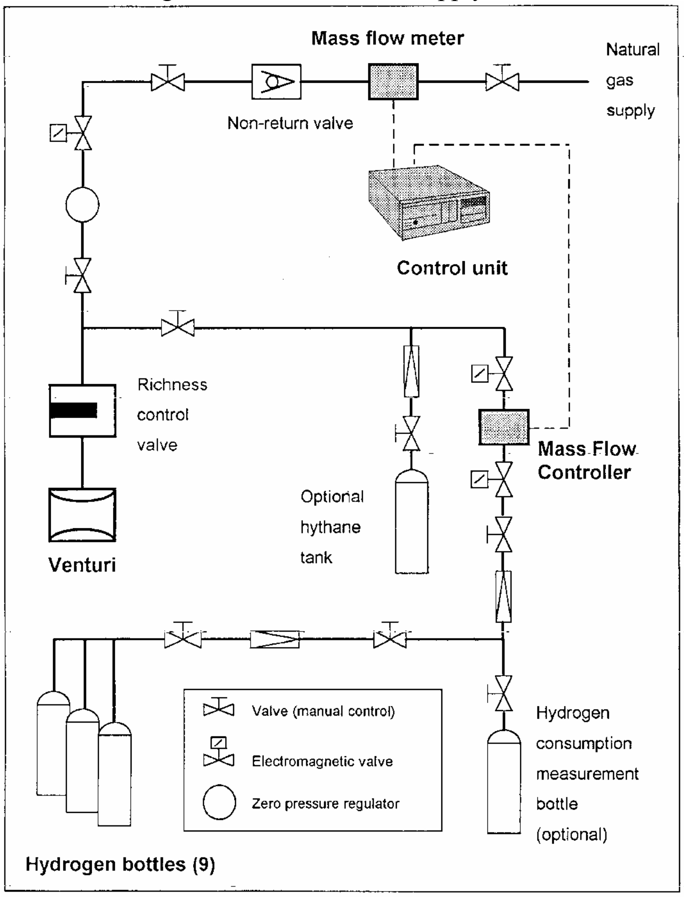

Fig. 1. The fuel preparation system

Both pure hydrogen and natural gas, as well as mixtures of these fuels can be used with this system. The hydrogen is stored in 9 steel bottles at a pressure of 200 bar; the natural gas is obtained from the city gas net at 30 mbar overpressure. Measurement of the fuel flow rate is made using mass flow meters in both of the supply lines.

If hythane is used, the hydrogen flow is controlled by a mass flow controller (the same device is used as mass flow meter for pure hydrogen). From the measured natural gas flow the necessary hydrogen flow is computed and supplied as input signal to the mass flow controller. This results in a constant hydrogen content, independent of engine speed and load. The hydrogen concentration is given in volume % (the term mass flow meter/controller only means that the measurement is automatically compensated for temperature and pressure changes, the reading is in Nm^3/h). Alternatively, hythane (or any other fuel) from a high pressure tank (200 bar) can be used for short runs: it contains a very limited amount of hythane and freezing problems can occur. By putting this tank on a scale fuel consumption can be measured.

The system described as above is only able to provide mixtures with up to 67 % hydrogen, due to limitations in the control unit. For higher hydrogen content the mass flow controller has to be regulated independently of the natural gas flow or opened completely. Doing the latter results in up to 79 % hydrogen. For even higher hydrogen fractions, the natural gas flow must be throttled.

2.3. Apparatus

The engine is fully equipped with the usual sensors. The measurement/control signals are read and controlled by a PLC system (Programmable Logic Controller). This system monitors engine speed, oil and coolant temperature, exhaust gas temperatures, etc. and shuts off the engine when necessary. With a Microsoft Excel worksheet all values are stored and can be made visible on a screen.

The exhaust temperatures and the exhaust gas composition can be measured at the exhaust of each cylinder and at the end of each bank (V engine). Two O_2 (λ)-sensors are installed at the common exhaust pipe of each bank, which allows an immediate value of the air-fuel ratio of each bank. The λ-sensors and the exhaust temperatures give the possibility to check if all cylinders behave the same.

The exhaust gas components are measured with the following methods of measurement : CO - CO_2 - NO - NO_2 (multor 610, non dispersive infra red); O_2 (servomex model OA 1100, paramagnetic), HC (Signal model 3000, flame ionization).

A high pressure transducer (type AVL QC 32) is located in one cylinder head (mounted flush with the combustion chamber wall of cylinder 1) and is used for the calculation of the heat release.

3. EXPERIMENTAL RESULTS AND DISCUSSION

All measurements were made at 3800 rpm, with the throttle wide open (WOT). These conditions are chosen because at that engine speed maximum power is reached. No other throttle positions were used because for hythane and hydrogen the best efficiency is achieved when regulating power through adjustment of the air excess ratio instead of the inlet pressure (except for the very low loads in which we are not interested in this study). This is not the case for natural gas because its combustion becomes so slow at lean mixtures that efficiency suffers and even misfire occurs below a certain equivalence ratio (this depends to great extent on the engine considered however). Therefore the lean limit as such has not been investigated here.

The measurements are analysed with respect to mean effective pressure instead of power, to make the results less dependent on engine size and speed. For a similar reason, exhaust emissions are represented in g/kWh instead of ppm or volume %: this way the dependence on effective engine power is avoided. More specifically, this also avoids the effect of the dilution with air of the noxious emissions in lean mixtures.

3.1. Natural gas and hythane with 10 and 20 volume % hydrogen

3.1.1. Brake mean effective pressure, fuel consumption, thermal efficiency, volumetric efficiency and ignition advance. In a first set of measurements a comparison is made between natural gas and two blends of hythane (hythane with 10 and 20 volume % hydrogen respectively). Each time, spark timing was optimised for maximum power (MBT).

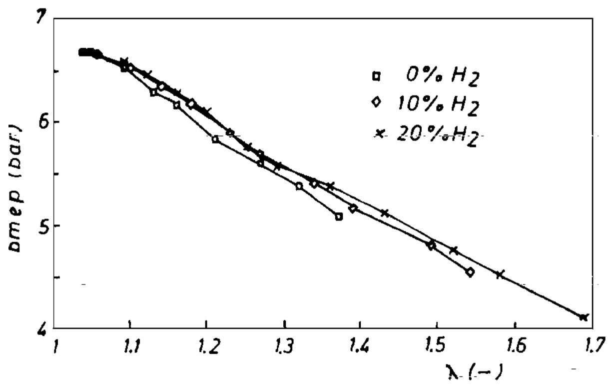

Fig. 2. The influence of the air excess factor on bmep

Fig. 2 shows how the air excess factor ϕ or air-fuel ratio λ ($\lambda = 1/\phi$) influences bmep. As can be seen clearly, the different fuel mixtures give very similar results: the only major difference is the ability of hythane to run leaner, the more so the higher the hydrogen content. The λ range is also determined by the fuel supply system: on the rich side the supply pressure of the natural gas is a limiting factor. For pure natural gas the lean limit gives a large reduction in engine power (no significant benefit is to be expected from such lean running however : low efficiency and large hydrocarbon emissions).

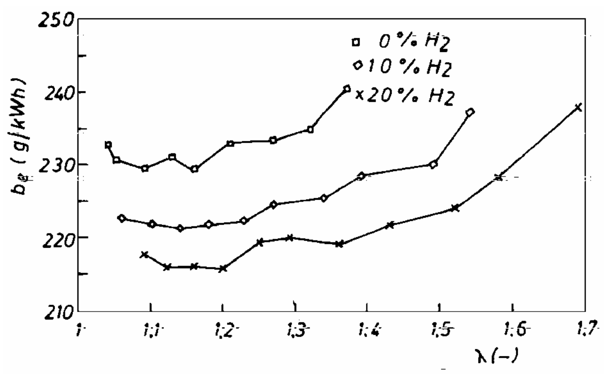

Fig. 3. The influence of the air excess factor on the specific fuel consumption

The specific fuel consumption b_e (expressed in g/kWh) for the same conditions is shown in Fig. 3. A λ-value between 1.1 and 1.2 gives best efficiency for each of the fuels. Because of the very different lower heat values of combustion (for hydrogen 120 MJ/kg and for methane 50 MJ/kg), efficiency comparisons should be made using Fig. 4. This shows that a hydrogen addition of 10 % increases efficiency moderately, whereas 20 % hydrogen gives no significant extra benefit (for the same λ-value). This result is in agreement with Bell [9], who also found a significantly higher efficiency when adding 10 % hydrogen, and no further improvement when going to 15 % hydrogen.

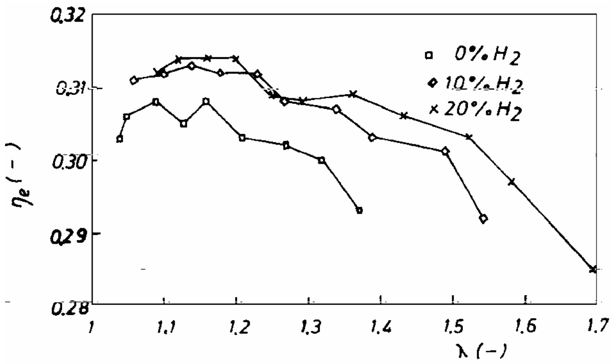

Fig. 4. The influence of the air excess factor on the efficiency

3.1.2. Emissions. The hydrocarbon (UHC) and the NO_x emissions as functions of the air excess factor are shown in Fig. 5 and 6. Minimum hydrocarbon and maximum NO_x emissions are found for an air excess factor of about 1.1. This result is in agreement with the literature, and holds for any fuel (see for example Heywood [14] for gasoline and Bell [9] for hythane).
For leaner mixtures the combustion temperature is lower because of the lower heat of combustion available in the mixture, which reduces NO_x, and for richer mixtures less oxygen is available for the formation of NO_x. The minimum in unburned hydrocarbon emissions in this range is caused by the impossibility of complete combustion on the rich side (local/global lack of oxygen) and on the lean side the flame extinguishes further from the walls resulting in more unburned emissions from walls and crevices. For even leaner mixtures the flame may be unable to travel through the whole combustion chamber and in some cases the spark may be unable to ignite the mixture at all.

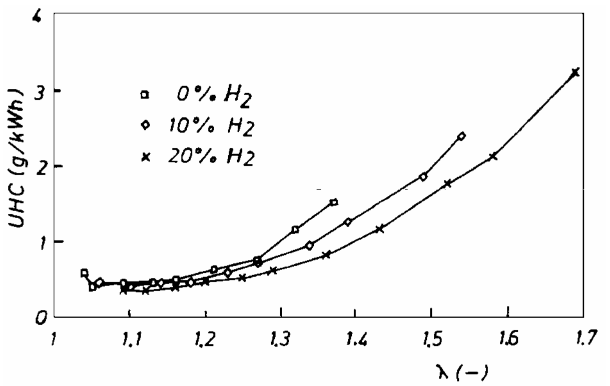

Fig. 5. The hydrocarbon emissions as a function of the air excess factor

For comparison at constant bmep, Fig. 7 and 8 are given. They show that 'at the same bmep' hydrogen addition increases NO_x and decreases unburned hydrocarbons, irrespective of bmep level. This result seems to disagree with results from the literature, which claim lower NO_x emissions when adding hydrogen. On inspection however, it becomes clear they compare natural gas with hythane with the latter running leaner

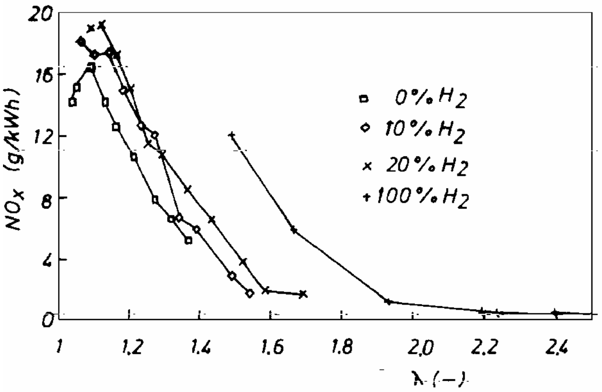

Fig. 6. The NO_x emission

(see for example Bell [9]), or with a reduced spark advance (see for example Raman [8]). Both these measures have the disadvantage of increasing unburned hydrocarbon emissions as well as decreasing efficiency. Also, the NO_x emissions can be reduced by retarding the spark from MBT for any fuel and therefore also for pure natural gas.

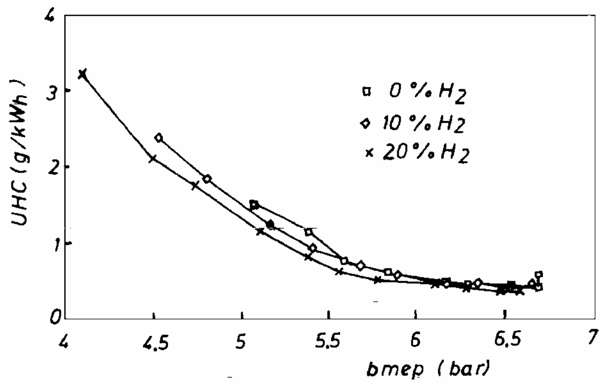

Fig. 7. The hydrocarbon emissions and bmep

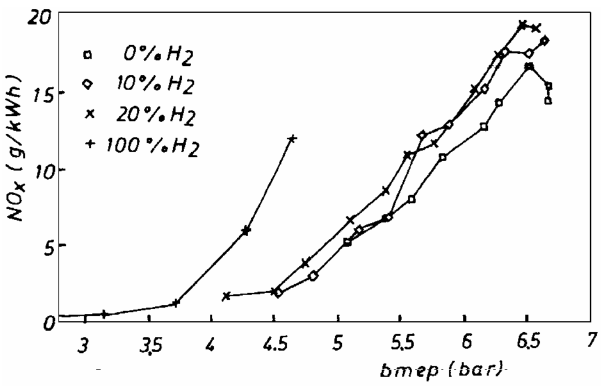

Fig. 8. The NO_x emission and bmep

3.2. Hythane with 50, 67, 84 volume % hydrogen and pure hydrogen

In a second set of tests a preliminary investigation was made on the use of higher hydrogen fractions. Due to time constraints (high hydrogen consumption) the ignition advance was not optimised for each condition in this set of measurements, but was kept constant at 20° before TDC, except for pure hydrogen where the optimal spark advance was used (which varies from 12° at $\lambda = 1.5$ to 22° at $\lambda = 3.5$).

For pure hydrogen, the engine could function reliably (without backfire) for $\lambda \geq 1.5$. Hythane with less than 79 % hydrogen never causes backfire nor knock in this engine, whatever the richness. During the measurements with 84 % hydrogen knock (without backfire) occurred for $\lambda = 1.06$. This suggests that a hydrogen content of 80 % or less guarantees safe operation of the engine, whatever the air excess factor.

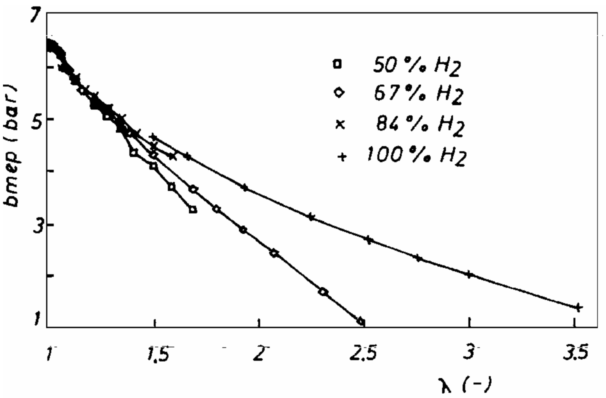

Fig. 9. The bmep for high hydrogen concentrations

In Fig. 9 the bmep achieved by pure hydrogen and hythane with 50, 67, and 84% hydrogen is shown. Remarkably all hythane mixtures give a very comparable output for λ between 1 and 1.3. This is in agreement with the results presented by Raman [8], who compared pure natural gas with hythane (30 % hydrogen) and found the same bmep at $\lambda = 1.3$ and only 2 % bmep loss at stoichiometry with hythane. The reason for this behaviour is that increasing the hydrogen content increases flame speed (increasing bmep) but decreases the volumetric energy content (for $\lambda < 1.43$) (decreasing bmep) (Raman [8]). In the λ region considered both effects compensate each other largely. For $\lambda > 1.43$ the volumetric energy content as well as the flame speed increase with higher hydrogen contents, which

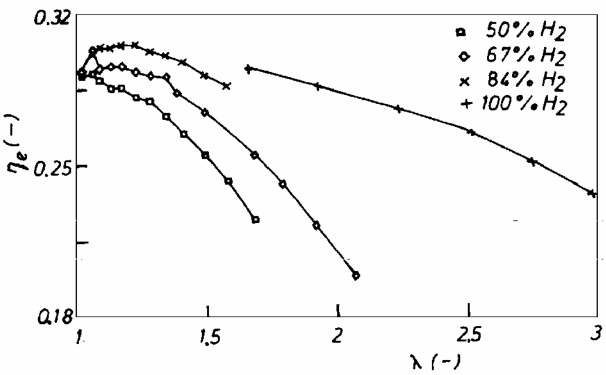

Fig. 10. The efficiency for high hydrogen concentrations

causes higher bmep for those mixtures. At these air excess ratios the bmep drops significantly though.

For strongly lean mixtures pure hydrogen gives the highest output, as it is the only fuel capable of fast combustion at these conditions.

4. THE LOW EMISSION POTENTIAL OF HYTHANE

Most - if not all - of the literature about the use of hythane concerns the possibility to achieve extremely low emissions. We will now investigate how this can be done.

Assuming that an ultra low emission vehicle (ULEV) with a fuel consumption equivalent to 10 l gasoline per 100 km (fuel density of 0.75 kg/l) during the test cycle has to be designed, the emissions performance of the engine can be estimated. If we take the average engine efficiency equal to 20 %, the ULEV limits (1.7 g CO per mile, 0.04 g NMOG per mile and 0.2 g NO_x per mile) convert to an average engine emission level of no more than 15 g CO per kWh, 0.35 g NMOG per kWh and 1.75 g NO_x per kWh. These average engine emission levels are roughly calculated as follows : 10 l/100 km x 0.75 kg/l x 0.20 (η_e) x 43.5 10^6 J/kg (H_u).(3600 10^3)$^{-1}$ kWh/J gives 0.181 kWh/km. For CO one finds 1.7 g/mile x 1.609 mile/km / 0.181 kWh/km ≅ 15 g/kWh.

Looking at the emission graphs, it seems that the hydrocarbon emissions are always too high. However, because the unburned hydrocarbons (UHC's) originate mainly from the fuel and more specifically from the natural gas fraction of the fuel, they are mostly methane. Because the ULEV limits concern non methane organic gases (NMOG; methane is not relevant with respect to the formation of tropospheric ozone), this improves the chances to comply with the limits. Unfortunately, the exhaust gas analysis equipment available for the experiments described here, didn't allow the measurement of NMOG directly.

The CO emissions from the tested engine are very low, as is the case for any gaseous fuel, and therefore doesn't cause mayor problems when trying to comply with the limits.

As can be seen from Fig. 6, the NO_x emissions are only low enough for very lean hydrogen and lean hythane mixtures. In the case of hythane, the higher the hydrogen fraction, the leaner the engine has to run and the lower the achievable bmep. At these conditions the engine produces a lot of UHC (see Fig. 5), which makes it extremely unlikely that the NMOG limit can be attained without exhaust gas treatment. The use of pure hydrogen gives no problem with NMOG, since only a negligible amount of UHC is emitted (no hydrocarbons in the fuel).

For lean mixtures, it is however possible to reduce CO and UHC (and therefore NMOG) emissions by using an oxidation catalyst. If we take into account that the NMOG fraction is reduced more strongly than the methane fraction, it is very probable that in this way the NMOG limits can be fulfilled.

Therefore the following strategy to run the engine can be proposed.

- At low loads, pure hydrogen is used ($\lambda > 2$), limiting NO_x and eliminating NMOG and CO emissions.
- At intermediate loads, hythane is used in such a way that NO_x remains low. The oxidation catalyst reduces CO and NMOG emissions to acceptable levels.
- At full load (nearly) pure natural gas is used, providing high maximum bmep.

Although this means high emissions at full load conditions, this is not a major concern with respect to the legal limits since these conditions don't occur in the test cycles used to measure emissions. Also, during everyday vehicle use the engine is mainly operated at part load reducing the need to optimise emissions at full load. If low emissions are to be reached at all operating conditions, a three way catalyst may be used with λ = 1 operation at high loads. This necessitates the use of a throttle to regulate power in the high load region and is unpractical without a drive by wire system. A less complicated solution would be to make the high load region simply unavailable for the driver (not an acceptable solution to the end user).

Such a system has the following advantages.

- High efficiency : WOT is used at all but the lowest loads, avoiding throttling losses. Fig. 4 and 10 show that near optimal efficiency is reached. Spark advance is set at MBT.
- Low emissions.
- Low knock sensitivity : at high loads pure natural gas with a high ON is used; at lower loads lean running avoids knock. This opens the possibility for high compression ratios, which increase efficiency.
- Backfire is avoided : since hydrogen is not used at $\lambda < 1.5$, backfire can be avoided without taking any special precautions.

The disadvantages are :

- weight and volume : two fuels must be stored.
- complexity and cost : a mixing system is necessary to control the hydrogen/natural gas ratio.

5. CONCLUSIONS

A fuel supply system was designed and implemented that provides the engine with a hydrogen/natural gas mixture in a variable proportion. The composition of this mixture can be set independently of engine operation. For future practical applications, minimal modifications should allow this variable mixture composition with respect to engine load for maximum efficiency and minimal pollution.

For hythane with low hydrogen content (up to 20 %) a limited improvement in emissions can be obtained. Because of the conflicting requirements for low hydrocarbons and low NO_x extremely low emissions are not possible without exhaust aftertreatment : to reduce hydrocarbon emissions λ must be less than 1.3, while for low NO_x λ must be at least 1.5.

For lower bmep, high efficiency can be achieved by increasing the hydrogen content and thus avoiding throttling losses. At the same time unburned hydrocarbon emissions are minimized, while (for lean mixtures) NO_x emissions stay limited. This means that for optimal results the composition of the fuel should depend on load.

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