Simulations of Multi Combustion Modes Hydrogen Engines for Heavy Duty Trucks

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Abstract

The paper presents the numerical study of a diesel direct injection heavy duty truck engine converted to hydrogen. The engine has a power turbine connected through a clutch and a continuously variable transmission to the crankshaft. The power turbine may be disconnected and by-passed when it is inefficient or inconvenient to use. The conversion is obtained by replacing the Diesel injector with a hydrogen injector and the glow plug with a jet ignition device. The hydrogen engine operates different modes of combustion depending on the relative phasing of the main injection and the jet ignition. The engine generally operates mostly in Diesel-like mode, with the most part of the main injection following the suitable creation in cylinder conditions by jet ignition. For medium-low loads, better efficienciy is obtained with the gasoline-like mode jet igniting the premixed homogeneous mixture at top dead centre. It’s permitted at higher loads or at very low loads for the excessive peak pressure or the mixture too lean to burn rapidly. The hydrogen engine has better efficiency than Diesel outputs and fuel conversion. Thanks to the larger rate of heat release, it has the opportunity to run closer to stoichiometry and the multi mode capabilities. The critical area for this engine development is found in the design of a hydrogen injector delivering the amount of fuel needed to the large volume cylinder within a Diesel-like injection time.

Keywords: heavy duty truck engine, hydrogen, alternative fuels, power turbine

1. Introduction

References [1] and [2] present a review of contemporary research on the hydrogen-fueled internal combustion engine. The emphasis of both is on light to medium-duty engine research. Reference [1] is mostly focused on port fuel injected, spark ignited hydrogen engines. Due to low volumetric efficiency and frequent pre-ignition combustion events, the power density of these premixed engines is diminished relative to the gasoline-fueled engines. The prospects for increasing efficiency, power density, and reducing emissions are indicated in the hybridization, multi-mode operating strategies, and advancements in ICE design and materials. Reference [2] offers a more recent and comprehensive overview of H2ICEs, but still mostly focusing on spark ignited engines. The hydrogen engines considered employing fuel injection, either in the intake manifold or directly into the combustion chamber. The paper mentions the use of igniting hydrogen jets in more advanced combustion concepts.

The same properties that make hydrogen become such a desirable fuel for internal combustion engines are also indicated to be responsible for abnormal combustion events associated with hydrogen [2]. In particular, the wide flammability limits, low

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required ignition energy and high flame speeds may result in undesired combustion phenomena generally summarized as combustion anomalies, including surface ignition and backfiring as well as auto-ignition. Backfiring is limited to external mixture formation operation and can be successfully avoided with direct injection (DI) operation. Proper engine design can greatly reduce the occurrence of surface ignition. Auto-ignition is shown as a controversial topic, with a wide range of octane rating for hydrogen as fuel being reported.

The most common strategies considered are the hydrogen port-fuel injection and hydrogen direct injection [2]. Inherent disadvantages of port-fuel injection include a lower power density compared to gasoline engines as well as operational limitations due to the occurrence of combustion anomalies. The wide ignition limits allow hydrogen port-fuel injected engines to operate almost un-throttled, and therefore they are efficient, over the entire operating regime. The dependence of the only relevant emissions component in hydrogen operation, the nitrogen oxides NOx, is well documented. At fuel-to-air equivalence ratios, 4, of less than 0.5 (λ> 2), the engine operates without emissions of NOx, increasing the fuel-to-air equivalence ratio beyond this critical threshold results in a sharp increase of NOx emissions. Measures to increase the power density of port-fuel injected hydrogen operation mainly focus on charging strategies. Hydrogen direct injection opens up another array of variables for influencing the mixture formation and combustion process.

The power density in hydrogen DI operation is shown to increase considerably. The emissions' reduction provisions considered in [2] include engine internal measures like EGR and water injection and after-treatment concepts including 3-way catalysts as well as lean NOx traps. Direct injection of hydrogen is shown to be effective in improving the performances of gasoline-like H2ICEs both experimentally and computationally also in [3].

This paper focuses on the opportunities to convert a Diesel direct injection heavy duty truck engine to hydrogen, a subject marginally covered so far in the literature.

To make the hydrogen in an internal combustion engines, three properties of hydrogen have to be considered. Hydrogen has a very low density, therefore delivery of large quantities within time frames small enough may be an issue. Hydrogen has a wide ignition limits of mixtures with air, from λ= 0.14 to λ=10 homogeneous. It permits the load control throttle-less typical of the Diesel. Hydrogen has auto ignition temperatures in air of about 850 K which ismuch larger than the Diesel. It requires much higher in-cylinder temperatures or the assisted start of combustion to produce Diesel-like combustion processes. With reference to Diesel, hydrogen also has the advantage to be delivered into the combustion chamber in its gaseous form. When the hydrogen jets penetrate the combustion chamber, they will mix with the surrounding air and the hydrogen will be heated up. Thanks to the faster combustion properties of the hydrogen vs. the Diesel made up of various hydrocarbon vapors, there is no need to vaporize the liquid droplets, and the combustion may proceed much faster with hydrogen after the more troublesome start-up phase.

The idea of two compression ignition hydrogen engines ideas had been developed recently, one working with a pilot Diesel injection preceding the main hydrogen injection [4] and the other one working with hydrogen only, but requiring a surface ignition [5-6]. Computationally, it has been demonstrated that a dual fuel pilot Diesel and hydrogen main direct injection permits similar to the operation and efficiencies of Diesel in both full and part load [4]. Experimentally, it has been demonstrated that direct injection of hydrogen only by using today's best single fuel injectors may permit similar to Diesel efficiencies continuously operating a glow plug providing surface ignition of the hydrogen and air mixture [5-6].

The computational results proposed in [4] with a pilot Diesel preceding a main hydrogen injection have shown that as soon as the main injection of hydrogen occurs in an environment with temperatures high enough and within the same injection duration of the main diesel injection, then the fuel conversion efficiency - the ratio of brake power to the product of fuel flow rate by fuel lower heating value - can be made even higher both full and part load and thanks to the ability to run richer λ than the Diesel, both the power and torque outputs may also be increased.

In the surface system ignition of [5-6] working with a single fuel injection pulse, the hydrogen is first ignited following impingement on the continuously operated glow plug. This surface ignition occurs 0.4 ms after the injector needle opens, immediately after the impinging of the hydrogen on the glow plug and the mixing with the surrounding air. The combustion evolution then sharply raises the chamber pressure and temperature, and once better auto-ignition conditions are globally reached within the chamber, then all of the hydrogen jets emerging from the multi holes injector ignite just following the mixing with air [5-6]. Assuming that there is adequate glow plug surface temperature, combustion starts as soon as one of the hydrogen jets reaches the glow plug as also shown in [7]. Knock is avoided by the then quick combustion of the hydrogen which is already available in the chamber. After the combustion start-up phase, the injection of hydrogen in high temperatures gases produces a combustion rate basically controlled by the rate of injection for the hydrogen that then burns almost instantaneously following mixing with air and temperature heating up.

The benefits of the operation Diesel-like with a high octane or low cetane number fuel has been shown in the Westport High Pressure Direct Injection (HPDI) design since more than 2 decades [8-15]. With the Westport HPDI, Diesel direct injection heavy duty truck (HDT) engines are converted to run mostly with liquefied natural gas (LNG) and minimally with the Diesel just replacing the Diesel direct injector with the HPDI dual fuel injector delivering a pilot Diesel and a main LNG injection. The pilot Diesel usually accounts for 5% of the total fuel energy input, so the engine is run 95% of total fuel energy on LNG. The HPDI LNG engine works, efficiencies are close to the Diesel all over the range of speeds, loads and even better than the Diesel outputs mostly thanks to the better mixing and combustion of the LNG and the consequent opportunity to run richer than the Diesel with higher energy release rates [8-15].

In this paper, a HDT Diesel direct injection engine is converted to run hydrogen replacing the main chamber Diesel injector with a hydrogen injector and replacing the glow plug with a jet ignition pre-chamber [7, 16-25]. The jet ignition pre-chamber accommodates a second hydrogen injector and a spark or glow plug to control the start of combustion. The hydrogen assisted jet ignition (HAJI) is a system considered since two decades to ignite premixed mixtures - homogeneous or stratified - much more efficiently than with a normal spark plug [7, 16-25]. In these devices, the hydrogen is injected in the pre-chamber and after it is fully mixed with air, a spark plug starts the combustion [16-25]. Alternatively, [7] the hydrogen enters the jet ignition pre chamber, and after impinging over the hot glow plug and mixing with air, it is heated up and it starts burning. In this case, as soon as the combustion evolves, the hydrogen mixes with air, heats up and burns even far from the glow plug. The high pressure and temperature of the combusting gases within the pre chamber produces the multiple jets entering the main chamber. These jets are made of hot combusting gases, fresh hydrogen and air.

Traditionally, [16-25] the igniting jets enter the main chamber where the fuel previously injected is already mixed with the air, homogeneously or stratified. This produces a gasoline-like operation. Alternatively, as it is proposed here, these jets are followed by the main injection hydrogen from the multi holes nozzles of the main chamber injector. This replaces the first phase of combustion build up designed with the pilot Diesel in the HPDI engine [8-15]. At this stage, the following main combustion event is mostly controlled by the rate of injection, as shown in [8-15] with the main LNG or in [5-6] with the main hydrogen. This produces a Diesel-like operation. If the main injection occurs before or after the jet ignition, before the quantity of fuel injected will burn mostly premixed, while after the quantity of fuel injected will burn mostly diffusion controlled. This will produce a mixed Diesel/gasoline-like operation. These modes of operations are numerically investigated here. The feasibility of the combustion processes described in the paper is not purely hypothetical, but relies on the prior extensive applications [4-25].

2. Engine

The engine is a 12.8 litre in-line six cylinder turbo charged directly injected HDT Diesel engine converted to hydrogen. The engine has a power turbine. Two turbines are used in series; one waste gated driving the upstream compressor, and the other one to produce additional power to the driveline. In traditional turbo compounding applications, the turbo compound includes a downstream axial flow power turbine plus the speed reduction gears, the fluid coupling and the final gear reduction to crankshaft to supplement crankshaft power. In the engine considered here, the traditional gear train coupling of the power turbine is replaced by a continuously variable transmission (CVT). A by-pass of the power turbine is also included. This permits to narrow the range of speeds where the turbine operates producing more power than the power loss for back pressures while permitting temperatures to the downstream after treatment system high enough. The power turbine is disengaged and by-passed when it's inefficient inconvenient to use. The Diesel engine without the power turbine is a production engine. The addition of the power turbine is only computed, as it is the use of the hydrogen fuel.

The engine is manufactured by a European company and the experiments have been performed in a European engine lab. The experimental map includes different speed RPM and load BMEP the following parameters: power, quantity of fuel injected during the main injection, start of the main injection, duration of main injection, delay time pre injection, quantity of fuel injected during the pre injection, duration of the pre injection, delay time pilot injection, quantity of fuel injected during the pilot injection, duration of the pilot injection, delay time post injection, quantity of fuel injected during the post injection, duration of the post injection, pressure of the injection rail, maximum pressure within the cylinder, boost pressure, pumping, temperature out of compressor, temperature intake manifold, temperature exhaust gas recirculation out, pressure intake, pressure compressor in, pressure compressor out, temperature turbine in, pressure turbine in, rpm turbo, EGR stroke, BSFC, fuel flow, air flow, Smoke, Combustion noise, CO, Soot, NOx, HC, O2 exhaust, volumetric efficiency, lambda, oil pressure, oil temperature, water temperature, total quantity of fuel injected. These data enable to properly tune all the model parameters for the baseline Diesel engine, and then forecast the performances of hydrogen fuel as well as the power turbine could be reasonable and accurate.

The combustion chamber is a traditional bowl-in-piston combustion chamber with central direct injector and glow plug in the Diesel version. In the hydrogen version, the glow plug is replaced by the jet ignition device. A picture of a Diesel direct injection combustion chamber is proposed in Figure 1 (from [26]). The engine geometry (baseline) is presented in [27]. The baseline Diesel engine has a target performance of 2,600 Nm torque 1,000-1,450 rpm, and of about 400 kW of power 1,450-1,900 rpm, with BSFC values around 190 g/kWh, corresponding to brake fuel conversion efficiencies of 44%. The engine is compliant with EURO-5 emission standards.

The additional cooling of the exhaust gases through the power turbine in principle reduce the effectiveness of the exhaust after treatment systems in principle. However, using the power turbine for high loads and speeds, this should not be an issue, because temperatures for these loads and speeds will be much higher than those of medium to low loads and speeds. Operating with hydrogen, only the NOx system is a concern and then the other after treatment devices are removed. With hydrogen, the temperatures of the power turbine are larger, with the subsequent opportunity to further increase the exhaust energy recovered.

Fig. 1 Typical HDT Diesel direct injection combustion chamber [26]

3. Combustion modeling in engine performance codes

Engine performance simulations have been performed with the GT-POWER code [28]. When the jet injection and the spark discharge precede the main injection, the main injection occurs in hot reacting gases. This operation is similar to the Westport HPDI concept which a small Diesel pilot injection precedes a main LNG injection. When the jet ignition and the spark discharge follow the main injection, the engine operates as a traditional gasoline engine with a high energy ignition. The experiences made both numerically and experimentally with the HAJI (hydrogen assisted jet ignition) support this operation.When part of the main injection occurs before the spark discharge, there is a mixed, premixed and diffusion combustion that does not fails within the two prior schemes which has not been experimentally investigated so far. Some 3D CFD simulations with detailed chemistry have been started to further study for this operation, but experimental data are certainly needed to support the modeling activities and to bring to a full understanding of the challenges and opportunities of the technique.

In theory, HCCI occurs without any igniting source. However, HCCI operation is very difficult to be achieved in practice with any fuel, and in particular with hydrogen, in a more than a few operating points in the speed and load map. As a matter of fact, HCCI has been studied since decades, but no production engine uses HCCI in present. The HCCI-like operation is obtained here having the jet ignition at top dead centre ensuring the start of combustion that otherwise will not occur on a wider range of loads and speeds. Therefore, this HCCI-like operation is actually a gasoline-like operation with a very fast combustion about top dead centre (TDC) in selected mid-low load points. At higher loads, this operation is impossible because the peak cylinder pressure rises too much. At very low loads, the ignition of a very lean mixture will be similarly difficult.

The combustion model is quite simple and all the equations used are presented below. Wiebe functions are used to model the Diesel-like as well as the gasoline-like operation. The mixed Diesel/gasoline-like operation is not modeled. The parameters of the Wiebe function for the Diesel-like combustion are assumed to be the same as the Diesel and the hydrogen operation. The parameters for the Diesel are experimentally measured, the parameters for the hydrogen are assumed to be the same as the Diesel for same percentage of maximum load and speed. This assumption is proved by the measurements done over 20+ years on the Diesel HDT engines converted to pilot Diesel and main LNG [8-15] in the HPDI design. Providing the in cylinder condition is made suitable for the following main fuel injection to burn diffusion controlled, then there are not too many differences between fuels with different cetane and octane numbers in general or hydrogen and LNG in particular.

Injection and combustion are linked together in a Diesel engine. Injection is prescribed in GT-POWER through an injection profile, while combustion is prescribed through a combustion profile. When the engine is experimentally tested, both the injection profiles and the combustion profile are those experimentally measured. For the combustion profile, rather than using the detailed profile, a Wiebe function approximation is better used to save data. Injection profiles and Wiebe function parameters are usually made dependent on speed and load. When direct experiments are not available, prior results on similar engines are used to predict the injection and combustion parameters to be used on a novel engine, and the expected accuracy will be related to the difference between the engine build tested and the engine under study.

For the engine operating Diesel-like with hydrogen, the assumption that the injection and combustion profiles are the same as the Diesel at same percentage of load and speed is both conservative and optimistic. It is conservative for the mixing of the fuel with the air, because the mixing of the hydrogen takes much less than the mixing of the Diesel, being the Diesel a mixture of complex liquid hydrocarbons requiring vaporization. However, The assumption is optimistic for what concerns the injection of the hydrogen, being the mass flow rate of the Diesel is quite difficult to be achieved with the extremely low density hydrogen.

The parameters of the Wiebe function for the gasoline-like combustion are guessed considering the experimental data collected during a much smaller number of experiments with jet ignition pre-chambers in different engine configurations [16-25]. However, the accuracy of the Wiede DI and SI functions for hydrogen is expected to be roughly the same.

3.1. Diesel-like operation – M1 mode

Injection occurs in the jet ignition pre-chamber before the main chamber fuel is injected and mixed with air. The engine operates Diesel-like. This mode of combustion assumes that the most part of the main chamber fuel is injected following the injection of the pre-chamber fuel and eventually of a small portion of the main chamber fuel to create proper conditions for the main chamber injection combustion to take place diffusion controlled. The fuel injected is roughly 5% pre-chamber and 95% main chamber for all the loads and speeds. This translates in an equivalent 5% pilot/pre and 95% main injection of the Diesel. The main injection combustion is modeled with the GT-POWER DI Wiebe model [28]. This model imposes the burn rate using a three-term Wiebe function (the superposition of three normal Wiebe curves). These Wiebe curves are approximate the "typical" shape of a DI compression ignition burn rate. The purpose of using three functions is to make it possible to model pre-ignition (the large initial spike) and larger tail. The injection profile does not influence the burn rate except if, at any instant, the specified cumulative combustion exceeds the specified injected fuel fraction. The inputs of the Wiebe equations are SOI = Start of Injection, ID = Ignition Delay, DP = Premix Duration, DM = Main Duration, DT = Tail Duration, FP = Premix Fraction, FT = Tail Fraction, EP = Premix Exponent, EM = Main Exponent, ET = Tail Exponent and CE = Fraction of Fuel Burned (also known as "Combustion Efficiency"). The calculated constants of these equations are FM = Main Fraction, WCP = Wiebe Premix Constant, WCM = Wiebe Main Constant and WCT = Wiebe Tail Constant given by equation below.

$$F_M = (1 - F_P - F_T) \tag{1}$$

$$\mathrm{WC}_P = \left[\frac{\mathrm{D}_P}{2.302^{1/(E_P+1)} - 0.1051^{1/(E_P+1)}}\right]^{-(E_P+1)} \tag{2}$$

$$\mathrm{WC}_M = \left[\frac{\mathrm{D}_M}{2.302^{1/(E_M+1)} - 0.1051^{1/(E_M+1)}}\right]^{-(E_M+1)} \tag{3}$$

$$\mathrm{WC}_T = \left[\frac{\mathrm{D}_T}{2.302^{1/(E_T+1)} - 0.1051^{1/(E_T+1)}}\right]^{-(E_T+1)} \tag{4}$$

With θ = Instantaneous Crank Angle, the burn rate is finally given by the equation below:

$$\begin{gathered} Combustion(\theta) = (CE)(F_P)\left[1 - e^{-(WC_P)(\theta - SOI - ID)^{E_P+1}}\right] \\ +(CE)(F_M)\left[1 - e^{-(WC_M)(\theta - SOI - ID)^{E_M+1}}\right] + (CE)(F_T)\left[1 - e^{-(WC_T)(\theta - SOI - ID)^{E_T+1}}\right] \end{gathered} \tag{5}$$

The cumulative burn rate is calculated, normalized to 1.0. Combustion starts at 0.0 (0.0% burned) and progresses to 1.0 (100% burned).

The Wiebe function parameters for the Diesel-like operation are obtained from measurements and they change with speed and load. Therefore they account for all the possible influences, including turbulence, not only changing the speed but also changing the load. 3D CFD simulations with detailed chemistry have been previously considered to study the injection and combustion evolution in a spark ignition engine equipped with a direct hydrogen fuel injector and a jet ignition pre chamber [29]. Considering the required hydrogen injector for the large bore cylinder is presently unavailable, simulations are not repeated here to study the influence that turbulence could have on the combustion evolution within the present engine, being a lack of mass flow much more penalizing the performances than a lack of turbulence.

In the diesel-like operation, the pre-chamber hydrogen injection precedes the main injection. It did not result in an indirect injection IDI type of combustion. The amount of fuel injected within the pre-chamber is small if compared to the main chamber injection (roughly 5%). In the IDI engines, all the fuel is injected in the pre chamber and not just the 5% of the total. The remaining 95% of the fuel injected in the main chamber burns almost completely diffusion controlled as in a Diesel DI the main injection fuel that follows the pilot or pre injected fuel. Therefore, the use of the DI combustion model is fully acceptable in present.

3.2. Gasoline-like operation – M3 mode

Injection occurs in the jet ignition pre-chamber after the main chamber fuel is injected and mixed with air. The engine operates gasoline-like. The main injection combustion is modeled with the GT-POWER SI Wiebe model [28]. This model can be used with any types of injection, but if fuel is injected directly into the cylinder, the start of injection must precede the start of combustion so that there is fuel in the cylinder to burn when combustion starts, and at any instant, the specified cumulative burned fuel fraction must not exceed the specified injected fuel fraction. The inputs of the Wiebe equations are AA = Anchor Angle; D = Duration; E = Wiebe Exponent ; CE = Fraction of Fuel Burned (also known as "Combustion Efficiency"); BM = Burned Fuel Percentage at Anchor Angle ; BS = Burned Fuel Percentage at Duration Start; BE = Burned Fuel Percentage at Duration End. The calculated constants of these equations are Burned Midpoint Constant BMC, Burned Start Constant BSC, Burned End Constant BEC, Wiebe Constant WC and Start of Combustion SOC given by equation below with θ the Instantaneous Crank Angle, BMC=-ln(1-BM), BSC=-ln(1-BS), BEC=-ln(1-BE):

$$\mathrm{WC} = \left[\frac{D}{BEC^{1/(E+1)} - BSC^{1/(E+1)}}\right]^{-(E+1)} \tag{6}$$

$$\mathrm{SOC} = AA - \frac{(D)(BMC)^{1/(E+1)}}{BEC^{1/(E+1)} - BSC^{1/(E+1)}} \tag{7}$$

$$Combustion(\theta) = (CE)\left[1 - e^{-(WC)(\theta - SOC)^{(E+1)}}\right] \tag{8}$$

The cumulative burn rate is calculated, normalized to 1.0. Combustion starts at 0.0 (0.0% burned) and progresses to 1.0 (100% burned).

Prior experiments and computations performed on hydrogen engines with jet ignition of a premixed mixture show that combustion occurs almost instantaneously with stoichiometric mixtures, and with burn rates much faster than traditional glow plug ignition with lean mixtures [16-25]. More than for stoichiometric operations, it’s quite difficult to be achieved knock-free in a high compression ratio engine (CR=18.5 in the specific), this model helps to reproduce the homogeneous charge compression ignition (HCCI) modes, which are very lean homogeneous mixtures to get self ignited and to burn almost instantaneously. When combustion profiles are not available experimentally, as it is the present case, the Wiebe function parameters are guessed from part experiences on different engines [16-25].

More than the pre-chamber injection, it is the pre-chamber ignition that controls the start of combustion within the engine. The pre-chamber injection must produce a homogeneous mixture within the pre-chamber at the time of the spark discharge. The pre-chamber is usually quite small, 5% of the total combustion chamber volume when the piston is at top dead centre, and the fuel-to-air mixture has to be close to stoichiometry at the time of spark [16-25].

The hydrogen assisted jet ignition concept has been studied since more than two decades to burn premixed mixtures of various fuels homogeneous as well as stratified. These mixtures were produced by carburettors, port fuel injection or direct injection [16-25]. Therefore, this gasoline-like combustion mode is the most widely studied with jet ignition pre-chambers so far[16-25].

Anchoring Angle (AA) and Duration (D) are determined in the model used by using experimental and computational results obtained on different engines. Ongoing 3D CFD and detailed chemistry simulations may certainly provide better estimates, considering the large bore, the bowl in piston combustion chamber and the large compression ratio never considered before.

3.3. Mixed Diesel/gasoline-like and HCCI-like operations – modes M2 and M4

In mixed Diesel/gasoline modes of operation, injection occurs in the jet ignition pre-chamber after part of the main chamber fuel is injected and mixed with air. The engine operates mixed Diesel-like/gasoline-like. Only the case of a small injection of main chamber fuel is considered in the model above. The novel design permits much more complicated strategies coupling premixed and diffusion combustions within the main chamber. Unfortunately, these strategies cannot be modeled with a Wiebe function approach. In the HCCI mode of operation, no injection occurs in the jet ignition pre-chamber and the engine operates HCCI-like self-igniting. It is supposed that the injection of the main chamber fuel is properly calibrated to achieve a start of combustion immediately following the top dead centre (TDC). Combustion then evolves almost instantaneously. Jet ignition helps to better control the process and make it more stable and repeatable.

4. Numerical results

Results for the baseline Diesel have been presented and discussed in [29]. Results are presented here for the engine fuelled with Diesel or hydrogen with or without the power turbine. Two different modes of operation are considered with the hydrogen fuel. The mode of Diesel-like diffusion combustion and the mode of HCCI/gasoline-like premixed combustion are considered. This latter mode of combustion is fairly optimistic, because it assumes that the premixed hydrogen would auto-ignite or ignite following jet ignition about top dead centre with a combustion event completed within 10 degrees of crank angle. All the results are computational. The Diesel model is set up with the support of experimental data but all the results proposed here are all numerical.

For the mode of Diesel-like diffusion combustion, the hydrogen is mostly injected once proper in-cylinder conditions are achieved, and combustion rapidly occurs following diffusion of the hydrogen in air. The rate of combustion is given by (5). The mode of HCCI/gasoline-like premixed combustion is obtained by introducing all the fuel within the in-cylinder first, and then having combustion started spontaneously (HCCI) or controlled by jet ignition about top dead centre. In both cases, the rate of combustion is given by (8) with a short 10-90% mass fraction burned and an anchor of the 50% mass fraction burned set to reproduce a quick combustion shortly after top dead centre. It is to be pointed out that this mode of combustion is not applicable to cases with small λ, because the in-cylinder pressure would rise too much. Similarly, operation with very large λ may also become troublesome even for the hydrogen. A minimum operational λ of 1.45 is arbitrarily selected for the Diesel. A minimum λ of 1.35 is considered for the hydrogen. The Diesel is a mixture of hydrocarbons injected in liquid phase, and this mixture has to vaporize before mixing with the air and then burning. The hydrogen, is a very simple gas, does not need vaporization. Besides, mixing and burning are much quicker and much cleaner. A λ about 1.45 produces similar peak in-cylinder pressure for the Diesel and the hydrogen around 180 bar.

For the operation with hydrogen and the mode of HCCI/gasoline-like premixed combustion, all the points producing a maximum in-cylinder pressure exceeding 180 bar are avoided. All the points requiring a λ exceeding a threshold value of 6 are also neglected. Auto-ignition or jet ignition of a premixed hydrogen air mixture with a rate of combustion with a λ higher than 6 are possible, but this option is not considered here just for the sake of simplicity.

Fig. 2 and 3 present the volumetric efficiency and the compressor pressure ratio vs. the engine speed and load for the Diesel engine operating without and with the power turbine. The maximum load BMEP, torque and power curves are also shown. With the power turbine, the engine generally works with a reduced compressor pressure ratio and with a reduced volumetric efficiency for the same BMEP. It's because the power turbine supplement the in-cylinder power supply when the power turbine is operational.

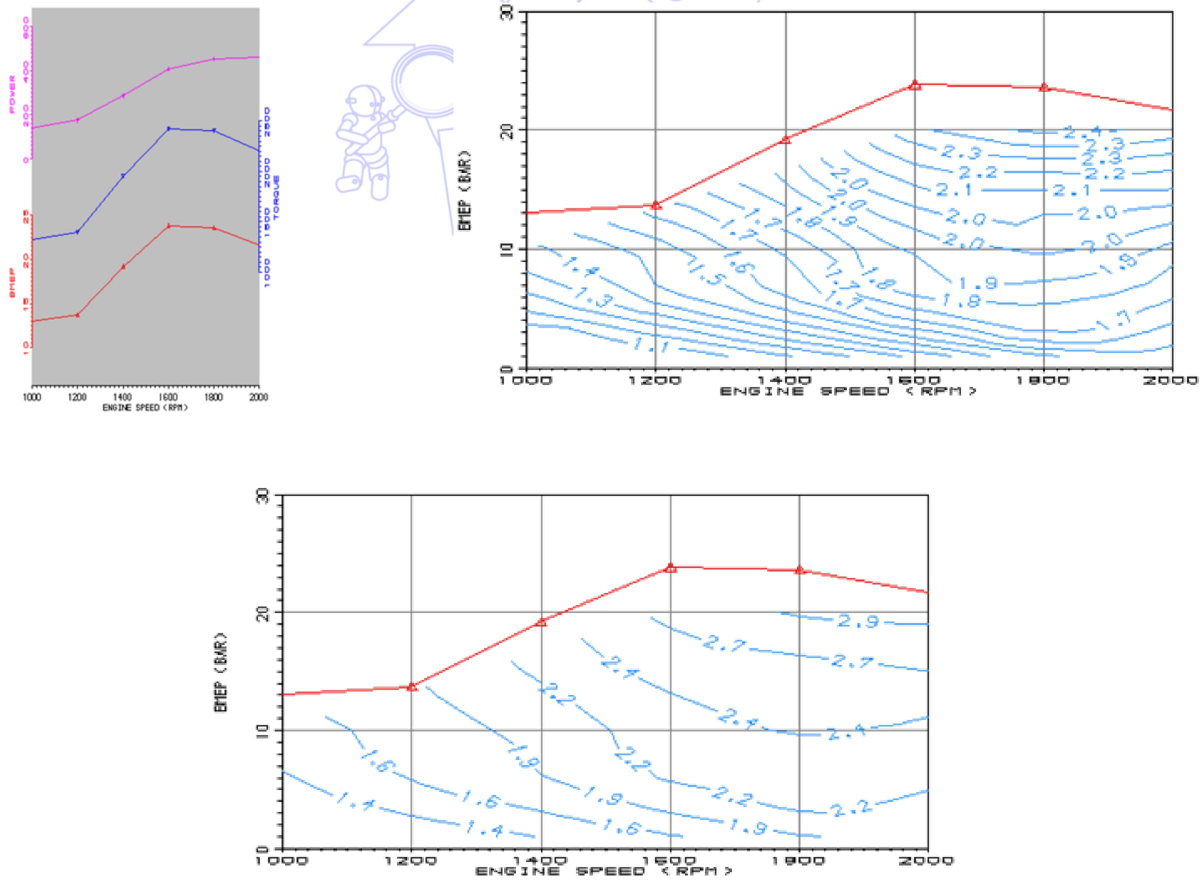

Fig. 2 Volumetric efficiency (top) and compressor pressure ratio (bottom) vs. engine speed and load for the Diesel engine operating without the power turbine. Full load λ=1.45 BMEP (bar), torque (Nm) and power (kW) vs. engine speed also shown

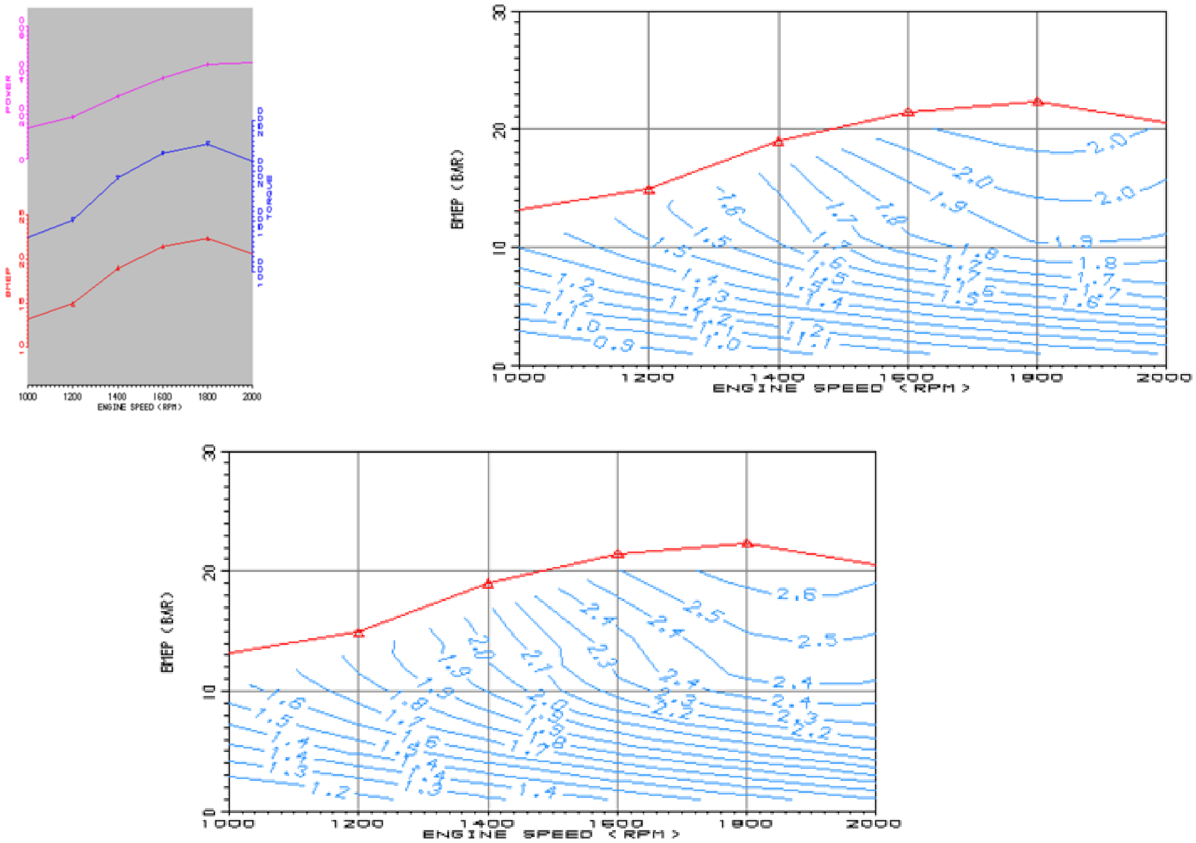

Fig. 3 Volumetric efficiency (top) and compressor pressure ratio (bottom) vs. engine speed and load for the Diesel engine operatingwith the power turbine. Full load λ=1.45 BMEP (bar), torque (Nm) and power (kW) vs. engine speed also shown

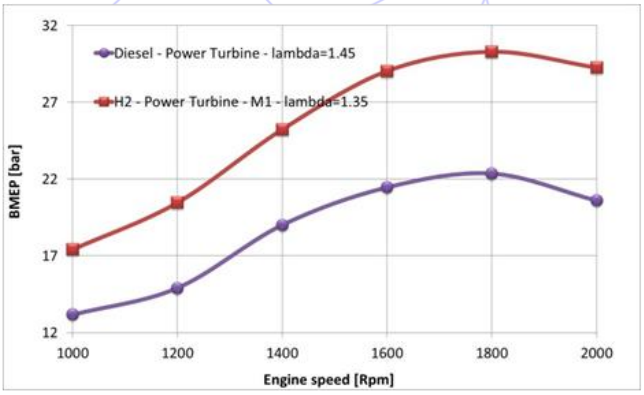

Fig. 4 BMEP of the engine operating λ=1.45 with Diesel or λ=1.35 with hydrogen, with Diesel-like combustion mode M1 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational

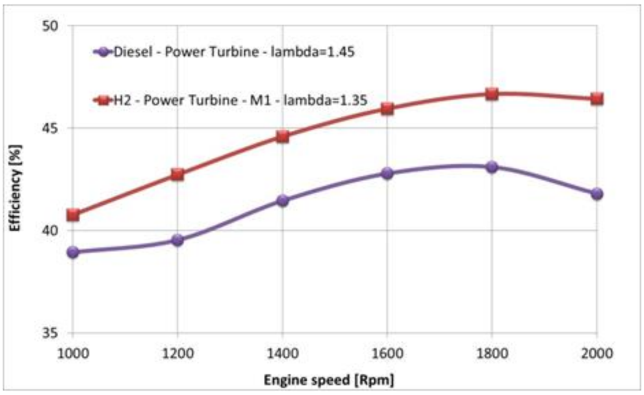

Fig. 5 Fuel conversion efficiency of the engine operating λ=1.45 with Diesel or λ=1.35 with hydrogen, with Diesel-like combustion mode M1 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational

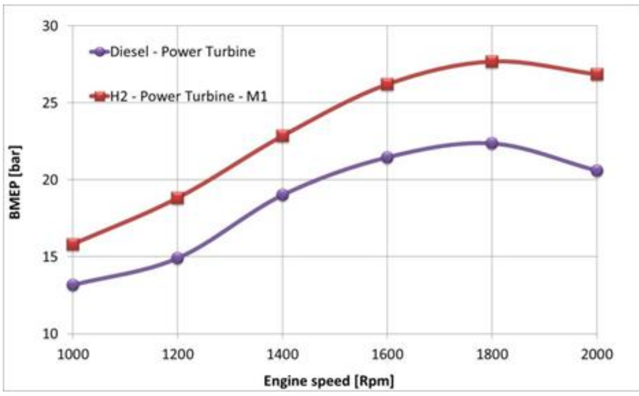

Fig. 6 BMEP of the engine operating λ=1.45 with Diesel and hydrogen, with Diesel-like combustion mode M1 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational

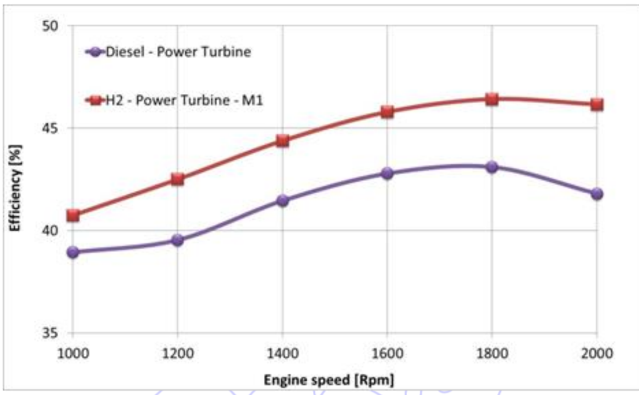

Fig. 7 Fuel conversion efficiency of the engine operating λ=1.45 with Diesel or hydrogen, with Diesel-like combustion mode M1 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational

Fig. 4 to 7 presents the BMEP and the fuel conversion efficiency with Diesel and hydrogen fuels operating in diffusion Diesel-like combustion mode. A λ=1.45 is always considered for the Diesel, where both λ=1.45 and λ=1.35 are considered for the hydrogen fuel. The operation with hydrogen and Diesel is compared with same λ=1.45. Then, because hydrogen permits much richer mixtures (the Diesel is a mixture of complex, liquid hydrocarbons which is much more difficult to burn closer to stoichiometry) a λ=1.35 is considered for hydrogen.

The gaseous fuel has the opportunity to increase both power and torque outputs and efficiency. The operation with same λ permits much higher BMEP with hydrogen than Diesel (5 bar more on average). The operation with same λ also permits much higher fuel conversion efficiencies with hydrogen than Diesel (2-3 percentage points on average). The Diesel fuel has a lower heating value of 43.25 MJ/Kg and a stoichiometric air-to-fuel ratio of 14.33. This translates in 2.82 MJ of energy per kg of stoichiometric mixture of Diesel and air. The hydrogen fuel has a lower heating value of 119.9 MJ/Kg and a stoichiometric air-to-fuel ratio of 34.04. This translates in 3.42 MJ of energy per kg of stoichiometric mixture of hydrogen and air. Clearly, running with hydrogen at similar fuel conversion efficiencies of the Diesel, the same BMEP is obtained with much leaner mixtures. The assumption of similar burning rates for the Diesel and the hydrogen at same percentage of maximum load produces a larger heat release rate with the hydrogen.

Fig. 8 presents the fuel conversion efficiency of the engine operating with Diesel or hydrogen at 20 bar BMEP, with Diesel-like combustion mode when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational or bypassed and disconnected. With Diesel the average working λ is 1.5-1.55 with or without power turbine, while with hydrogen the working λ is 1.73-1.85 with or without power turbine. With the power turbine, the power produced in-cylinder decreases less than the power produced in the power turbine, and the efficiency of the fuel conversion process increases. With hydrogen, the engine works more fuel energy efficiently than with the Diesel. Some points are missed in the speed range because at these speeds the engine delivers less than the listed BMEP.

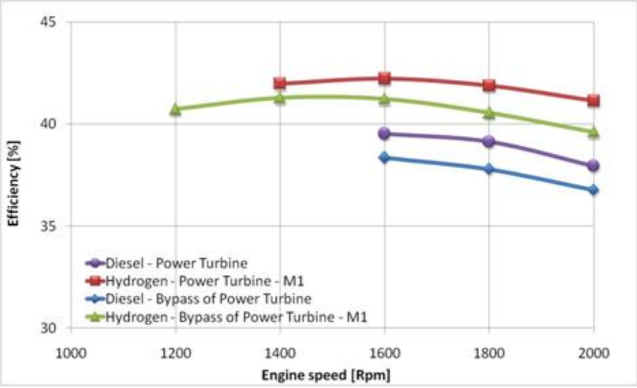

Fig. 8 Fuel conversion efficiency of the engine operating at 20 bar BMEP with Diesel or hydrogen, with Diesel-like combustion mode M1 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational bypassed and disconnected

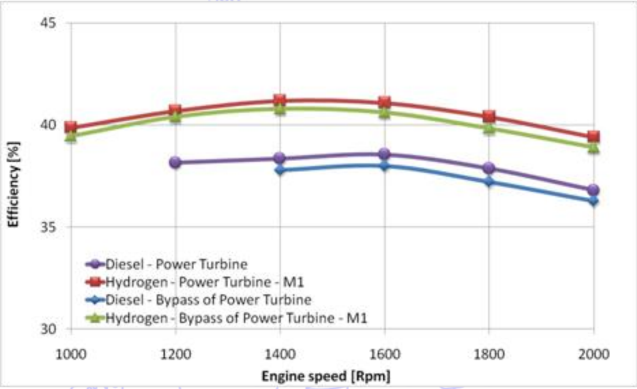

Fig. 9 Fuel conversion efficiency of the engine operating at 15 bar BMEP with Diesel or hydrogen, with Diesel-like combustion mode M1 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational bypassed and disconnected

Fig. 9 presents the fuel conversion efficiency of the engine operating with Diesel or hydrogen fuel at 15 bar BMEP, with Diesel-like combustion mode when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational or bypassed and disconnected. With Diesel the average working λ is 1.79-1.77 with or without power turbine, while with hydrogen the working λ is 1.94-1.98 with or without power turbine. Operating with hydrogen in the mode of HCCI/gasoline like premixed combustion the maximum pressure within the cylinder would be 204-210 bar with or without power turbine. This operation is therefore not shown. At 15 bar, the benefits of the power turbine are almost cancelled. The power turbine is producing a power marginally larger than the power lost within the cylinder for the increased back pressure. With hydrogen, the engine works fuel energy much efficiently than with the Diesel.

Fig. 10 presents the fuel conversion efficiency of the engine operating with Diesel or hydrogen fuel at 10 bar BMEP, with Diesel-like injection and combustion mode Diesel-like or HCCI/gasoline-like when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational or bypassed and disconnected. With Diesel, the average working λ is 2.14-2.17 with or without power turbine, while with hydrogen, the working λ is 2.53-2.61 in combustion mode Diesel-like and 2.35-2.60 in combustion mode HCCI/gasoline-like with or without power turbine. At 10 bar, the benefits of the power turbine are lost. The power turbine is producing a power smaller than the power lost within the cylinder for the increased back pressure. With hydrogen, the engine works fuel energy much efficiently than with the Diesel. Replacing the Diesel-like operation with the operation HCCI/gasoline-like, the fuel conversion efficiency increases. Improvements of efficiency are 1-3 percentage points. Intermediate results are expected to operate the engine in the mixed Diesel/gasoline-like mode and increase the amount of main chamber fuel injected before the injection in the jet ignition pre-chamber.

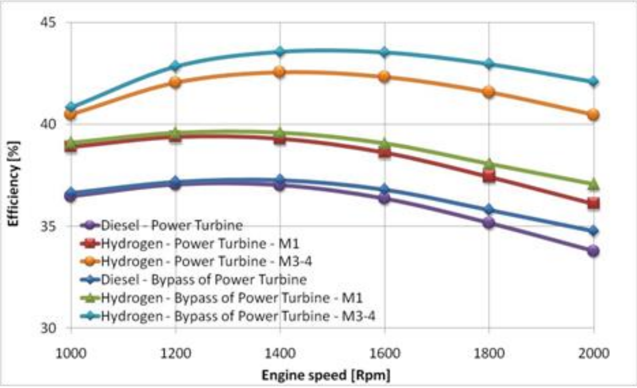

Fig. 10 Fuel conversion efficiency of the engine operating at 10 bar BMEP with Diesel or hydrogen, with Diesel-like combustion mode M1 or HCCI/gasoline-like combustion modes M3-4 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational bypassed and disconnected

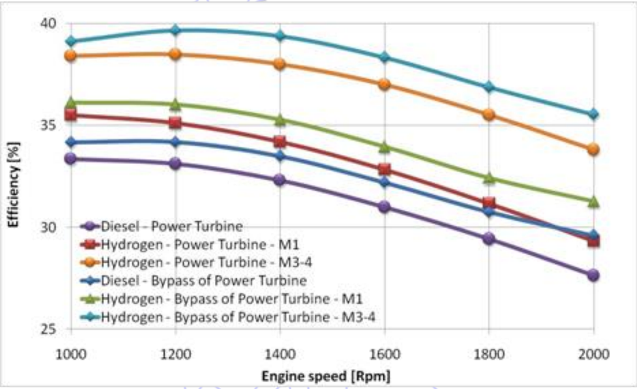

Fig. 11 Fuel conversion efficiency of the engine operating at 5 bar BMEP with Diesel or hydrogen, with Diesel-like combustion mode M1 or HCCI/gasoline-like combustion modes M3-4 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational bypassed and disconnected

Fig. 11 presents the fuel conversion efficiency of the engine operating with Diesel or hydrogen fuel at 5 bar BMEP, with Diesel-like combustion mode or HCCI/gasoline-like combustion modes when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational or bypassed and disconnected. With Diesel, the average working λ is 2.98-3.49 with or without power turbine, while with hydrogen, the working λ is 3.64-4.18 in combustion mode Diesel-like and 3.48-4.21 in combustion mode HCCI/gasoline-like with or without power turbine. At 5 bar, the power turbine has no benefits.

With hydrogen, the engine works fuel energy much efficiently than with the Diesel. Replacing the Diesel-like operation with the operation HCCI/gasoline-like the fuel conversion efficiency increases. Improvements of efficiency are 1 percentage point. Intermediate results are expected to operate the engine in the mixed Diesel/gasoline-like mode increasing the amount of main chamber fuel injected before the injection in the jet ignition pre-chamber.

Fig. 12 presents the fuel conversion efficiency of the engine operating with Diesel or hydrogen fuel at 2.5 bar BMEP, with Diesel-like or HCCI/gasoline-like combustion modes when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational or bypassed and disconnected. With Diesel, the average working λ is 3.98-4.90 with or without power turbine, while with hydrogen, the working λ is 4.88-5.99 in Diesel-like operation and 4.98-6.11 in operation

HCCI/gasoline-like with or without power turbine. Comments on fuel efficiency with hydrogen or Diesel, with or without power turbine and with different modes using hydrogen are same as above for 5 bar.

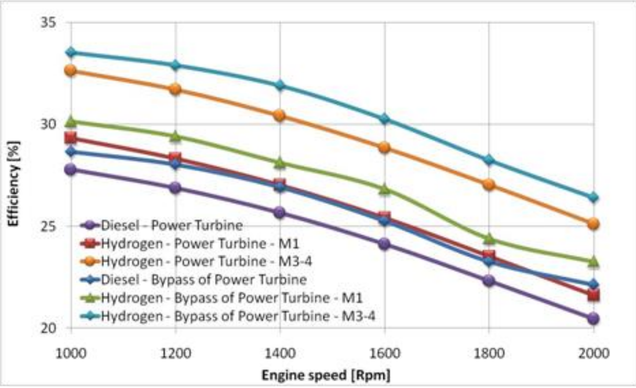

Fig. 12 Fuel conversion efficiency of the engine operating at 2.5 bar BMEP with Diesel or hydrogen, with Diesel-like combustion mode M1 or HCCI/gasoline-like combustion modes M3-4 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational bypassed and disconnected

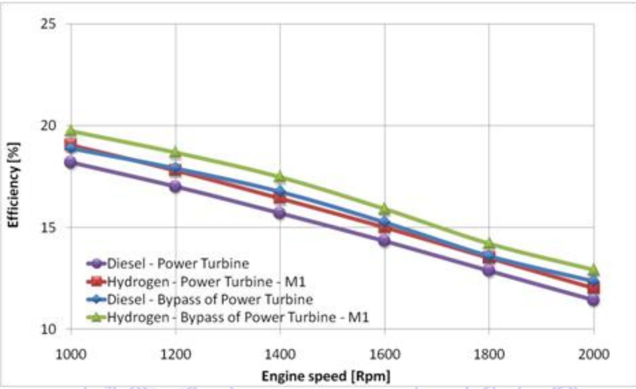

Fig. 13 Fuel conversion efficiency of the engine operating at 1 bar BMEP with Diesel or hydrogen, with Diesel-like combustion mode M1 when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational bypassed and disconnected

Fig. 13 finally presents the fuel conversion efficiency of the engine operating with Diesel or hydrogen fuel at 1 bar BMEP, and with Diesel-like combustion mode when fuelled with hydrogen and power turbine supplementing the power output of the crankshaft operational or bypassed and disconnected. With Diesel, the average working λ is 5.46-6.75 with or without power turbine, while with hydrogen, the working λ is 6.67-8.21 with or without power turbine. With hydrogen in operation HCCI/gasoline-like the working λ would be 7.13-8.72 with or without power turbine and these points are omitted because it is not expected that combustion may occur rapidly at these λ. Comments on fuel efficiency with hydrogen or Diesel, with or without power turbine are same as above, even if differences are smaller because of the much larger importance of friction losses.

Fig. 14 finally presents the maximum efficiency vs. BMEP at different speeds with mode of combustion Diesel-like and power turbine engaged and fluxed, while Figure 15 presents the maximum efficiency vs. BMEP at different speeds with different modes of combustion – Diesel-like or mixed HCCI/gasoline-like - and power turbine operational or bypassed and disconnected. The mixed HCCI/gasoline-like mode is only allowed at BMEP values of 2.5 to 10 bar. The largest BMEP are obtained without the power turbine. The power turbine helps at high loads to improve the fuel conversion efficiency. These figures give the trends of fuel conversion efficiency vs. the load.

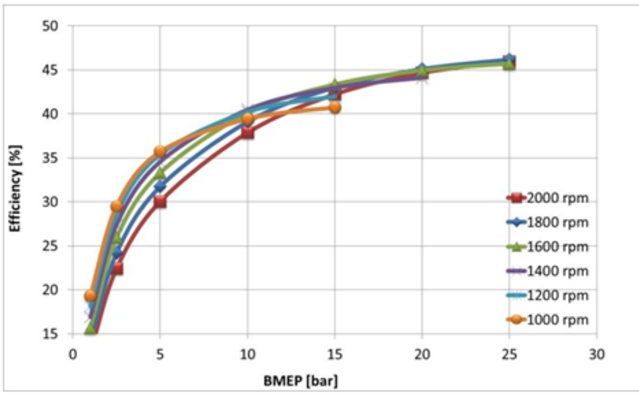

Fig. 14 Maximum efficiency vs. BMEP at different speeds with mode of combustion M1 and power turbine operational

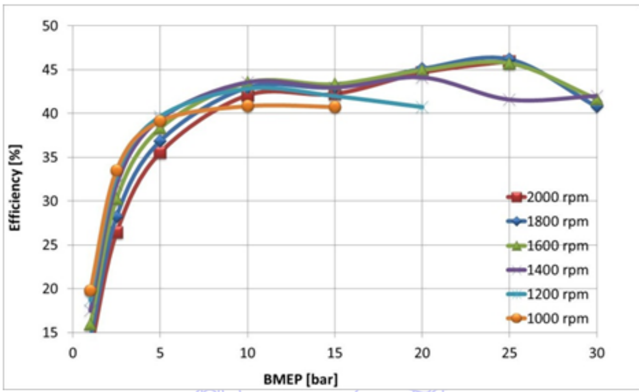

Fig. 15 Maximum efficiency vs. BMEP at different speeds with different modes of combustion – M1 or M3-4 - and power turbine operational or disengaged and bypassed. The M3-4 mode is only allowed at BMEP values of 2.5 to 10 bar. The largest BMEP are obtained without the power turbine

Reducing the engine speed for a given BMEP, the maximum in the fuel conversion efficiency is clearly shifting to the left to lower engine speeds. This is mostly the result of a friction component of BMEP is weakly dependent on the load but is linearly and quadratically dependent on the speed. The friction mean effective pressure FMEP is computed through the Chen and Flynn correlation [30]. The correlation has a constant term (for accessory friction), a term which varies with peak cylinder pressure, a third term is linearly dependent on mean piston velocity (for hydrodynamic friction) and a fourth term quadratic with mean piston velocity (for windage losses). The equation used to calculate the Friction Mean Effective Pressure (FMEP) is given below:

$$FMEP = A_{cf} + \frac{1}{ncyl}\sum_{i=1}^{ncyl}\left[B_{cf}\cdot\left(P_{cyl}\right)_i + C_{cf}\cdot\left(S_{fact}\right)_i + Q_{cf}\cdot\left(S_{fact}\right)_i^2\right] \quad (9)$$

with: Sfact=½•RPM•S, Acf, Bcf, Ccf and Qcf user inputs, Pmax the maximum cylinder pressure, RPM the cycle-average engine speed and S the cylinder stroke. Each cylinder has its own contribution to the total engine friction based upon its own maximum cylinder pressure and stroke (folded into the speed factor, Sfact). Acf is the constant portion of the Chen-Flynn friction correlation, Bcf the term which varies linearly with peak cylinder pressure, Ccf the term accounting for hydrodynamic friction in the power cylinder which varies linearly with the piston speed and Qcf the term which varies quadratically with the piston speed and accounts for windage losses in the power cylinder. As soon as the load is reduced, being BMEP=IMEP-FMEP, with IMEP the indicated mean effective pressure resulting from the cylinder pressure work, the operation at higher engine speeds clearly result to be less efficient.

5. Conclusions

The paper has presented the numerical study of a Diesel direct injection heavy duty truck engine converted to hydrogen. The conversion is obtained by replacing the Diesel injector with a hydrogen injector and the glow plug with a jet ignition device. The engine has a power turbine connected through a clutch and a continuously variable transmission to the crankshaft and a by-pass. The power turbine may be disconnected when it's inefficient or inconvenient to use.

The power turbine increases the fuel conversion efficiency at medium-high loads. Top outputs are however obtained without power turbine. The power turbine is more effective with the hydrogen than with the Diesel. Thanks to the higher temperatures of the exhaust gases, the power turbine helps with hydrogen more than with the Diesel.

The hydrogen engine operates different modes of combustion depending on the relative phase of the main injection and the jet ignition. Therefore, it can optimize the fuel conversion efficiency of every map point following different strategies.

The engine generally operates in Diesel-like mode, with the most part of the main injection following the suitable creation in cylinder conditions by jet ignition. For medium-low loads, better efficiencies are obtained with the gasoline-like mode, jet igniting the premixed homogeneous mixture at top dead centre, it's not permitted at higher loads or at very low loads for excessive peak pressure or mixture too lean.

Thanks to the larger rate of heat release, the hydrogen engine has better outputs and fuel conversion efficiencies than Diesel and has the opportunity to run closer to stoichiometry and the multi mode operation.

The practical design of these hydrogen truck engines requires the development of better fuel injectors which are able to deliver the mass of hydrogen requested by the large displacement cylinder within small time frames.

Further optimization is also needed for the jet ignition pre-chamber concept to properly deliver the hot reacting gases within the cylinder that are required to make the combustion of the following main injection of hydrogen diffusion controlled or ignite the premixed mixture depending on the operating mode selected.

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Abbreviations

BDC bottom dead centre

CNG compressed natural gas

CVT continuously variable transmission

EGR exhaust gas recirculation

EVO exhaust valve opening

EVCexhaust valve closure

HPDI high pressure direct injection

IVO intake valve opening

IVC intake valve closure

LHV lower heating value

LNG liquefied natural gas

TC turbocharged

TDC top dead centre