Numerical Evaluation of the Performance of a Compression Ignition CNG Engine for Heavy Duty Trucks with an Optimum Speed Power Turbine

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Abstract

The turbocharged direct injection lean burn Diesel engine is the most efficient engine now in production for transport applications. CNG is an alternative fuel with a better carbon to hydrogen ratio therefore permitting reduced carbon dioxide emissions. It is injected in gaseous form for a much cleaner combustion almost cancelling some of the emissions of the Diesel and it permits a much better energy security within Australia. The paper discusses the best options currently available to convert Diesel engine platforms to CNG, with particular emphasis to the use of these CNG engines within Australia where the refuelling network is scarce. This option is determined in the dual fuel operation with a double injector design that couples a second CNG injector to the Diesel injector. This configuration permits the operation Diesel only or Diesel pilot and CNG main depending on the availability of refuelling stations where the vehicle operates. Results of engine performance simulations are performed for a straight six cylinder 13 litres truck engine with a novel power turbine connected to the crankshaft through a constant variable transmission that may be by-passed when non helpful to increase the fuel economy of the vehicle or when damaging the performances of the after treatment system.

Keywords: alternative fuels, power turbine, dual fuel engines

1. Introduction

Compressed natural gas (CNG) is a fossil as well as a renewable fuel substitute for traditional gasoline (petrol) or Diesel fuels. Renewable natural gas, also known as sustainable natural gas, is a biogas obtained from biomass that may possibly supplement if not replacing the fossil natural gas in the future. Natural gas is one of the world's most abundant sources of primary energy, and has been in use as a transportation fuel for more than 60 years. Natural gas is non-toxic. It is much lighter than air and diffuses quickly if released, unlike diesel fuel that pools on the ground. When it comes to protecting public health and the environment, natural gas is the cleanest fuel that is widely available today. Natural gas is composed primarily of methane, and burns more cleanly than diesel fuel or gasoline because it has more hydrogen and less carbon. Although its combustion also produces carbon dioxide CO_2 at the tail pipe, it is a more environmentally clean alternative to those fuels for the much better C to H ratio permitting a lower value of CO_2 per MJ of fuel energy. CNG is much safer than other fuels in the

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event of a spill because it is lighter than air and disperses quickly when released. When produced from landfills or waste water, the fuel life cycle carbon dioxide CO_2-e per MJ of fuel energy may be one order of magnitude lower than with fossil gasoline and Diesel. Fig. 1 presents the CARB fuel cycle values of CO_2-e per MJ of fuel energy for Diesel, gasoline/petrol and some renewable and non renewable alternative fuels (picture from [1], with data presented in [2]). The combustion of the CNG delivered in gas phase is much cleaner than the combustion of the Diesel delivered as a mixture of liquid hydrocarbons. And this reduces the formation of pollutants, smoke and soot first of all. Finally, virtually all of Australia's natural gas supply is continentally sourced, making natural gas less vulnerable to foreign supply disruption and price volatility.

CNG is made by compressing natural gas which is mainly composed of methane [CH_4] to less than 1% of the volume it occupies at standard atmospheric pressure. It is stored and distributed in hard cylindrical or spherical containers at a pressure of 200–250 bar. CNG is widely used in traditional spark ignition gasoline internal combustion engines that have been converted into bi-fuel vehicles (gasoline/CNG). CNG may be used as a replacement of gasoline in a spark ignition engine. However, it is as a replacement of Diesel in a compression ignition engine that it may work better thanks to the better top fuel conversion efficiencies and part load efficiency penalty changing the load. In response to fuel prices, energy security and environmental concerns, CNG is starting to be used also in commercial applications. The major downfalls of CNG are the low volumetric energy density estimated to be 40% of LNG and 25% of Diesel and the unavailability of a capillary network for distribution. CNG vehicles require a greater amount of space for fuel storage than conventional gasoline and Diesel powered vehicles and this results in a loss of payload for commercial applications. Furthermore, the coverage of long distances as those typical of the freight transport within Australia will be prohibitive without refueling stations.

The Westport engines set the benchmark for CNG heavy-duty trucks [3-4]. Westport has developed two technologies for CNG. The HPDI technology [3-13] uses a dual fuel High Pressure Direct Injection (HPDI) injector for both the Diesel and the CNG fuels delivering a small Diesel pilot spray and a larger gas spray. The diesel averages only about 5% of the total energy input and is used to start the main combustion of high-pressure directly-injected natural gas. Efficiencies of the converted engine are Diesel-like, with BMEP values that have been increased up to 24 bar. The CNG-DI technology [5] uses a Compressed Natural Gas Direct Injection (CNG-DI) injector for only the CNG. Ignition is then forced with either a hot surface or a spark. In the HPDI technology, basically the main CNG injection occurs within an environment where the diffusion controlled combustion of the CNG is made possible by the temperature and pressures created by the previous pilot Diesel injection. In the CNG-DI technology, the start of combustion is more complicated requiring a continuously operated glow plug to bring the temperatures up to the CNG auto ignition or even using a spark discharge to initiate combustion changing the Diesel-like operation to gasoline-like. The CNG-DI technology certainly requires much more effort to work properly than the HPDI technology.

In the present paper, the adoption of two injectors, one for the Diesel and one for the CNG, is considered to deliver the amount of fuel needed in the two possible mode of operation, Diesel only and Diesel pilot and CNG main. In the dual fuel operation, more complicated strategies of simultaneous Diesel and CNG injections are possible but not considered just for sake of simplicity.

The engine considered as the baseline is a Diesel engine having a power turbine. Internal combustion engines have a maximum theoretical fuel conversion efficiency that is similar to that of fuel cells and considerably higher than the mid-40% peak values seen today [14-15]. The primary limiting factors to approaching these theoretical limits of conversion efficiency

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start with the high irreversibility in traditional premixed or diffusion flames, but include heat losses during combustion and expansion, untapped exhaust energy, and mechanical friction. One area where there is large potential for improvements in ICE efficiency is losses from the exhaust gases [14-15].

Exhaust losses are being addressed by analysis and development of compound compression and expansion cycles achieved by variable valve timing, variable stroke use of turbine expanders, regenerative heat recovery, and application of thermoelectric generators. Employing such cycles and devices, it has been claimed to have the potential to increase engine efficiency by 10% [14-15]. Of all these technologies, the more mature is the use of two turbines in series; one waste gated driving the upstream compressor, and one to producing additional power to the driveline, historically known as turbo compound.
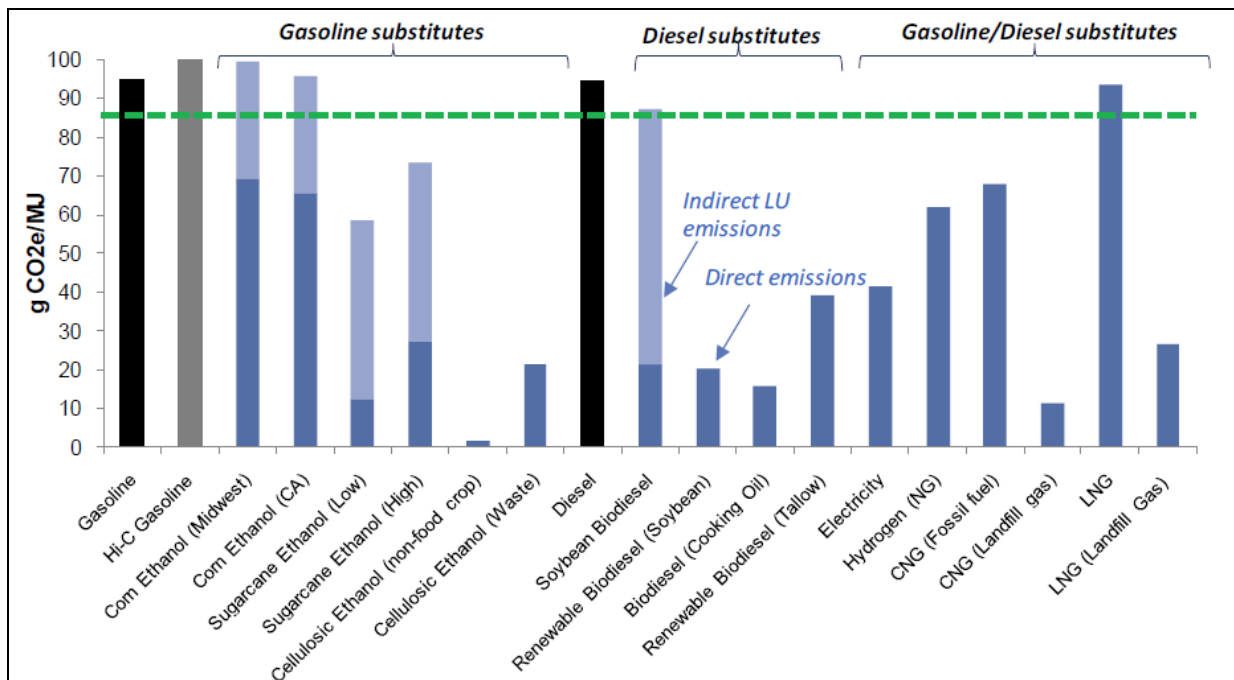

Fig. 1 – CARB fuel cycle values of CO_2-e per MJ of fuel energy for Diesel, gasoline/petrol and some renewable and non renewable alternative fuels [1, 2]. CNG fossil is rated about 70 gCO_2-e/MJ vs. the 95 gCO_2-e/MJ of gasoline and Diesel. CNG from landfill is rated about 10 gCO_2-e/MJ and LNG from landfill is rated about 25 gCO_2-e/M. Results of life cycle analyses of transportation fuels vary from country to country and from author to author depending on the many factor affecting the result and the future scenarios that are predicted.

The use of turbo compounding goes back quite a long way. The concept was originally used back in the late 1940s on aircraft engines [16]. Its promise of low fuel consumption was soon overtaken by the rapid development of the turboprop engine. For automotive diesel engines, the introduction of a power turbine downstream of the turbocharger generates more work by re-using the exhaust gases from the conventional turbocharger [17-18]. The work generated by the power turbine is then fed back into the engine crankshaft via an advanced transmission. A gear is fitted to the power turbine shaft. To assist the power turbine, the turbine for the conventional turbocharger is designed for a reduced expansion ratio. This small turbocharger gives another system advantage to the turbo compound engine, providing better transient response and higher boost pressure for improved low speed torque. The behavior of the two turbines in series - turbocharger and power turbine – offers a dynamic response across the engine speed and air flow range. However, it requires optimum matching of the turbines.

Fig. 2 presents one of the most successful applications of turbo compound technology to a truck engine (from [17]). 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 mechanical transmission made up of a gear train and

a hydraulic coupling, the turbine is running with maximum speeds of up to 70,000 rpm. The gear ratio from the exhaust turbine to the turbo compound intermediate shaft is around 6:1, and the gear ratio from the intermediate shaft to the crankshaft is around 5:1 for an overall gear ratio of about 30:1. Torsional vibrations caused from the internal combustion engine process would be increased by the overall gear ratio exceeding 30:1 and could possibly destroy the turbine. To reduce torsional vibrations the turbo compound intermediate shaft is equipped with a hydrodynamic coupling with a slip inside the coupling that is normally around 2%.
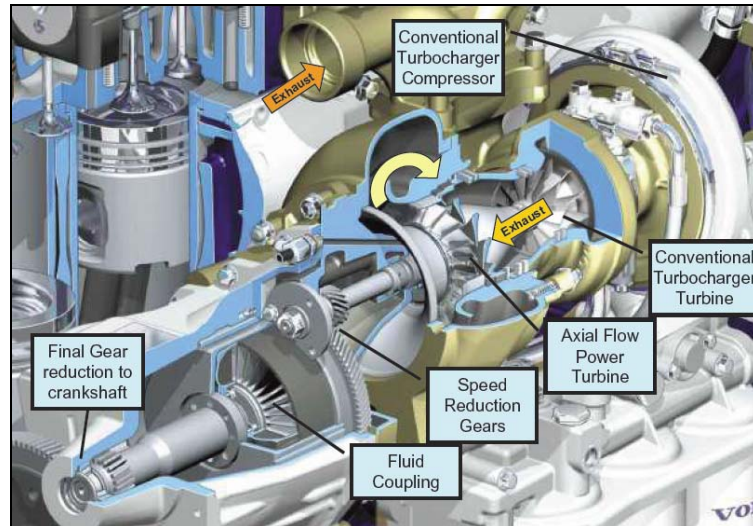
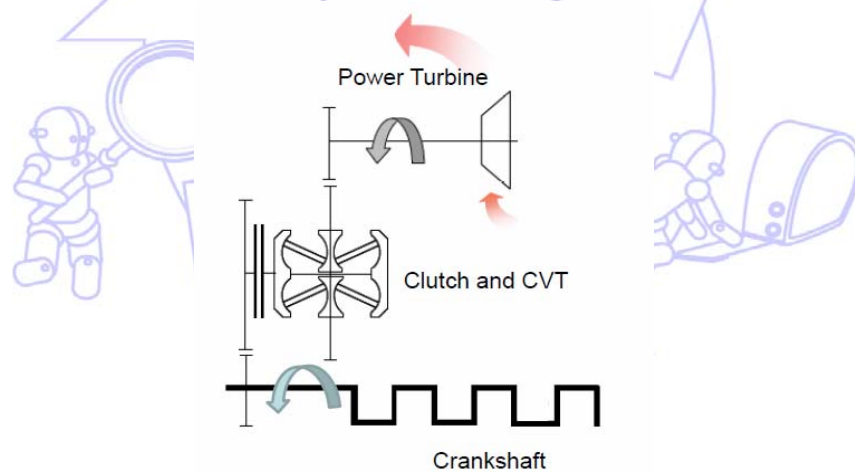Fig. 2 – Traditional turbo compound design (from [17]).

Fig. 3 - Sketch of the novel power turbine layout. With reference to the arrangement previously proposed with just a gear in between the crankshaft and the power turbine shaft, this design now include a clutch and a continuously variable transmission to disconnect the power turbine when bypassed by the exhaust gases and to operated the power turbine at the most favorable speed.

Despite few truck manufacturers' claim of better efficiency and increased power output as a result of the additional power turbine, this solution has not encountered so far the favor of the vast majority of truck and the totality of car manufacturers. Because the amount of energy available downstream of the turbocharger turbine is small and the complexity to add a second power turbine geared to the crankshaft is high, turbo compound is expected in principle to provide limited advantages in a limited area of operation of the engine at high costs in terms of design, control, packaging and weight. Engine manufacturers that have had or will have Turbo compound Engines include Volvo, Iveco (off-highway) and Scania. Detroit Diesel, Cummins, CAT, Mercedes, and International have also considered the technology.

In the present paper, the traditional gear train coupling of the power turbine is replaced by a constant 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. A sketch of the novel design is proposed in Fig. 3.

Engine performance simulations are performed to show the benefit of this turbo compound in a 12.8 liters straight 6-cylinder Diesel engine with turbocharger and intercooler and Diesel fuel only. The engine is then fuelled with both the Diesel and the CNG to simulate the pilot Diesel and main CNG performances and the relative benefits.

2.   Dual Fuel Diesel-like combustion in engine performance codes

The direct injection of gas and Diesel within the cylinder is not a novelty. Since the early 1980's, several researches have been made on the subject [6-13]. A small amount of Diesel fuel was injected before a main injection of natural gas to start the combustion of the engine. With this technology, Diesel engines were able to run on natural gas without compromises in performance and fuel efficiency.

The major advantage of the Direct Injection Diesel engine is the bulk, lean, diffusion combustion in a high compression ratio highly boosted engine with reduced heat losses to the liner and head walls. The engines using the stoichiometric, premixed combustion controlled by a spark discharge suffer the disadvantage of the higher heat losses to the liner and head walls. The combustion is wall initiated, and the flame has to reach all the walls for burning all the homogeneously distributed fuel. Furthermore, the compression ratio is reduced to avoid the abnormal combustion phenomena. Furthermore, in the lean burning Diesel the load may be controlled by the quantity of fuel injected, while the stoichiometric gasoline requires throttling the intake to vary the load and this bring a further advantage of the Diesel able to perform with smaller penalties from already higher efficiencies reducing the load.

In case of dual fuel operation of a Diesel engine with CNG as the second fuel, it seems therefore logical to inject both the Diesel and the CNG fuel within the cylinder to achieve the same as Diesel efficiencies both full load and part load, with the load being controlled by the quantity of fuel injected.

The single dual fuel injector has the advantage to be easily incorporated into an existing Diesel engine with minimal or no modifications to the engine cylinder head, while the two injectors' option requires a new design of the cylinder head only possible on novel engines of large bores. Late-cycle, high-pressure direct injection ensures diffusion type combustion for the CNG and therefore retains the high power, torque, and efficiency of the Diesel engine with possibly the advantages of the higher power densities.

Not considered here, flexible operation of the two injectors may also permit different combustion strategies, not only Diesel-like but also gasoline-like or alternative low temperatures as the homogeneous charge compression ignition modes. Homogeneous charge compression ignition modes may be achieved directly injecting the CNG early during compression, Diesel-like operation may be achieved preceding a late CNG main injection with a Diesel pre injection, gasoline-like operation may be achieved injecting the Diesel after the CNG, and mixed Diesel-like and gasoline-like operation may be finally obtained performing simultaneously the two injections. Finally, operating the glow plug continuously may permit auto ignition of the CNG without the Diesel injection.

The CNG injector has to manage the high volumetric flow rates required for the CNG fuel much less dense than Diesel

and the carbonizing and thermal fluctuations that can result from the reduced cooling capacity of a gaseous fuel. The major disadvantage of the CNG-Diesel fuel injector is the complex fuel routing for two fuels within one injector. This injector provides a small diesel pilot spray and a much larger CNG gas spray. The Diesel averages only about 5% of the total energy input and is used to start the main combustion of high-pressure directly-injected CNG gas and the reduced pollutants of the CNG.
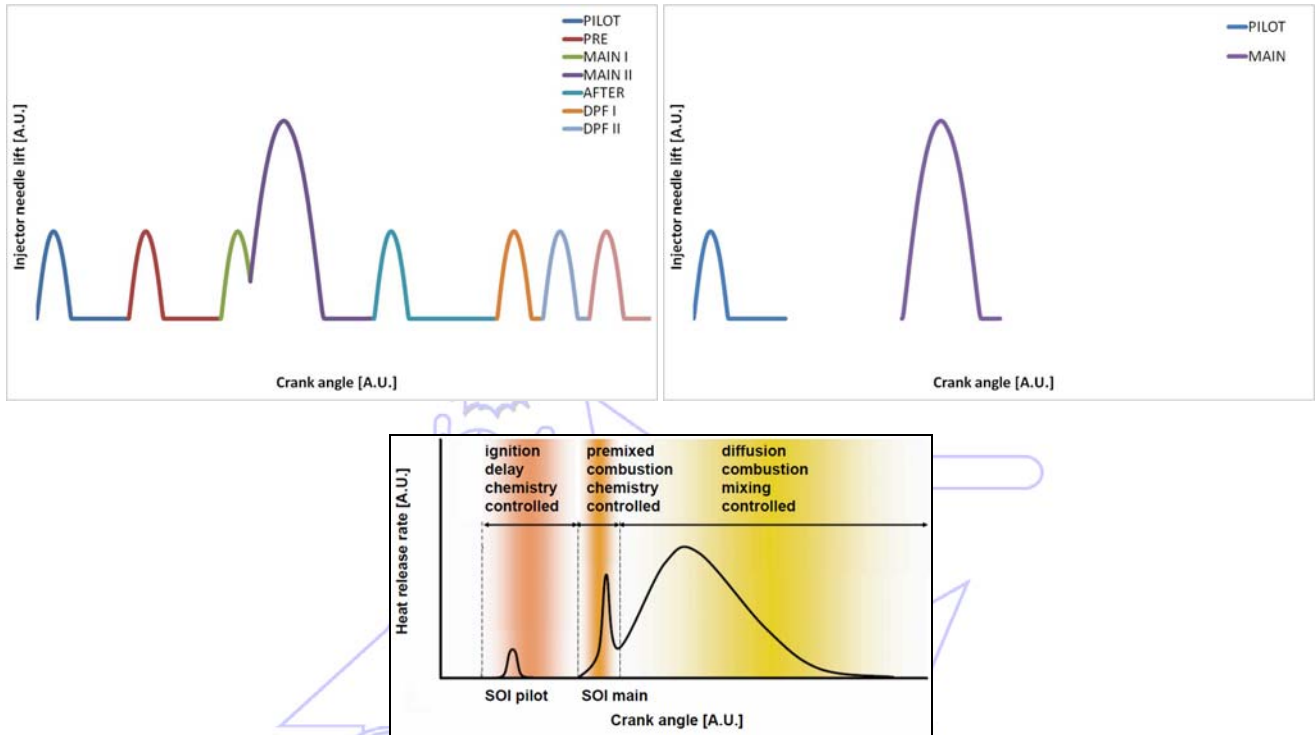

Fig. 4 - Latest generation common-rail multiple injection strategy (top left), first generation common-rail injection strategy (top right) and modeled heat release rate profile vs. crank angle with Diesel pilot and main Diesel or gas injections (bottom). This heat release profile is the one adopted in the simulations described below where a Diesel Wiebe function is used to describe the combustion evolution.

Fig. 4 shows one latest generation common-rail multiple injection strategy (top), first generation common-rail injection strategy (middle) and a typical heat release rate profile vs. crank angle for this latter injection profile (bottom). Diesel particulate filter regenerations obviously do not apply to steady state map points.

For the purpose of fuel conversion efficiency computations only, the complex injection of Fig. 4 on top may be replaced by a single equivalent injection event. Obviously, soot, smoke, NOx, HC and CO emissions are not included in the prediction. Engine performance models are not able to represent complicated injection and combustion phenomena as those of Fig. 4 on top, but they can tackle reasonably well single injection or pilot and main combustion events. Therefore, having measured pressure traces for the Different map points with the Diesel, the heat release analysis may provide the equivalent injection and combustion model parameters for all the different (BMEP, speed) map points. Then, after tuning on the start of injection and the injection duration, the engine model may provide maps of brake specific fuel consumption vs. BMEP and speed very close to the experimental ones. Even when the accuracy of the combustion model parameters reduces, tuning on the start of injection and the injection duration may still produce acceptable results.

In what follows, the assumption made is that with Diesel only and Diesel pilot and CNG main, the only thing that changes are the fuel properties, but the combustion parameters and the start and duration of injection set for the Diesel are also valid for the Diesel and the CNG working at same speed and percentage of maximum load. The fuel properties are therefore the only parameters changed in the engine model.

The main injection combustion is modeled with the GT-POWER DI Wiebe model [19]. This model imposes the burn rate using a three-term Wiebe function (the superposition of three normal Wiebe curves). These Wiebe curves 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.

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

$$WC_P = \left[\frac{D_P}{2.302^{1/(E_P+1)} - 0.1051^{1/(E_P+1)}} \right]^{-(E_P+1)} \tag{2}$$

$$WC_M = \left[\frac{D_M}{2.302^{1/(E_M+1)} - 0.1051^{1/(E_M+1)}} \right]^{-(E_M+1)} \tag{3}$$

$$WC_T = \left[\frac{D_T}{2.302^{1/(E_T+1)} - 0.1051^{1/(E_T+1)}} \right]^{-(E_T+1)} \tag{4}$$

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

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

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 values of the model inputs may be obtained using the excel worksheet accompanying the software from a recorded pressure trace [19]. With pressure traces recorded at different loads and speeds, then the model inputs can be made a tabular function of speed and load [19] for all the inputs. Tables of combustion parameters may be transferred in between similar engines

The Westport HPDI concept is very well established. It has been demonstrated in a huge number of heavy duty truck Diesel engine conversions to run dual fuel Diesel and LNG both engine dynamometer and road vehicle tested [3-7, 20]. After roughly 5 per cent of the total fuel energy is introduced with the Diesel burning as usual, the pressures and temperatures and the availability of radicals of the partially burned combustion gases makes the combustion of the LNG introduced next a diffusion controlled more than a kinetically controlled process. It has been proved numerically with coupled fluid dynamic and detailed chemical kinetic simulations that as soon as the in-cylinder conditions have been made suitable for the diffusion combustion of a low Cetane number gas, either hydrogen, methane or propane, then the gas burns diffusion controlled once injected. This is the rationale behind the use of the Diesel Wiebe equation for representing the heat release rate with fuels of different Cetane number [21-23]. Their injection does not occur in air, but in hot partially burned combustion products.

3. Numerical results

Engine performance simulations have been performed with the GT-POWER code [19] for a 12.8 Liter in-line six cylinder turbo charged directly injected Diesel engine. GT-POWER is the industry leading engine simulation tool used by every major engine manufacturer worldwide. Engine simulation, sometimes referred to as cycle simulation, is a mainstream activity conducted by all engine manufacturers well documented in the literature. Within this activity, GT-POWER is used to perform a variety of both steady state and transient analyses. Engine simulations usually provide accurate descriptions of the full load torque curve and the brake specific fuel consumption map. Conversely, these simulations are not able to accurately predict the brake specific fuel consumption emissions maps.

Table 1 – 12.8 Liter in-line six cylinder turbo charged directly injected Diesel engine

displacement per cylinder [l]	2.13
number of cylinders	6
engine layout	I-6
compression ratio	18
bore [mm]	131
stroke [mm]	158
connecting rod length [mm]	300
wrist pin offset [mm]	0
clearance volume [l]	0.125
engine type	C.I.
number of intake valve per cylinder	2
Intake valve diameter [mm]	46.3
Intake valve maximum lift [mm]	12.1
number of exhaust valve per cylinder	2
IVO [deg]	314(-46)
IVC [deg]	602 (+62)
Exhaust valve diameter [mm]	41.5
Exhaust valve maximum lift [mm]	13.4
EVO [deg]	100 (-80)
EVC [deg]	400(+40)

The engine geometry (baseline) is presented in Table 1. IVO is the intake valve opening, IVC the intake valve closure, EVO the exhaust valve opening and EVC the exhaust valve closure. IVO of 314° means the intake valves open 46° before TDC. IVC of 602° means the intake valves close 62° after BDC. EVO of 100° means the exhaust valves open 80° before BDC. EVC of 400° means the exhaust valves close 40° after TDC. A power turbine downstream of the turbocharger is used to recover a percentage of the thermal energy that would normally be lost through the engine's exhaust.

Engine simulations have then been performed without a power turbine. A by-pass of the power turbine is also included. This permits to operate the power turbine when the turbine produces more power than the power loss for the back pressures while permitting temperatures to the downstream after treatment system high enough. The CVT permits operation of the power turbine at the more efficient speeds. The 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 efficiencies of 44%. (The Volvo D13 and D16 engines are now a clear benchmark for truck manufacturers, delivering up to 2,600 Nm of torque and up to 400 kW of power the 13 liter, and up to 3,150 Nm of torque and 515 kW of power the 16 liter [24]).

The engine is compliant with EURO-5 emission standards. The additional cooling of the exhaust gases through the power turbine may in principle reduce the effectiveness of the exhaust after treatment systems. In principle, this may require more active regenerations for particulate filter, or less use of cooled EGR. The additional cooling may certainly reduce the time when NOx systems are effective (LNA, SCR, or LNC). However, using the power turbine for high loads and speeds only, this is not an issue, cause temperatures for these loads and speeds will be otherwise much higher than those of medium to low loads and speeds.

The Diesel fuel is considered a liquid hydrocarbon of composition C=13.5 and H=23.6, fuel heating value at 298.15 K of 43.25 MJ/kg and stoichiometric air-to-fuel ratio of 14.33. The CNG fuel is considered methane gas of composition C=1 H=4, fuel heating value at 298.15 K of 50 MJ/kg and stoichiometric air-to-fuel ratio of 17.12.

3.1.   Diesel only with and without power turbine

Fig. 5 presents the full load brake efficiency, power, and BMEP. Fig. 6 presents the brake efficiency operating at 23 bar BMEP, 19 bar BMEP and 15 bar BMEP. The target values are those of an existing heavy duty truck engine setting the benchmark in terms of efficiency and specific power for the category. This engine serves as the baseline for the development of the model. The validated model is first modified to accommodate the specific engine parameters of Table 1 then extended to add a power turbine and operate dual fuel. No further validation exercise is possible on the modified configurations.
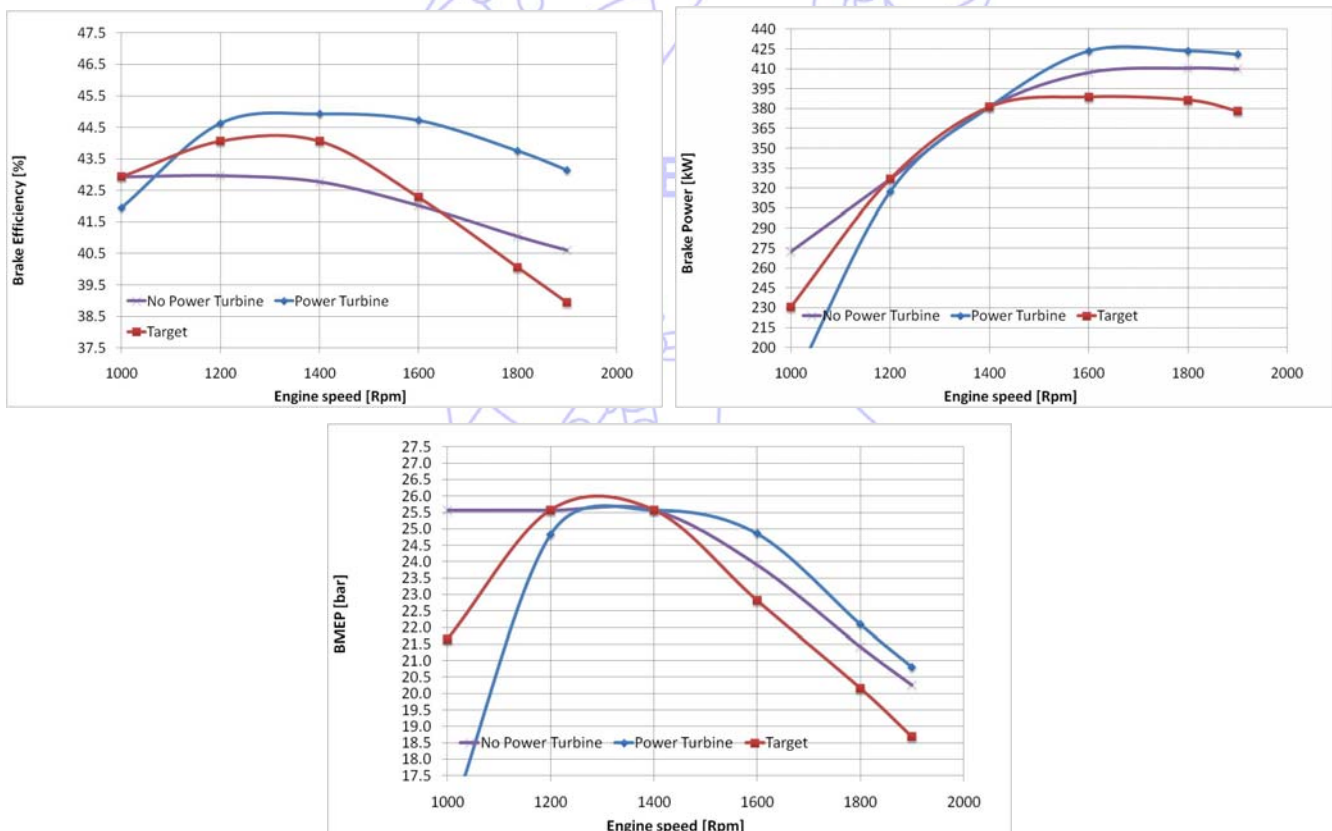

Fig. 5 –Full load brake efficiency, power and brake mean effective pressure – Diesel engine. The target values are those of an existing heavy duty truck engine setting the benchmark in terms of efficiency and specific power for the category.

The power turbine does not help too much to increase the torque or the power output except than at very high speeds. It is therefore effective for this criterion only over a very narrow portion of the speed range about maximum speed. This is because using the exhaust energy in the turbine turbocharger permits to introduce much more air within the cylinder that

translates in much more fuel within the cylinder operating with a fixed air-to-fuel ratio. However, sometimes the turbocharger turbine does not use the available energy about the maximum speed and maximum load operating point because of the requirements to properly cover all the speeds and loads range, and there is therefore space for further improvements there.

The power turbine helps considerably in increasing the efficiency, with increases of almost 4 percentage points in the brake efficiency. These numbers are slightly better than those already provided by other researchers, where improvements of 2 percentage points where obtained with a power turbine connected to the crankshaft through a fixed gear ratio. The reason is the operation of the power turbine in a better speed range thanks to the continuously variable transmission (CVT). The CVT can change steplessly through an infinite number of effective gear ratios between maximum and minimum values in contrasts with other mechanical transmissions that only offer a fixed number of gear ratios or even a single gear ratio as proposed in the first generation of power turbines. The flexibility of the CVT allows the power turbine shaft to maintain a nearly optimum angular velocity over a range of output velocities for the crank shaft.

Thanks to the CVT and the by-pass, the power turbine is operated close to the area of high efficiency when the conversion of the energy available produces more benefits than the increased back pressure and the temperature downstream of the power turbine remains high enough for the after treatment.
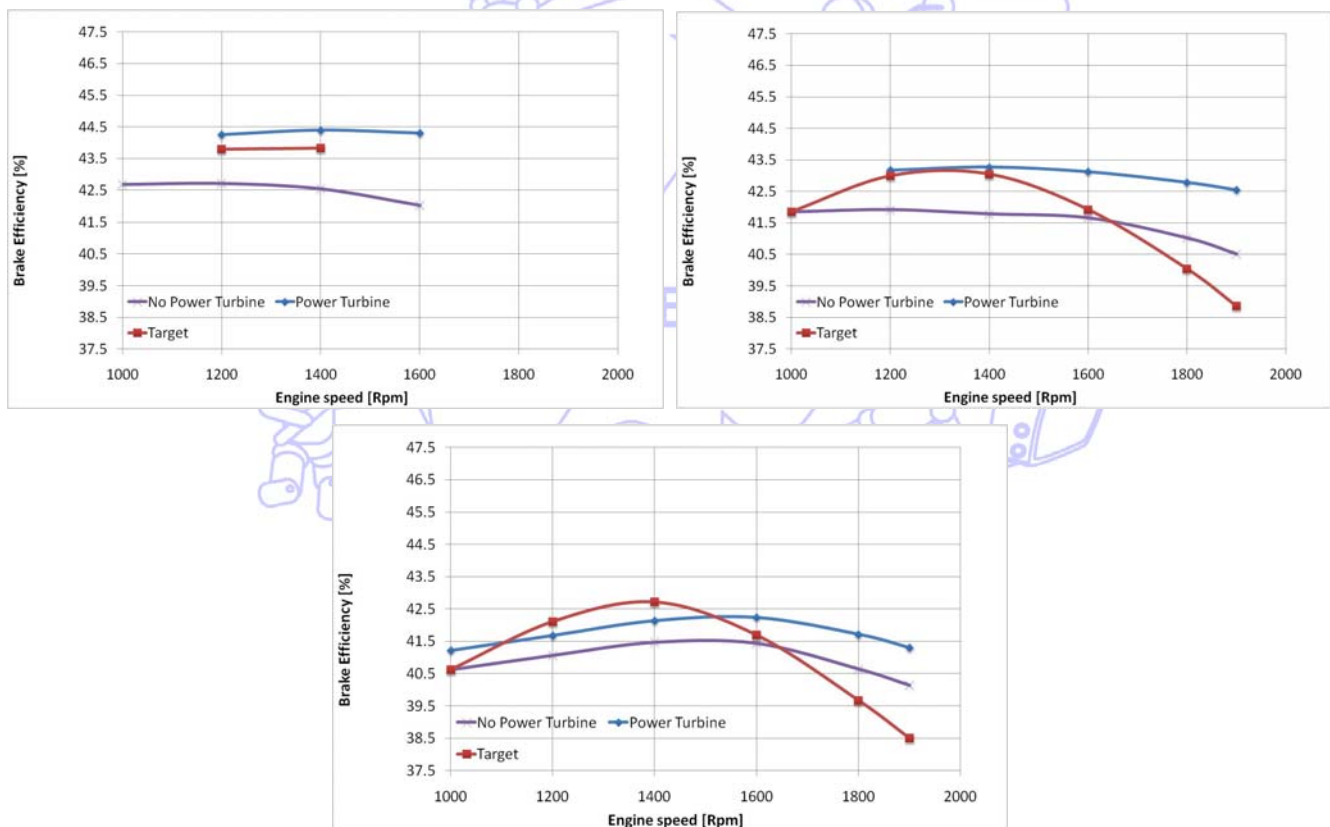
Fig. 6 – Brake efficiency at 23, 19 and 15 bar BMEP– Diesel engine. The target values are those of an existing heavy duty truck engine setting the benchmark in terms of efficiency and specific power for the category.

When exhaust gases pass through the turbine, the pressure and temperature drops as energy is extracted and because of the inevitable losses. The power taken from the exhaust gases with the power turbine is larger than without. To make this possible the pressure in the exhaust manifold is higher and this increases the pump work that the pistons have to do. The power turbine permits better efficiencies at high loads and high speeds, when the energy available downstream of the turbocharger turbine is significant, while at low to medium speed and low to medium loads the effect is opposite.

Operation at 10 bar BMEP with a power turbine downstream of the turbocharger turbine already show a negative trend,

with back pressure accounting more than the recovery of the limited energy available downstream of the power turbine. The after treatment may also be an issue in these conditions for the temperature reduction due to the expansion within the power turbines.

Decoupling the speed of rotation of the power turbine from the speed of the crankshaft and by passing the power turbine when it is not convenient for after treatment or fuel conversion efficiencies, turbo compounds may still be attractive. This option could make turbo compounds appealing also for passenger cars application, where so far almost all of the major Original Equipment Manufacturers (OEM) have tested this opportunity but decided not to move further. Additional benefits may arise from designing specific low pressure ratio turbines that would maximize the trade-off in between drop of pressure and temperature within the turbine and work extracted.

Cost, weight and complexity will certainly remain an issue of turbo compounds however shared with the most part of the techniques proposed to recover the exhaust energy, first of all those based on Organic Rankine Cycles (ORC).

The effectiveness of the power turbine over the speed and load range obviously depends on the selection of the turbocharger and the power turbine. The proposed turbocharger and power turbine have efficiency benefits at high loads. However, operation with the minimum permissible air-to-fuel ratio produces a decrease of power and efficiency at low speeds and an increase of power and efficiency at low speeds. This is due to the particular selection of the turbocharger and the power turbine. . The turbo compounding does not improve the power density, because the power density is actually larger without.

The Constant Variable Transmission (CVT) has some benefits, but obviously also some downfalls vs. a standard gear train. In principle a CVT has higher losses. However, as demonstrated by the F1 KERS [25], the increased frictional losses can be made small.
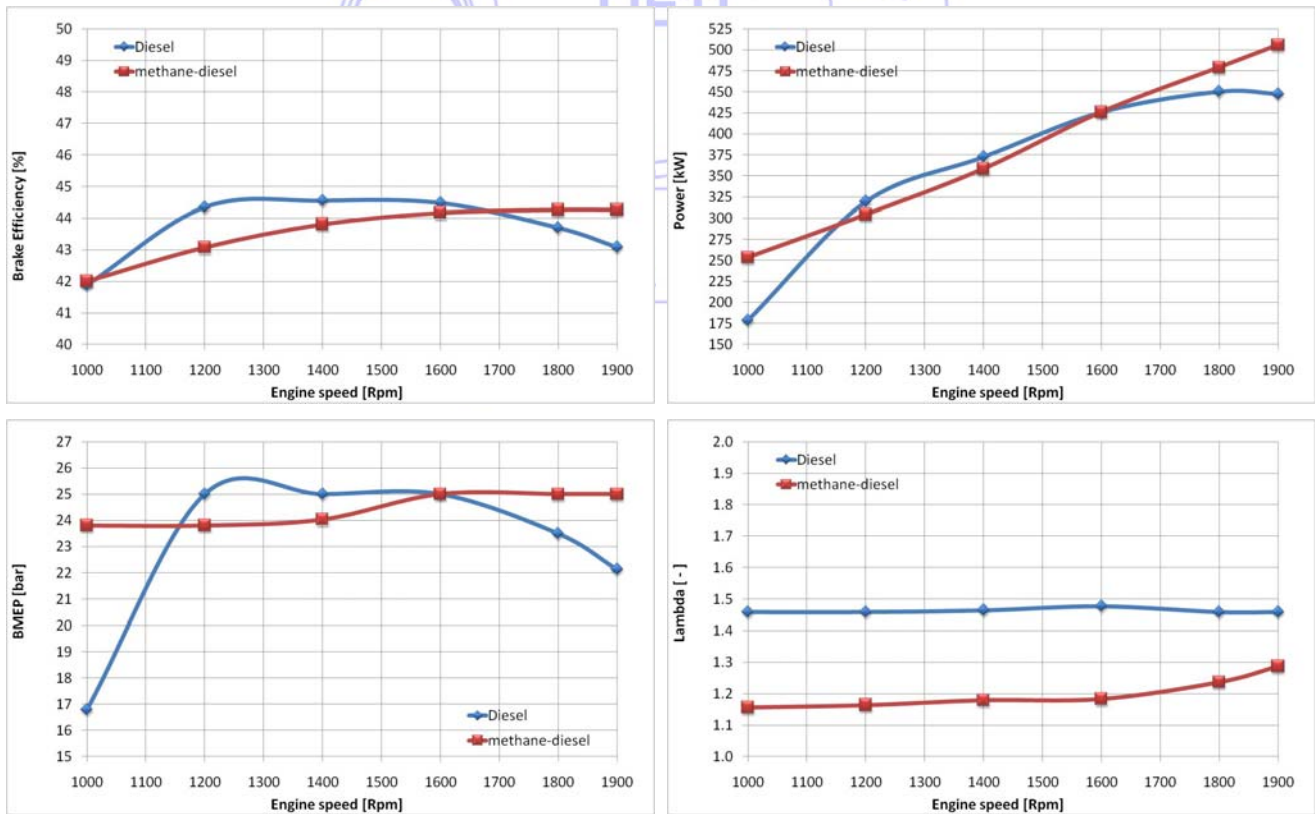

Fig. 7 – Full load brake efficiency, power and mean effective pressure and lambda (ratio of air-to-fuel operational to air-to-fuel stoichiometric) of the Diesel and the Diesel/CNG engines.

3.2.   *Diesel only or Diesel and CNG with power turbine*

As in the Diesel engine model, also in the CNG model only the main injection event is considered. In both cases, the details of the previous pilot and pre injection with the Diesel or the pilot Diesel with the CNG are neglected. Then, we suppose the same injection durations and the same combustion evolution will apply for the Diesel and the CNG.

Fig. 7 presents the full load lambda (ratio of air-to-fuel operational to air-to-fuel stoichiometric), brake efficiency, brake mean effective pressure (BMEP) and brake power. While a stoichiometric mixture of Diesel and air has an energy content of 2.82 MJ/kg of mixture, a stoichiometric mixture of CNG and air has an energy content of 2.76 MJ/kg of mixture 2.2% lower. The CNG engine has to run with a smaller lambda than the Diesel to achieve the target BMEP of 25 bar. A minimum value of lambda of 1.18 is used for the CNG engine, while a minimum value of 1.46 is used for the Diesel engine. This compensates for the much faster mixing of the CNG injected in gas phase and the therefore much quicker combustion that permits to lower the lambda value with CNG. It is at higher speeds that the CNG fuel permits to improve all the performances in both fuel conversion efficiency and power output. This is partly due to the presence of the power turbine. At medium speeds, the Diesel engine performs better than the CNG engine. Reducing lambda, the fuel conversion efficiency reduces because of the increased heat losses. The dual fuel Diesel and CNG engine permits close to Diesel full load brake efficiencies. The slightly lower values are the result mainly of the operation at lower lambda.

The dual fuel Diesel and CNG engine also permits better than Diesel BMEP at high speeds, and therefore a larger power output. In addition, CNG has better than Diesel carbon dioxide emissions, both tailpipe for the reduced amount of CO_2 per MJ of fuel resulting from the better carbon to hydrogen ratio, and better pollutant emissions, as very well known thanks to the Westport HPDI exercise that has produced many heavy duty truck Diesel engine conversions with measured reductions of particulate matter (PM) of 60% compared to Diesel.

Simulations are not repeated to analyse the effect of control parameters such as injection timing, EGR, valve timings (scavenging effects of turbo compounding), air-handling controls and others because this activity can be better realized experimentally rather than to use the propose model certainly not adequate to address all the design issues of a modern compression ignition engine.

The air and fuel flow rate and the different fuel lower heating value are not the only factors to explain the engine power. The model considers many phenomena that are affected by many others parameters. The influence of air and fuel flow rates and the different fuel lower heating value are simply those whose discussion is easier.

4.   **Discussion and conclusions**

Up to 20-25% of the fuel energy in a modern heavy duty diesel is exhausted, and by adding a power turbine in the exhaust flow, theoretically up to 20% of this exhaust energy can be recovered. In reality, only a minor part of this fuel energy can be practically recovered, first of all because of the added exhaust back pressure also increasing the pumping losses. Further limits to the use of a power turbine arise from the after treatment requiring temperatures above threshold values to operate efficiently. The increasing use of cooled EGR further limits the perspectives of this technique. Crank train coupling and power turbine add weight, complexity in design, control and service and costs, with a definitively negative trade-off in light load applications.

The additional cooling of the exhaust gases reduces the effectiveness of exhaust after treatment systems, requiring more active regenerations for particulate filter and reducing the time NOx control systems are effective. This is not an issue for the

proposed application where a by-pass system avoids the cooling for expansion in the power turbine except than when useful and without implications on the after treatment. The use of external cooled EGR is negative with a turbo compound because the exhaust energy decreases due to energy extracted into cooling system reducing the energy available to turbo charger and power turbines. Space requirements of turbo compound further constrain packaging of exhaust gas recirculation and turbochargers.

Simulations performed for a 12.8 Liter in-line six cylinder turbo charged directly injected Diesel engine have shown the opportunity to gain up to 2 percentage points in efficiency at high loads and speeds where the wasted energy is relatively large. At low engine loads the converted energy available downstream of the turbine is not enough to compensate for back pressure losses and the efficiency deteriorates. The turbo compound has therefore the advantage of improved fuel conversion efficiency in applications where high engine loads are dominating, but negative impact at light load. Also worth of mention the increase in maximum power output about maximum speed, where otherwise the power output sharply reduces. These improvements do not impact on the efficiency of the after treatment as well as on the use of cooled EGR to control the emissions, because the power turbine is by-passed when needed.

The use of a constant variable transmission to replace a gear ratio in between the power turbine and the crankshaft has the advantage of permitting operation of the power turbine within a narrow range of speeds decoupled from the speed of the crankshaft. The better efficiency of the power turbine translates in a better recovery of the exhaust energy. The idea of using a CVT and a by-pass is relatively novel and the model will certainly need further refinements. Prototyping of the solution will certainly help considerably in understanding the limits of the technology only partially addressed in this paper.

Dual fuel CNG and Diesel internal combustion engines may operate more efficiently and with increased power density adopting direct injection with high pressure and fast actuating injectors. The key area of concern for these engines is the further development of injectors able to independently inject the Diesel and the CNG fuels with high flow rates and speeds of actuation.

An equivalent single injection event may replace the multiple event injection of the latest Diesel-only engines for brake engine thermal efficiency simulations. The model set up for Diesel-only injection and combustion may then be used as a first approximation of the pilot Diesel and main CNG injection and combustion. Same combustion model parameters and time of start of injection and end of injection are used with Diesel only or with Diesel pilot and CNG main injections for same speed and percentage of load. Providing the volumetric flow rate is fast enough for the CNG injection, the differences in terms of combustion evolution should then be minimal.

The dual fuel CNG-Diesel internal combustion engine has top brake engine thermal efficiency approaching 45% as in the original Diesel-only engine and similarly reduced penalties in efficiency reducing the load. However, the novel dual fuel engine permits larger brake mean effective pressures thanks to the much closer to stoichiometric operation with CNG permitted by the injection of a gas rather than a mixture of liquid hydrocarbons and the then much easier mixing with air and combustion.

The opportunity to run the engine almost stoichiometric full load follows the availability not only of an extremely fast and high flow rate dual fuel injector, but also the adoption of a water injector to control the temperature of gases within the cylinder and to the turbine at higher loads.

The major advantages of the dual fuel CNG Diesel operation are the reduced CO_2 emissions, both tailpipe and fuel life

cycle analysis, for the better C-H ratio of the CNG fuel and the lower CO_2 emissions when producing and distributing the fuel, the reduced smoke and particulate emissions thanks to the gaseous fuel, and finally the better energy security. Running lower lambda because of the better mixture formation and combustion evolution properties of the CNG, the CNG engine may also deliver more power at high speeds. The dual fuel operation also permits acceptable driving ranges also in cases of a minimal refueling network for CNG, with CNG being the fuel used around capital cities and the Diesel being the fuel used elsewhere.

The injection system is the major area where to focus the research on these engines. A further developed dual fuel injector or the use of two off-the-shelf fuel specific injectors are enablers of novel modes of operation possibly delivering even better fuel efficiencies than the Diesel-like operation when preceding the CNG injection with a Diesel pre injection. These other modes may include homogeneous charge compression ignition directly injecting the CNG early during the compression stroke; gasoline-like operation injecting the Diesel after the CNG; and finally mixed Diesel-like and gasoline-like combustions performing simultaneously the two injections. Furthermore, operating the glow plug continuously may also permit the auto ignition of the CNG without any Diesel injection.

This paper presents simulation trends and the case to improve turbo compounding. The baseline engine model was delivering same performances of the production target engine in terms of power and brake specific fuel consumption. The model for the engine with a power turbine and for the engine operating Diesel-CNG are derived from this model using the same consolidated process where a single design variable is changed at the time and the results are guessed by simulations. The claims about the CVT follow a very solid logic, because it is very well known that a power turbine speed dictated by the crank shaft through a fixed gear ratio is much less efficient that the best speed permitted. The fact that turbo compounding improves efficiency is rather well known. The turbo compounding does not improve the power density, because the power density is actually larger without, and this is another interesting finding. It is not known the amount possible with an optimum speed power turbine as the one enabled here by the addition of the clutch and the CVT, and this is shown in the paper.
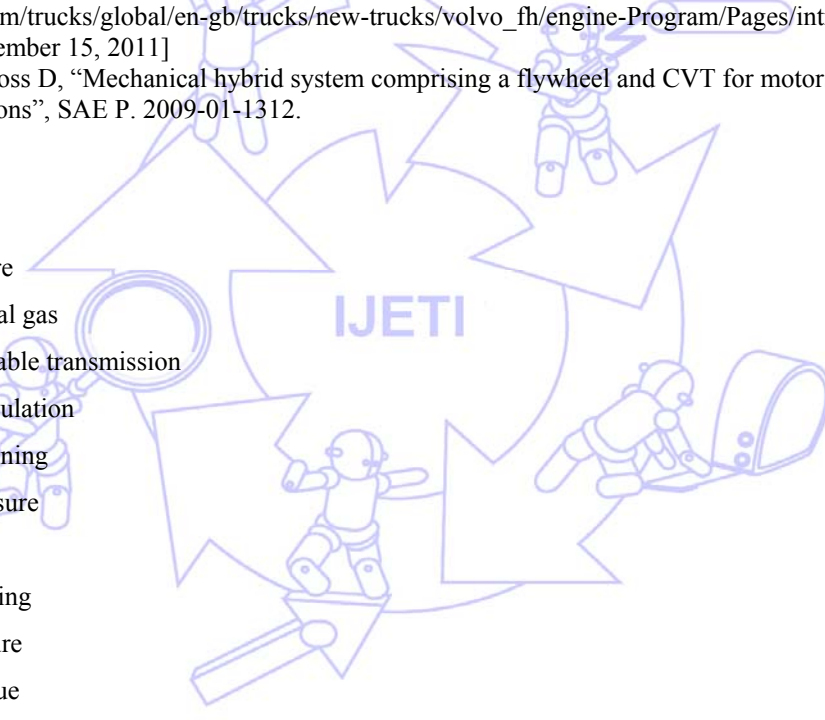
The further development of the concept that would also include the experimental testing to verify the results of Fig. 5 to 7 is subject to the funding of the activity.

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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

EVC   exhaust valve closure

HDT   heavy duty truck

IVO   intake valve opening

IVC   intake valve closure

LHV   lower heating value

LNG   liquefied natural gas

TC   turbocharged

TDC   top dead centre