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Half/Full Toroidal, Single/Double Roller, CVT Based Transmission for a Super-Turbo-Charger

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Abstract

Turbocharging dramatically improves the power density of internal combustion engines. However, when the energy to turbine is either smaller or larger than what is needed by the compressor for a specific steady state or transient point, there are downfalls, such as turbo-lag, lack of boost, or wasted exhaust energy. Here we propose a super-turbo-charger, where the turbo-charger shaft is connected to the crankshaft through a continuously variable transmission (CVT) and gears' pairs. Different designs are considered, half-toroidal and full-toroidal, single-roller or double-roller, single-CVT or two-CVT-in-series. Energy is drawn from the crankshaft or delivered to the crankshaft to improve the steady or transient operation. In this paper, a six-cylinder, dual fuel diesel injection, ignition engine is super-turbocharged. Engine performance simulations show the opportunity to achieve a nearly flat maximum torque all over the range of engine speeds and very high efficiencies from one fourth of the load up-wards.

Keywords: supercharging, turbocharging, dual fuel, diesel injection ignition, LNG

1. Introduction

Here we report the simulated performances of a dual fuel double direct injector diesel-LNG engine featuring a super-turbo charger [1] and [2]. While in a turbocharger the compressor energy is obtained from the exhaust gases, in a super-charger the compressor energy is extracted from the engine crankshaft. In the design proposed in [1] and [2], an over-sized turbo-charger is connected to the crankshaft through a continuously variable transmission (CVT) based transmission to provide a large variation of the speed ratio between the engine and the turbocharger. A clutch may also be added to also permit the balanced operation of the turbocharger decoupled from the speed of the engine when deemed appropriate [3].

The opportunity to run the turbocharger at a speed different from the equilibrium speed may improve steady and transient performances, either as fuel conversion efficiency and power/torque output. Power can be delivered to or drawn from the crankshaft. Improved power and torque are expected, especially at low speed. Additionally, the excess exhaust energy may be recovered at high speed. The super-turbo also helps with transient, either accelerations, removing the turbo-lag, or deceleration, permitting, in case of a vehicle equipped with a kinetic energy recovery system, also the recovery of the exhaust energy.

The optimum operating point of the engine and turbo-charger may differ from a point of balanced works of compressor and turbine, and the additional degree of freedom in the optimization is beneficial. As the extra work from the turbine can be collected at the crankshaft, the turbo-charger can be selected larger than in a traditional installation, because the crankshaft can collect the extra energy from the turbine. The control of the turbo-charger speed, and therefore of the flow of power to or from the crankshaft, and the pressure boost, is achieved by controlling the speed ratio across the mechanism. This is done through the CVT based mechanism. The CVT may be based on half-toroidal or full-toroidal design.

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Natural gas can be either turned into a drop-in diesel, liquid fuel by using Gas-To-Liquids (GTL) technologies or used compressed (CNG) or liquified (LNG) in a dual fuel conversion of the diesel engine. Either way, there are advantages for emissions. GTL diesel is colorless and odorless. It contains less impurities, mostly Sulphur, aromatics and nitrogen, than diesel from crude oil. In the GTL process, synthesis gas (hydrogen and carbon monoxide), is produced from natural gas by partial oxidation. Impurities are removed. Then the synthesis gas is turned into liquid hydrocarbons using a catalyst. Cracking and isomerization is finally applied to tune the molecular chains needed in the desired product.

The most successful GTL plant to date, produces a drop-in diesel fuel (GT-Fuel), plus kerosene, paraffin, various fluids and solvents, oils, naphtha, waxes and phase change materials [4]. This GTL Fuel consists mostly of straight chain normal-paraffins and branched iso-paraffins, has a much higher cetane number, a higher mass calorific value, lower levels of Sulphur and aromatics, and a lower density. These different properties may be used in a certain extent to reduce emissions of particulate matter, NOx, hydrocarbons and carbon monoxide. Test results have been promising, however, with benefits that are reducing in engines satisfying more strict emission standards.

The option to use CNG or LNG is more complicated for the engine and vehicle hardware and software, as major changes are needed. However, the advantages in terms of emissions may be even much larger, being natural gas much cleaner to burn than any one of the liquid fuels produced by refining crude oil. Westport has provided so far, the most brilliant and widespread dual fuel conversion of diesel heavy duty engines with their HPDI concept [5]. The HPDI is based on the direct injection of both the diesel and the LNG, with a pre/pilot injection of the diesel, and then a main injection of the LNG, from a same dual fuel injector. We proposed an enhanced version of the HPDI concept, where two injectors were used for every cylinder, one for the diesel, and one of the alternative fuel, to deliver improved combustion modes working on the relative timings of diesel and alternative fuel injection, and therefore on the premixed and diffusion combustion of the alternative fuel [6-9].

2. CVT Based Super-Turbo-Charger

The toroidal CVT based transmission here proposed is a variant of the one anticipated by Torotrack for their Variable-speed Supercharging Technology [10], Hu et al. [11]. The Torotrak V-Charge supercharger is a centrifugal compressor linked to the crankshaft through a CVT based mechanism. We use the toroidal CVT based transmission to connect the crankshaft to a turbine and compressor sharing same shaft rather than a compressor.

The Torotrack V-Charge comprises a traction drive variator and a traction drive epicyclic. The traction drive variator changes the drive ratio from 0.355: 1 (underdrive) up to 2.820: 1 (overdrive, 2.820 = 1/0.355). Before the traction drive variator, there is a 3:1 overdrive from the engine. After the traction drive variator, there is a 12.67:1 overdrive through the traction drive epicyclic. The CVT is full toroidal. The external overdrive is 312.67 = 38:1. The spread of the CVT speed ratios is 2.820/0.355 = 7.930.

Different variants are being considered to make the design simpler and better cover the speed ratios of turbocharger and engine. The total overdrive may be changed, as well as the splitting of the total overdrive before and after the traction drive toroidal CVT variator. By increasing the first overdrive, the traction drive toroidal CVT variator operates at higher speed and reduced torque for same power transmitted. The spread of the CVT speed ratios is the most difficult part to change. With a full-toroidal design, the spread of the CVT speed ratios is limited to around 8. With a half-toroidal design, the spread of the CVT speed ratios is limited to around 4.

Opposite to the gasoline engine application of [3], where both the engine and the turbocharger have large spreads of speed, in this case, where the spread of speed is reduced in both the engine and the turbocharger, a single full-toroidal CVT may be used. Opposite to [3] where the minimum speed is 1,000 rpm, but the maximum speed of the engine is 5,500 rpm and the maximum speed of the turbocharger is 200,000 rpm, in the application here considered the maximum speed of the engine is

4,500 rpm and the maximum speed of the turbocharger is 150,000 rpm. To achieve larger CVT speed ratios, [3] use two half-toroidal CVT in series, giving a CVT speed ratios of about 16, or a full-toroidal CVT giving a CVT speed ratio of about 8 plus a clutch. The clutch prevents the engine crankshaft to run the turbocharger at excessive speeds reducing the loads.

A CVT based crankshaft to turboshaft connection, with the CVT mechanism composed of two half-toroidal CVT in the series, is shown in Fig. 1. The CVT mechanism is composed of two half-toroidal CVT in the series. Alternatively, it is composed of one full-toroidal CVT. N_1 is the first overdrive. N_3 is the last overdrive. The CVT delivers a speed ratio from N_2:1 to 1/N_2:1. This CVT model is reproduced, modified after [12]. With one half-toroidal CVT, at the most N_2=2, for a spread of speeds ratio across the CVT of N_2^2=4. With two half-toroidal CVT in series, at the most N_2=4, for a spread of speeds ratio across the CVT of N_2^2=16. With one toroidal CVT, at the most N_2=2.828, for a spread of speeds ratio across the CVT of N_2^2=8.

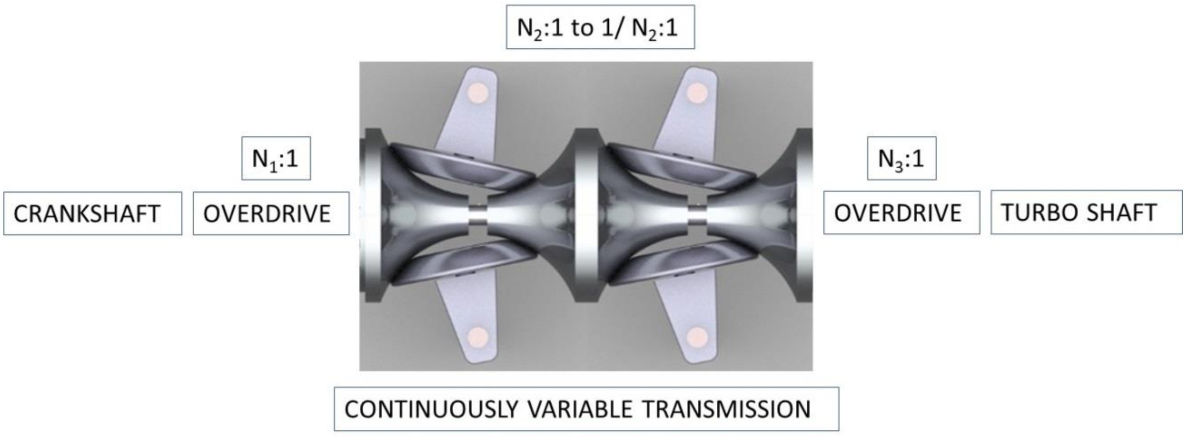


Fig. 1 CVT based crankshaft to turboshaft connection. (reproduced and modified after Kamel [12])

3. Efficiency of the CVT

After Applications of half-toroidal and full-toroidal single-roller CVT are covered in [10,13-14]. Design and analysis of half-toroidal and full-toroidal single-roller CVT are covered in Verbelen et al. [15], Zhanget al. [16], Zou et al. [17], Carbone et al. [18], Delkhosh et al. [19], Verbelen et al. [20] and Verbelen et al. [21]. While these designs are based on multiple single-roller, the opportunity to use multiple double-roller is considered in Carbone et al. [22], De Novellis et al. [23].

Verbelen et al. [20] discusses in a synthetic way the efficiency of a CVT of half-toroidal design. The efficiency η is expressed as a function of input speed, load torque, speed ratio and clamping force. The impact of clamping force can be eliminated if a proper slip controller is considered. Thus, the efficiency may be expressed as a function of speed ratio, load torque and input speed.

Verbelen et al. [20] show that after having computed (or measured) an efficiency map $\eta_{ref}(\tau, T_1)$, with τ the speed ratio across the CVT and T_1 the input torque, then there is a linear relation between efficiency and input speed. Hence, Verbelen et al. [20] then rescale the efficiency map computed at a certain input speed to a different input speed proportional to the difference of input speed.

In a half-toroidal CVT, power is transmitted from the input disc in the output disc through a system of rollers. The geometrical speed ratio, is the ratio of r_1 and r_2. The contact points can be changed by changing the tilting angle γ of the rollers. To avoid wear, a traction fluid is used. To limit the slip in the device, the clamping force F is applied at the input disc. This is shown in Fig. 2.

Verbelen et al. [20] model of a half-toroidal CVT by using the approach detailed in Carbone et al. [18] and Verbelen et al. [21] here briefly summarized. Only steady state results are considered.

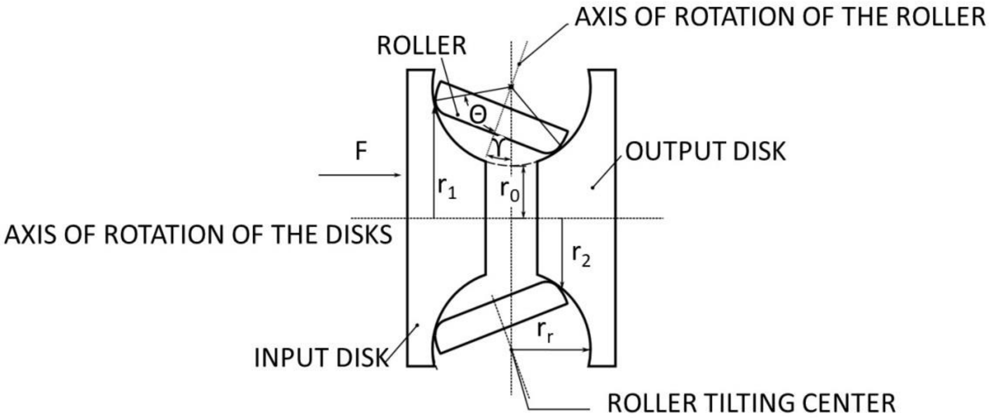


Fig. 2 Half-toroidal single-roller CVT geometry

The model consists of a contact model, a mechanical model, a simple load and a slip controller. In the contact model the traction conditions are determined based on the clamping force F, the tilting angle γ, the input speed N_1 and output speed N_2 of the CVT. The traction conditions are used to calculate the generated torque at the input T_1 and output T_2.

Slip is inevitable, needs control, and it can be calculated as:

$$\text{Slip} = 1 - \frac{\frac{N_1}{N_2}}{\frac{r_1}{r_2}} \tag{1}$$

The speed ratio range is defined by geometrical parameters and is limited by the input speed. Based on the geometrical parameters the maximum speed ratio is:

$$\tau_{\max} = \frac{1+\alpha}{1+\alpha-\cos\left(2\theta-\frac{\pi}{2}\right)} \tag{2}$$

where α is the aspect ratio, equal to r_0/r_r and θ is the half cone-angle. The minimum speed ratio is then the inverse of the maximum speed ratio, and the spread of speeds across the CVT is the maximum speed ratio squared. If the input speed increases, the roller speed increases. The heavy loading of the rollers limits the maximum permissible speed. The maximum load torque T_{max} then depends on the position of the roller γ which is related to the speed ratio τ, the maximum allowable pressure p_{max} imposed by the clamping force F and the geometrical parameters of the CVT. The steady state efficiency of the CVT is then:

$$\eta = \frac{N_2 \cdot T_2}{N_1 \cdot T_1} \tag{3}$$

This efficiency is a function of the four input parameters input speed, load torque, speed ratio and clamping force, but if unstable conditions are excluded from proper slip control through F, then only three parameters may be considered, input speed, load torque, speed ratio.

Verbelen et al. [20] proposes an efficiency map $\eta_{ref}(\tau, T_1)$ as a function of load torque T_1 and speed ratio τ, for a specific reference speed $N_{1,ref}$=3,000 rpm. Their efficiency increases with the load torque, and it reduces with the speed ratio, even if there is a limiting line on top of the operating range from the maximum allowable torque. Minimum efficiencies are found for load torque zero. Verbelen et al. [20] then observed a quasi linear relation between input speed and efficiency.

Their speed ratio τ vary between 0.5 and 2. Their maximum load torque is about 400 N·m. They propose to take:

$$\eta(\tau, T_1, N_1) = k(\tau, T_1)\cdot(N_1 - N_{ref}) + \eta_{ref}(\tau, T_1) \tag{4}$$

with N_{ref} the reference speed and η_{ref} the efficiency at this of reference speed. For a given speed ratio, the efficiency increases with speed, in addition to increase with the load torque.1

By using the approach of Verbelen et al. [20], we may then obtain an efficiency map at the reference speed characterized by high efficiencies 91-94% in the high load torque range for every speed ratio, but efficiencies 82-88% in the low load torque range for every speed ratio, and efficiencies as low as 60% for zero load torque. The efficiencies slightly increase with the speed ratio. These efficiencies, then slightly increase or reduce by increasing or decreasing the speed.

The efficiencies of Verbelen et al. [20] are referred to specific values of input torque T_1 and speed N, for a specific geometry, of giving dimensions. By changing the design of the CVT, dimensions, and input torque and speed, the efficiency values will change.

Discussing of the different design opportunities, we note that:

(1) Reduced efficiencies are obtained by using a full, rather than a half, toroidal CVT for the same spread of speeds ratio.

(2) Better efficiencies are obtained adopting a double-roller vs. a single-roller design.

(3) Efficiencies reduce with two CVT in the series, being the total efficiency the product of the two efficiencies.

(4) The AC load is inductive: R=64.81Ω; L=0.488H; Cos (φ) =0.38.

(5) For the specific application requiring a large spread of speeds ratio, the best option for what concerns efficiency is the full toroidal design, as two half-toroidal CVTs in series are needed to deliver same, a large spread of speeds ratio.

Because Verbelen et al. [20] is practically ignoring clamping control, they get very low efficiency figures when the module of the load torque reduces.

The efficiencies of Verbelen et al. [20] may generally be improved if the clamping force is managed to cause the traction coefficient to always sit around 0.07. In this case, the efficiency can be maintained above 94% for the double-roller variants, and above 88% for the single-roller variants, almost regardless of input speed and torque.

By assuming similarity between designs, i.e. the non-dimensional maps:

$$\mathrm{x} = \eta_{ref}\left(\frac{\tau}{\tau_{max}-\tau_{min}}, \frac{T_1}{T_{1,max}}\right) \tag{5}$$

$$y = \frac{\mathrm{k}\left(\frac{\tau}{\tau_{max}-\tau_{min}}, \frac{T_1}{T_{1,max}}\right)}{N_{1,ref}} \tag{6}$$

are about the same, apart from a multiplying technology factor χ larger than unity accounting for the full-toroidal and double-roller advantages over half-toroidal and single -roller designs, following Verbelen et al. [20], we may define for the single-roller, full-toroidal CVT the efficiency map $\eta_{ref}(\tau, T_1)$ of Fig. 3, and the speed correction factor map $k(\tau, T_1)$ of Fig. 4.

For sake of simplicity, we also assume that same η apply for transmission of power to, and from, the super-turbocharger, obviously with entry conditions on one side or the other of the single-roller full-toroidal CVT.

The proposed efficiency map is only preliminary. While a good design with enhanced clamping control may certainly deliver an area of operating points well, will above 90%, albeit with efficiencies still dropping below 90% when seal and parasitic losses will dramatically increase at very low torques, this novel application of the CVT certainly needs specific research and development with prototyping and testing a critical aspect. The design, and consequently the efficiency map, is very application specific, not generic.

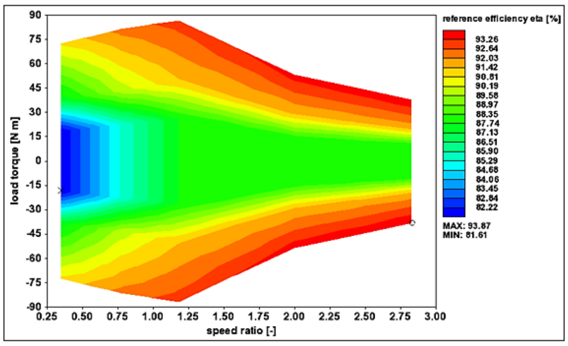


Fig. 3 Reference efficiency map $\eta_{ref}(\tau, T_1)$ of the CVT based transmission

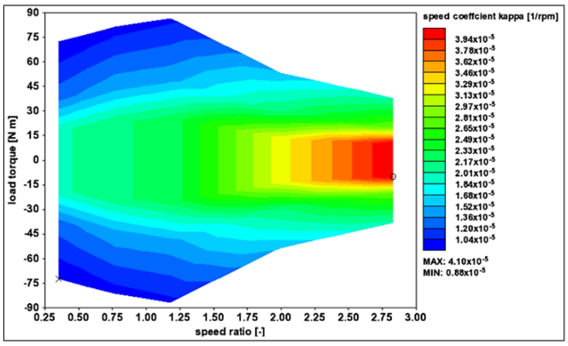


Fig. 4 Speed coefficient map $k(\tau, T_1)$ of the CVT based transmission

4. Performance of the Engine

We consider here a total external overdrive of 37.6:1, and a single full-toroidal CVT of overdrive 2.820 and underdrive 0.355 = 1/2.820, for a spread of the CVT speed ratios 2.820/0.355 = 7.930. Opposite to Boretti and Castelletto [3], a clutch is not essential with a full-toroidal CVT.

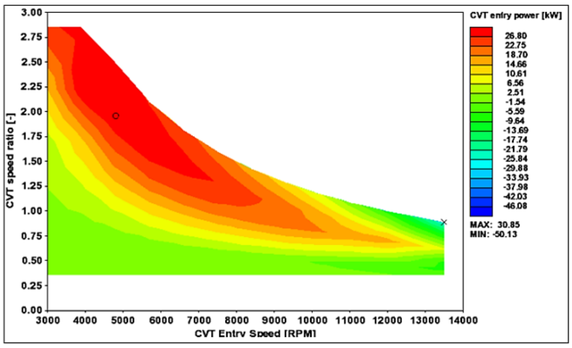


Fig. 5 CVT entry power in kW vs. CVT entry speed in rpm and CVT speed ratio

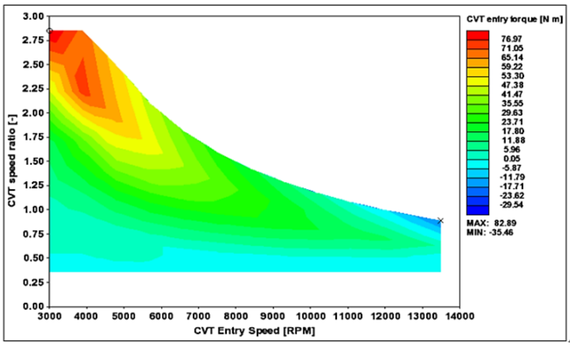


Fig. 6 CVT entry torque in N·m vs. CVT entry speed in rpm and CVT speed ratio

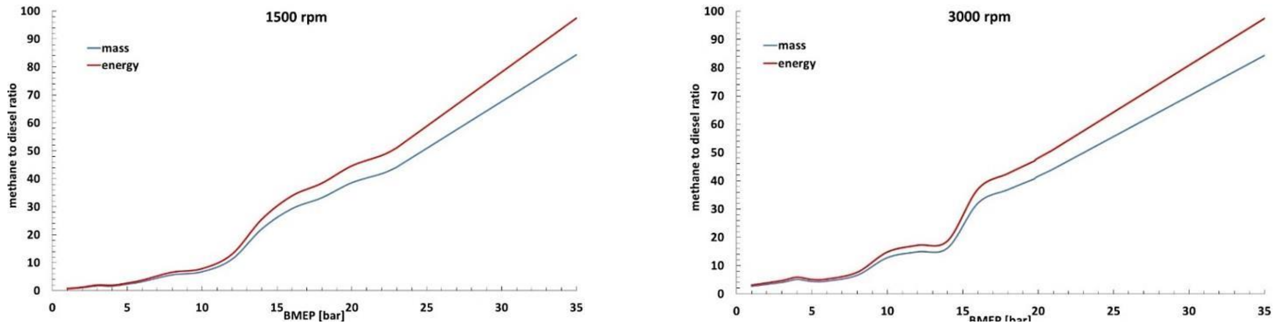
Fig. 5 presents the CVT entry power, side of the crankshaft, versus the CVT entry speed, side of the crankshaft, and the CVT speed ratio. The power is in kW and the speed is in rpm. External overdrive is 37.6:1. The spread of CVT speed ratios is 2.8202 = 7.93. There is only one single-roller, full-toroidal CVT. The Overdrive before the CVT is 3:1. The power is positive if delivered to the CVT (extra power to run the compressor). The overdrive before the CVT is 3:1, as it in the design proposed by Torotrack for the V-charge. Fig. 6 presents the CVT entry torque, side of crankshaft, versus the CVT entry speed, side of the crankshaft, and the CVT speed ratio. The torque is in N·m and the speed is in rpm. In the engine performance simulations, the efficiency of the CVT transmission, only weights on the computation of the power to and from the turbocharger, i.e. only on the difference in between the compressor and turbine powers, when these two parameters are unbalanced.

We only consider the steady operation of a turbocharged diesel injection, ignition dual fuel diesel-LNG engine featuring this super-turbocharger in very preliminary results. The transient operation is not considered, but the turbo-lag is expected to be completely cancelled.

The diesel is assumed to have carbon atoms per molecule 13.50, hydrogen atoms per molecule 23.60, LHV of 43.25 MJ/kg. The LNG (methane) is assumed to have carbon atoms per molecule 1, hydrogen atoms per molecule 4, and LHV of 50 MJ/kg.

The model is of a dual-fuel, diesel - LNG injection ignition engine, with each cylinder supplied with two independent injectors for the diesel and the LNG. The turbo-charger is oversized. Also oversized are the port areas and the valves' diameters and lifts. The turbo-charger is intended to be used for a maximum speed-times-displacement that is 30% larger than the present 4,500 rpm-times-3.8 liters.

Maximum speed of the turbo-charger is 150,000 rpm. Minimum speed is 13,000 rpm. Minimum speed of the engine is 1,000 rpm and maximum speed 4,500 rpm.



(a) at 1,500 rpm

(b) at 3,000 rpm

(c) at 4,500 rpm

(d) at different speeds

Fig. 7 LNG to diesel ratio by mass and by energy as a function of the BMEP in bar

Fig. 7 finally presents the ratio of LNG (methane) to diesel by mass or by energy as a function of the BMEP in bar at different speeds. The diesel injected is basically the fuel needed in the diesel only operation during the pilot and pre-injections, while the LNG injected is basically the fuel needed in the main injection during the diesel only operation.

At minimum loads, the pilot and pre-injection diesel represents the most part of the fuel. At higher loads, the main-injection LNG dramatically increases.

While the C/H ratio of the diesel is 13.50/23.60 = 0.57, the C/H ratio of the LNG is 0.25. Hence, because of the LPG fuel replacement of the main injection diesel, the equivalent C/H ratio of the fuel changes with the load from about 0.56 at minimum loads to about 0.30 at maximum loads. Hence, approaching top loads, the CO_2 emission is almost halved. Similarly, at top loads, the PM production within the cylinder is expected to be minimal, as it should reduce more than proportionally to the diesel fuel replacement by mass.

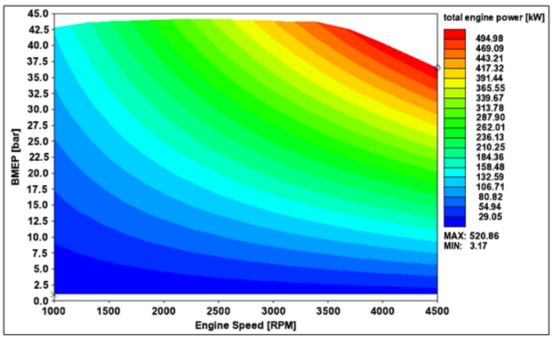


Fig. 8 Total engine power in kW, as a function of engine speed and brake mean effective pressure (BMEP) in bar

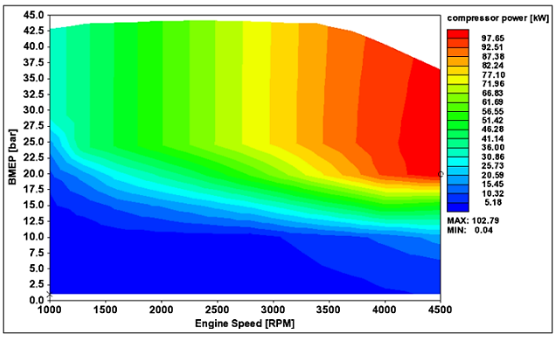


Fig. 9 Compressor power in kW, as a function of engine speed and brake mean effective pressure (BMEP) in bar

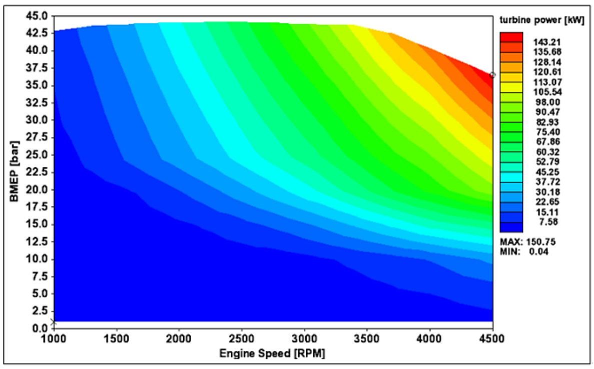


Fig. 10 Turbine power in kW, as a function of engine speed and brake mean effective pressure (BMEP) in bar

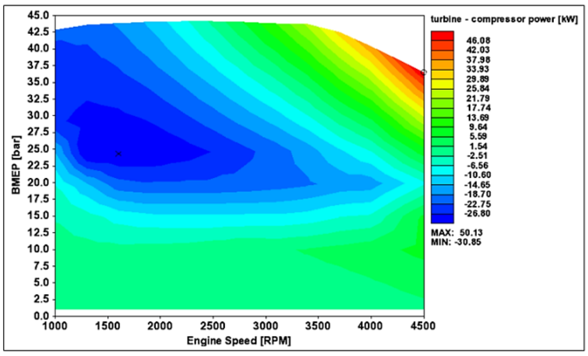


Fig. 11 Difference between turbine and compressor power in kW, as a function of engine speed and brake mean effective pressure (BMEP) in bar

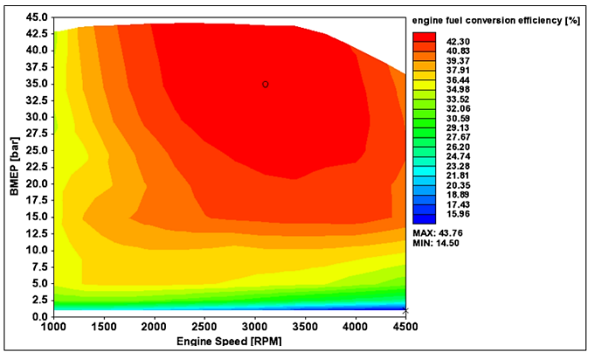


Fig. 12 Fuel conversion efficiency η^*, ratio of the total power to the fuel flow power, in %, vs. engine speed in rpm and brake mean effective pressure (BMEP) in bar

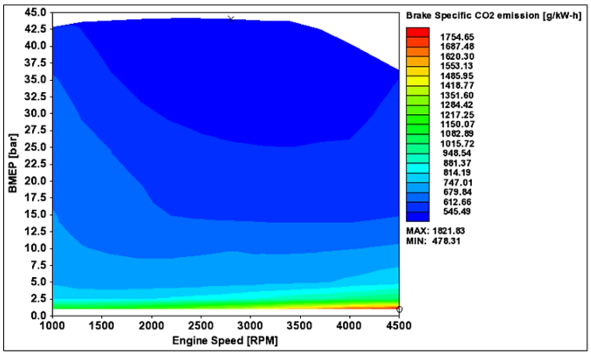


Fig. 13 Brake Specific CO_2 emission, in g/kW-h, vs. engine speed in rpm and brake mean effective pressure (BMEP) in bar

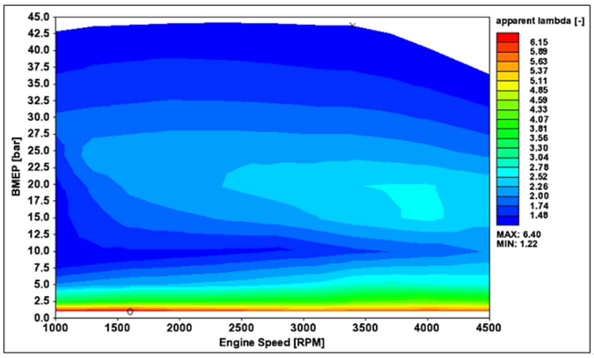


Fig. 14 Apparent equivalent air-to-fuel ratio λ vs. engine speed in rpm and brake mean effective pressure (BMEP) in bar

Figs. 8 to 11 present the total engine power, compressor power, turbine power, and difference between turbine and compressor power in kW, as a function of engine speed in rpm and brake mean effective pressure (BMEP) in bar.

Fig. 12 presents the fuel conversion efficiency η^*, now ratio of the total power, from the pistons and the turbocharger at the crankshaft, to the fuel flow power, vs. the speed of the engine in rpm and the brake mean effective pressure (BMEP) in bar.

Fig. 13 presents the Brake Specific CO_2 emission, in g/kW-h, vs. engine speed in rpm and brake mean effective pressure (BMEP) in bar. Figure 14 finally presents the apparent equivalent air-to-fuel ratio λ vs. engine speed in rpm and brake mean effective pressure (BMEP) in bar.

The turbo-charger continues to receive energy at low speeds, mid-high loads, while at high speeds, mid-high loads, the turbocharger usually provides energy. The low speed torque (BMEP) is increased up to the medium speed values. With the super turbocharger, the maximum torque and the maximum power are both increased.

The efficiency is about constant 2,000 to 4,000 rpm, and one third to full load, only slightly but mostly improved vs. the baseline turbocharged configuration. At a maximum load, the engine runs lean, however with λ values much smaller than in a diesel. This may require some LNG to be introduced before the diesel is injected. Above 4,000 rpm, the efficiency reduces mostly because of the combustion becoming more difficult.

5. Discussion

The Torotrak V-Charge super-charger was a production-ready CVT based supercharger. The technology was developed to the point that the concept was proven and ready to commercialization after prototyping and testing in laboratory experiments and on the road [11, 24-26]. The V-Charge was installed on a small Ford Focus Titanium 1.0T Ecoboost. Power was risen from 92 to 118 kW and torque was up from 169 to 249 N m. The improvements in fuel economy were significant, rated at 12% better fuel economy versus the naturally aspirated larger engine. Only the recently announced freeze in the development of new internal combustion engines across the world, due to the possible dismissal of the internal combustion engine in favor of the electric mobility promote by European lawmakers within two decades, has halted the project. The proposed innovation differs from the V-Charge only for the connection to a turbocharger rather than a compressor. The additional packaging issues are minimal, and the system may well fit the engine compartment of even small cars such as the Ford Focus, as it was done for the V-Charge.

The dynamic model of the mechanical system, as well as of the engine, is missing. In this preliminary phase, we only consider the steady state operation. While the dynamic models will be needed in a better proof of concept and obviously product development, the static model is here used to highlight some of the advantages of the proposed innovation. The proposed efficiency map is a reference map obtained from load and speed extrapolation from an existing design. As the efficiency of the CVT only weights on the difference between turbine and compressor power, the influence on engine efficiency of small variations in CVT efficiency are negligible.

Only a steady state model of the CVT is used. While it is correct to assume that the efficiency depends on input speed, load torque and speed ratio, however, during transient operation, there is always an increase in slip and therefore a drop-in efficiency.

6. Conclusions

This work follows the prior works by the author on dual fuel diesel injection, ignition with traditional turbochargers, and the super turbocharging of direct injection diesel engines and direct injection jet ignition gasoline engines.

The super turbocharger connecting the turbocharger shaft to the crankshaft through a CVT and a gear is an option to improve both fuel conversion efficiency and power/torque output, either steady state and transient.

It is expected that the proposed innovation will drastically improve the transient operation cancellation of the turbo-lag. There is no reason why the proposed innovation should not deliver same improvements, under this criterion, of F1 turbocharged engines with MGU-H, or the Ford Focus Titanium 1.0T Ecoboost with the V-Charge.

The preliminary results proposed here are specific for a dual fuel diesel-LNG diesel injection ignition engine. The engine delivers high boost at any speed, plus high fuel conversion efficiency, above 40%, over the most part of the load-speed map.

Albeit the results are qualitative, the trends are trustworthy, as demonstrated by the latest F1 experience, where a turbo-charger similarly operated through a motor-generator unit connected to the turbo-charger shaft, and the use of direct injection and jet ignition, translate into strongly improved fuel economies running lean of stoichiometry stratified with minimal cyclic variability, no turbo-lag, and dramatically improved low speed torque.

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This preliminary simulation work was performed in 2013/2014 to understand the areas where to focus further research and development in F1 and Le Mans LMP1-H powertrains, within the limits of the rules of the time, and on the hypothesis of relaxed constraints. The use of high pressure (500 bar) direct injection and jet ignition, coupled to a carefully redesigned, oversized, turbocharger connected to a motor-generator unit (MGU), was the proposition for F1 gasoline applications, as well as for LMP1-H gasoline powertrains. Similarly, an oversized turbocharger fitted to a motor-generator unit and extremely high pressure direct injection (3000 bar) was the suggestion for LMP1-H diesel powertrains. The author received no funding for this work.

Conflicts of Interest

The author declares no conflict of interest.

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