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Optimum Speed Power Turbine to Recover the Exhaust Energy of Compression Ignition Diesel and Gas Engines

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Abstract. The efficiency of internal combustion engines may be improved recovering the fuel energy loss in the exhaust gases or the coolant that is the predominant portion of the fuel energy, being the amount of fuel energy transformed in mechanical energy usually less than 50% in the top efficiency operating points and well below that figure in the other operating points of the load and speed map. This paper consider the opportunity to recover part of the exhaust energy with a power turbine that is operational only when convenient and it is run at optimum speed thanks to a by-pass and a continuously variable transmission link to the crankshaft. The power turbine operates at optimum speed only when producing more power than the power loss for back pressure while permitting temperatures to the downstream after treatment system high enough. Simulations for a 12.8 liters straight 6-cylinder Diesel engine with turbocharger and intercooler show improvements in both the fuel conversion efficiency at medium-to-high speeds and medium-to-high loads.

Keywords: Diesel engine, exhaust energy recovery, power turbine, continuously variable transmission

1 Introduction

The primary limiting factors to approach the theoretical efficiency limits of internal combustion engines (ICE) are the high irreversibility in traditional premixed or diffusion flames, the heat losses during combustion and expansion, the untapped exhaust energy, and the mechanical friction. Two areas where there is large potential for improvements in ICE efficiency are especially the losses from the exhaust gases and the heat transfer losses. The amount of fuel energy in the exhaust gases and in the coolant is a significant percentage of the fuel energy, more than one half and above, and recovering this fuel energy loss is crucial to improve the ICE efficiency.

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%[12, 13]. 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.

The use of turbo compounding goes back quite a long way. The concept was originally used back in the late 1940s and 50s on two notable aircraft engines[8, 10, 14, 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[3–5, 7, 9, 11, 15, 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

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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 behaviour 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.

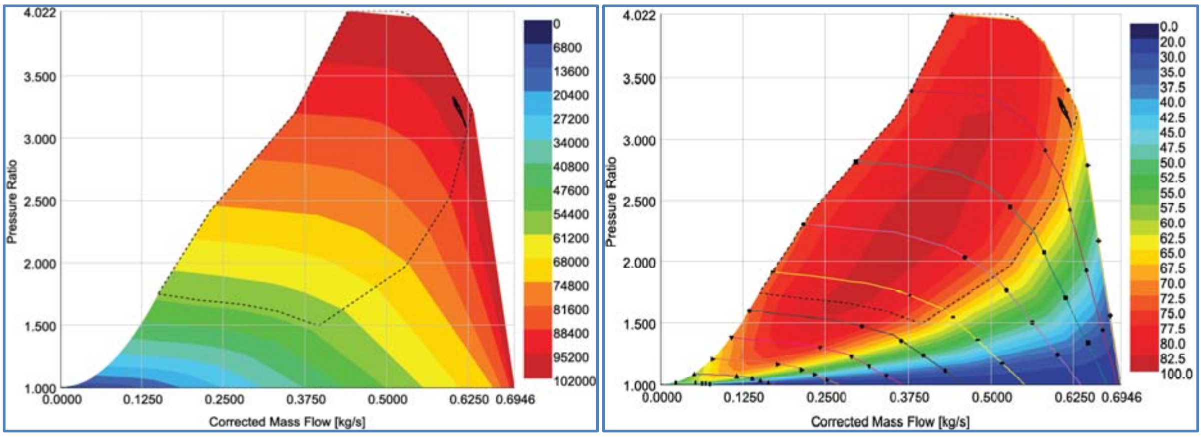

Figure 1 - Maps of the compressor. Left to right: compressor corrected speed and efficiency vs. pressure ratio and corrected mass flow rate.

Fig. 1: Maps of the compressor. Left to right: compressor corrected speed and efficiency vs. pressure ratio and corrected mass flow rate

Fig. 1 presents one of the most successful applications of turbo compound technology to a truck engine (from [9]). 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%.

Despite few truck manufacturers claim better efficiency and increased power output as a result of the additional power turbine, this solution has not encountered so far the favour 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 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. A sketch of the proposed design is presented in Fig. 2. 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. 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.

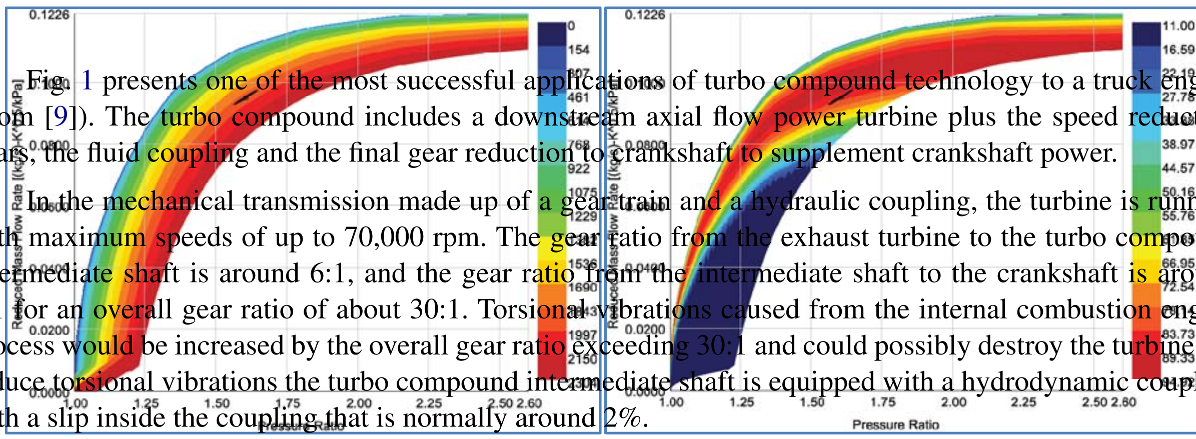

Figure 2 - Maps of the turbo charger turbine. Left to right: turbo charger turbine speed and efficiency vs. pressure ratio and corrected mass flow rate.

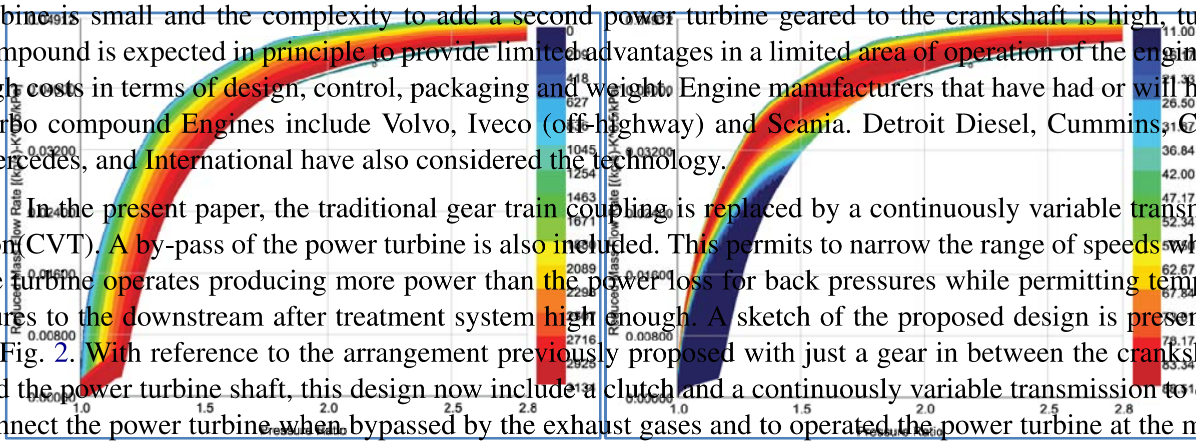

Figure 3 - Maps of the power turbine. Left to right: power turbine speed and efficiency vs. pressure ratio and corrected mass flow rate.

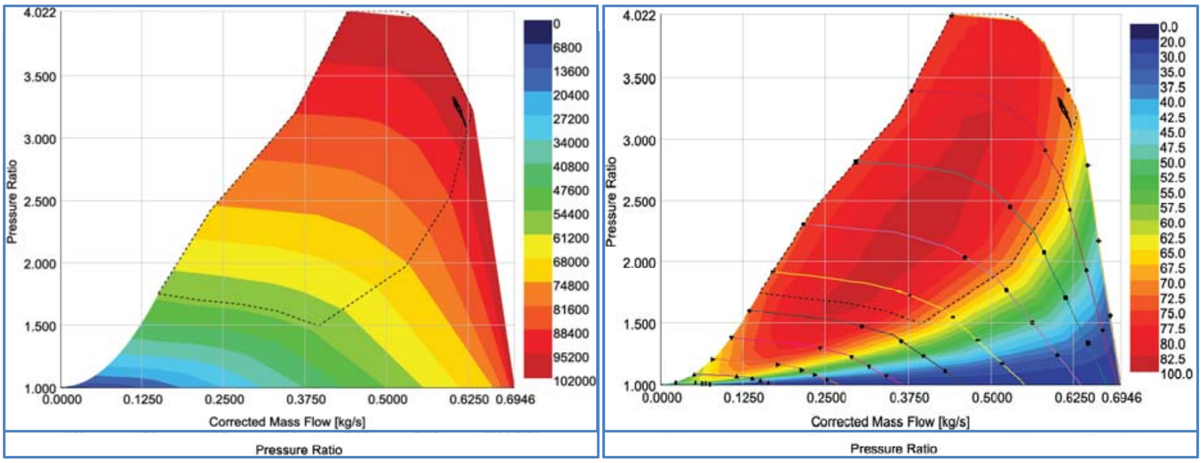

Fig. 2. Maps of the turbo charger turbine. Left to right: turbo charger turbine speed and efficiency vs. pressure ratio and corrected mass flow rate

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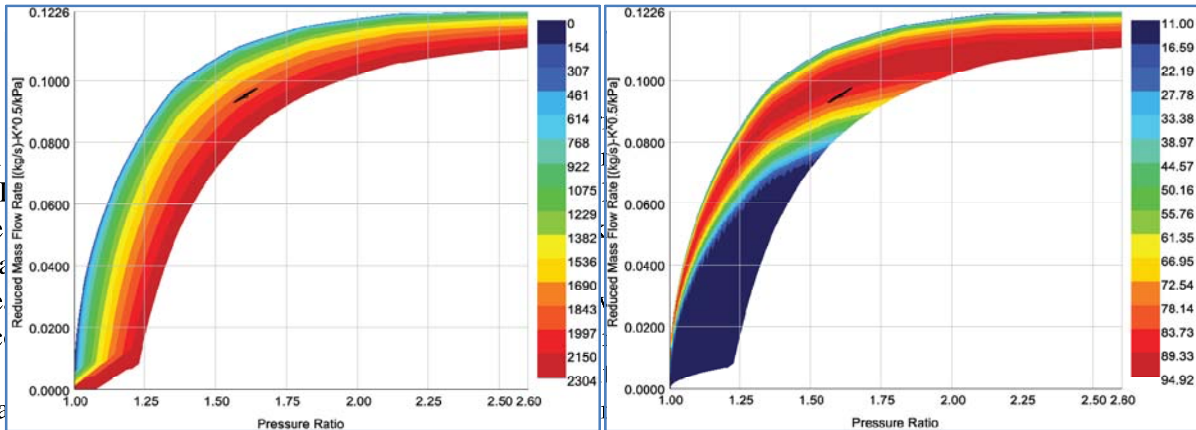


power loss for the back pressures while permitting temperatures to the downstream after treatment system high enough. The CVT permits operation of the power turbine at the more efficient speeds.

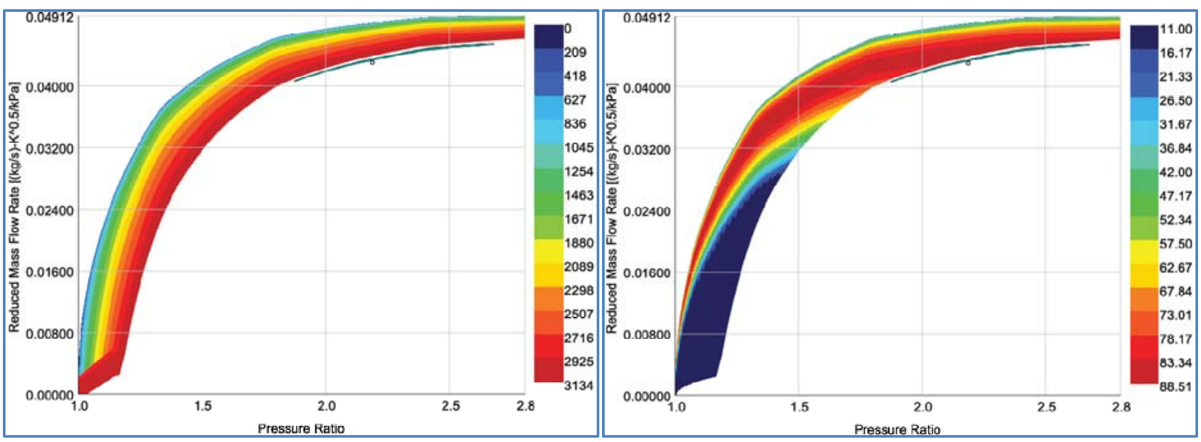


Fig. 3. Maps of the power turbine. Left to right: power turbine speed and efficiency vs. pressure ratio and corrected mass flow rate

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 litre, and up to 3,150 Nm of torque and 515 kW of power the 16 litre[2]). 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 high speeds will be otherwise much higher than those of medium to low loads and speeds.

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.

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.

Table 1. Litre in-line six cylinder turbo charged directly injected Diesel engine

Displacement per cylinder [liter]	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 dia. [mm]	46.3
Intake valve maximum lift [mm]	12.1
number of exhaust valve per cylinder	2
IVO [deg]	344(−16)
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 Continuously 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 [17], the increased frictional losses can be made small.

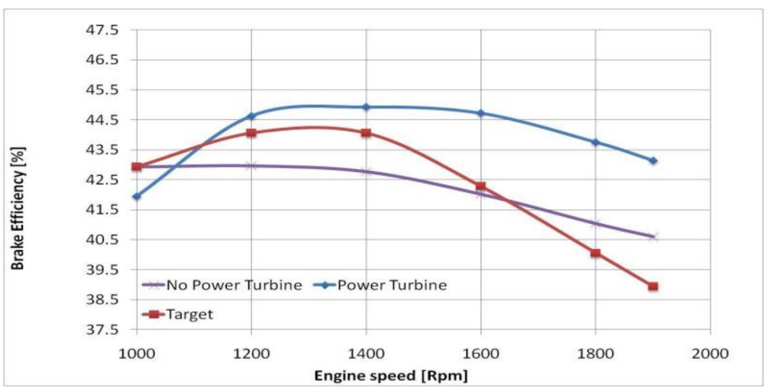


Figure 4 – Full load brake efficiency– Diesel engine.

Fig. 4. Full load brake efficiency-Diesel engine

The present analysis if performed by using one first turbo charging unit, where the turbine drive the compressor, and then a second power turbine connected to the crankshaft through a continuously variable transmission permitting the operation (when not by-passed) with optimum speed. The turbo charging unit is not optimal for the operation without the power turbine. Both the turbo charging unit and the power turbine have not been optimised for the specific engine design. The engine has an external cooled exhaust gas recirculation (EGR) to control the emissions, but to keep the comparison straight in between operation with and without power turbine the cooled EGR is not considered. The original Diesel engine has a multi event injection strategy made of pilot, pre, main and post injections. For the purposes of computing the fuel economy (but obviously not the pollutant formation) the complex injection strategy is replaced by an equivalent single injection. The equivalent single injection parameters, start of injection and duration, are tuned to reproduce in the model the experimental peak pressure and brake mean effective pressure of a reference engine. The equivalent single injection parameters are obviously dependent on the heat release and the heat transfer models. The models adopted are non-predictive, correlative models[1], Wiebe [17] type functions to describe combustion, and Woschni [19] type formulation for heat transfer. The Wiebe combustion model is obviously linked to the injection parameters. Equivalent single injection parameters and the Wiebe combustion parameters are set up for every operating point of the BMEP and speed map modelling the target Diesel engine of known experimental behaviour. It is then assumed that same BMEP and speed point with the novel turbocharger and the addition when needed of the power turbine is obtained with same start of equivalent injection and end of equivalent injection (the amount of fuel is obviously free to change) and same Wiebe function constants.

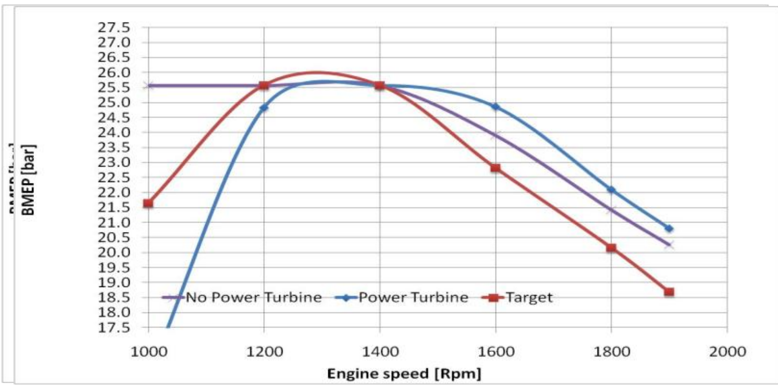


Figure 5. Full load BMEP – Diesel engine.

Fig. 5. Full load BMEP-Diesel engine

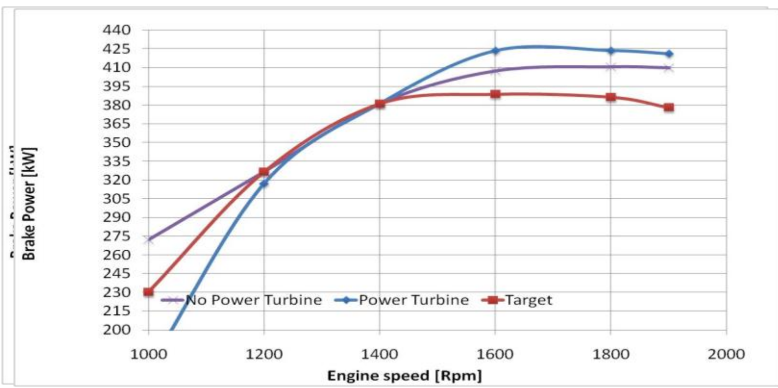


Figure 6. Full load Power – Diesel engine.

Fig. 6. Full load Power-Diesel engine

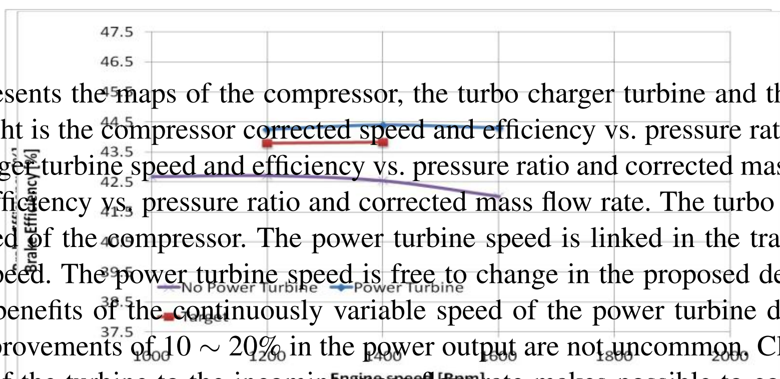
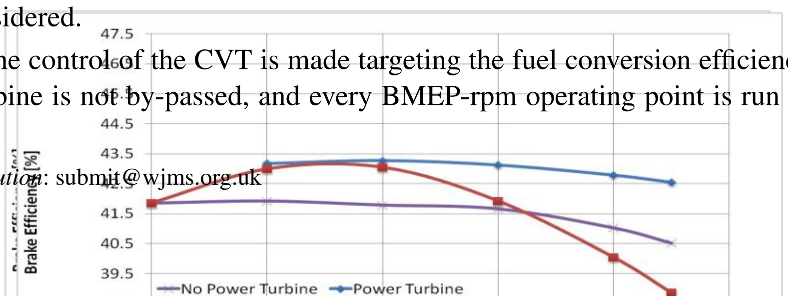


Figure 7. Brake efficiency at 23 bar BMEP – Diesel engine.

Figs. 1 ∼ 3 presents the maps of the compressor, the turbo charger turbine and the power turbine. Top to bottom, left to right is the compressor corrected speed and efficiency vs. pressure ratio and corrected mass flow rate, turbo charger turbine speed and efficiency vs. pressure ratio and corrected mass flow rate and power turbine speed and efficiency vs. pressure ratio and corrected mass flow rate. The turbo charger turbine speed is linked to the speed of the compressor. The power turbine speed is linked in the traditional configuration to the crank shaft speed. The power turbine speed is free to change in the proposed design to maximize the power output. The benefits of the continuously variable speed of the power turbine depend on the specific operating point. Improvements of 10 ∼ 20% in the power output are not uncommon. Clearly, the opportunity to adapt the speed of the turbine to the incoming mass flow rate makes possible to completely redesign the power turbine blades for a better efficiency over the smaller range of incoming flow directions, but this option is not presently considered.

In the model, the control of the CVT is made targeting the fuel conversion efficiency for a given BMEP when the power turbine is not by-passed, and every BMEP-rpm operating point is run with and without the



power turbine. Obvic … CVT controlled speed ratio may be directly …

Figure 5 – Full load BMEP – Diesel engine.

2.1 Diesel with an …

Figs. 4 ∼ 6 pres … ts the brake efficiency operating at 23 bar … es not help too much to increase the torqu … efore effective for this criterion only over a … This is because using the exhaust energy i … ithin the cylinder that translates in much m … . However, sometimes the turbocharger turl … d and maximum load operating point beca … ds range, and there is therefore space for fu …

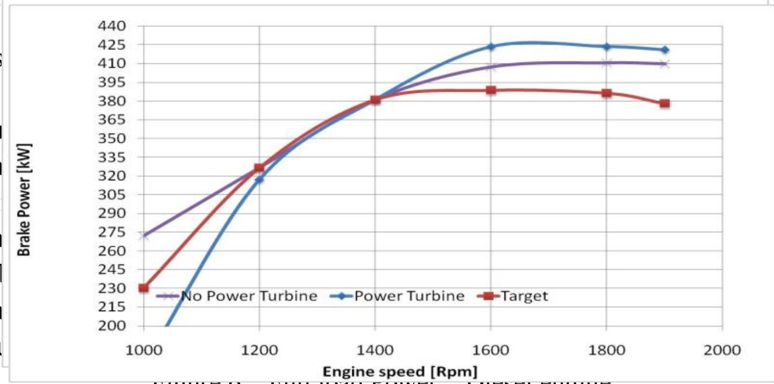


Figure 6 – Full load Power – Diesel engine.

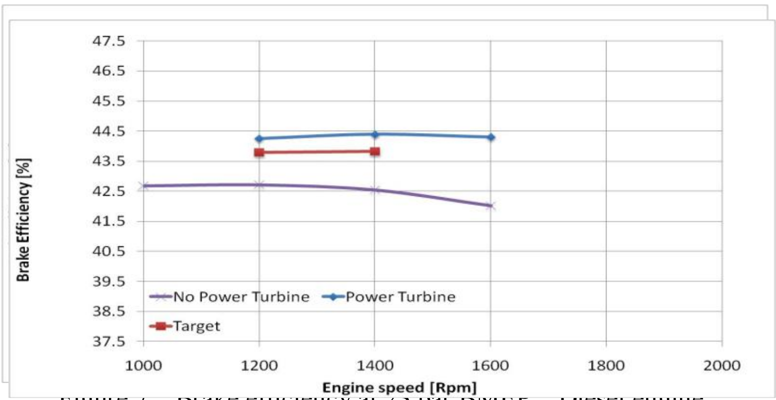


Figure 7 – Brake efficiency at 23 bar BMEP – Diesel engine.

Fig. 7. Brake efficiency at 23 bar BMEP-Diesel engine

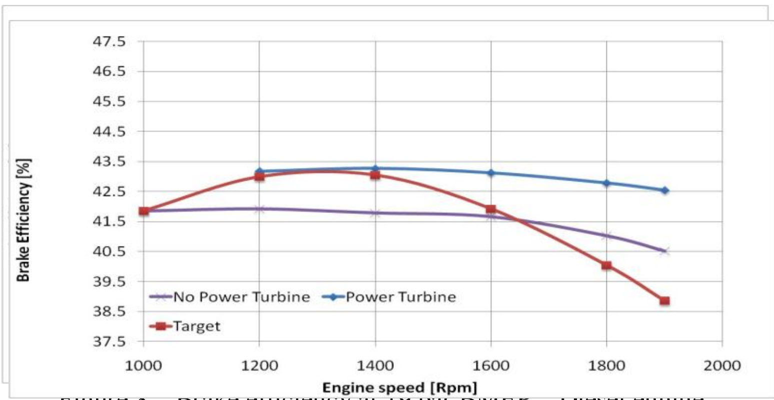


Figure 8 – Brake efficiency at 19 bar BMEP – Diesel engine.

Fig. 8. Brake efficiency at 19 bar BMEP-Diesel engine

The power turbine helps considerably in increasing the efficiency, with increases of almost 4 percentage points in the brake efficiency. These figures are slightly better than those already provided by other researchers. The reason is the operation of the power turbine in a better speed range thanks to the CVT.

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.

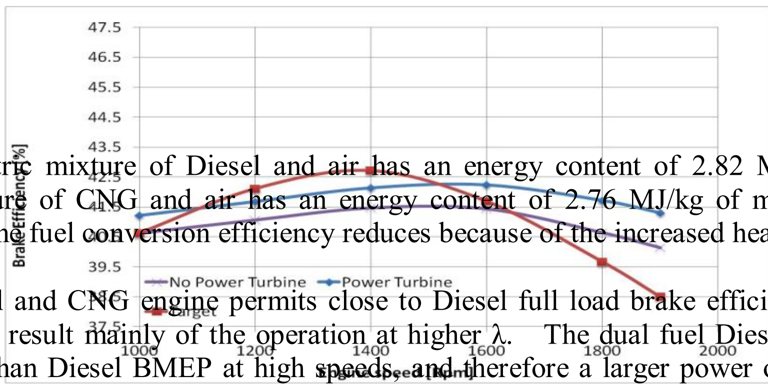
When exhaust gases pass through the turbine, the pressure and temperature drops as energy is extracted and because of the inevitable losses. The power taken from the exhaust gases with the power turbine is larger than without. To make this possible the pressure in the exhaust manifold is higher and this 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.

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 Continuously 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 [6], the increased frictional losses can be made small.



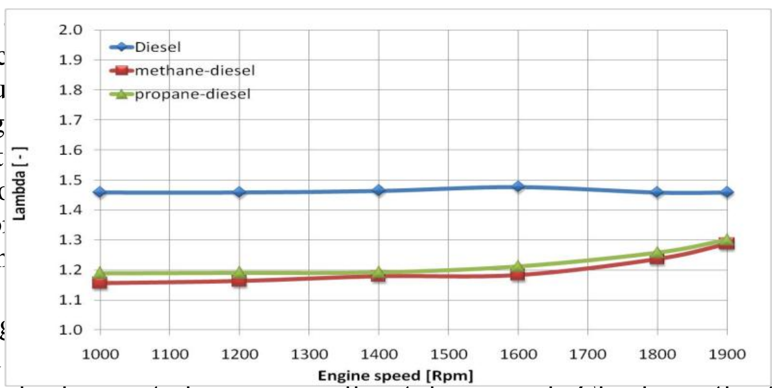
While a stoichiometric mixture of Diesel and air has an energy content of 2.82 MJ/kg of mixture, a stoichiometric mixture of CNG and air has an energy content of 2.76 MJ/kg of mixture 2.2% lower. Increasing lambda, the fuel conversion efficiency reduces because of the increased heat losses.

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 higher λ. 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 pollutants and carbon dioxide emissions.

Figure 9 – Brake efficiency at 15 bar BMEP – Diesel engine.

Fig. 9. Brake efficiency at 15 bar BMEP-Diesel engine

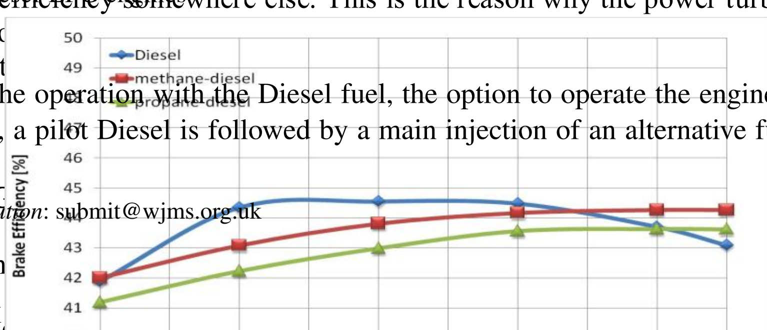
In addition to the … ne dual fuel is also considered. In this c… ve fuel (LPG, CNG). With start of combu… tion may possibly be omitted and the eng… assumption made for the dual fuel operat… alent single injection Diesel only are also… producing a similar mostly diffusion co… separate fuel routes for the Diesel and th…



In understanding fi… me with and without power turbine, and … oo much of exhaust energy after the turbocharger turbine, especially at low speed. Clearly, without the opportunity to disengage and by-pass the power turbine this device improves the engine efficiency in some areas but reduces the engine efficiency somewhere else. This is the reason why the power turbine with a fixed gear ratio has not been sc… nity to disengage and by-pass the power t… operations, with and without.

Figure 10 – Full load full load lambda (ratio of air-to-fuel operational to air-to-fuel stoichiometric) – Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines.

Fig. 10. Full load full load lambda (ratio of air-to-fuel operational to air-to-fuel stoichiometric)-Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines



In addition to the operation with the Diesel fuel, the option to operate the engine dual fuel is also considered. In this case, a pilot Diesel is followed by a main injection of an alternative fuel (LPG, CNG). With

2.2. Diesel-CNG…

As in the Diesel en… in injection event is considered. In both … e Diesel or the pilot

start of combustion a … y possibly be omitted
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are also valid for an … ilar mostly diffusion
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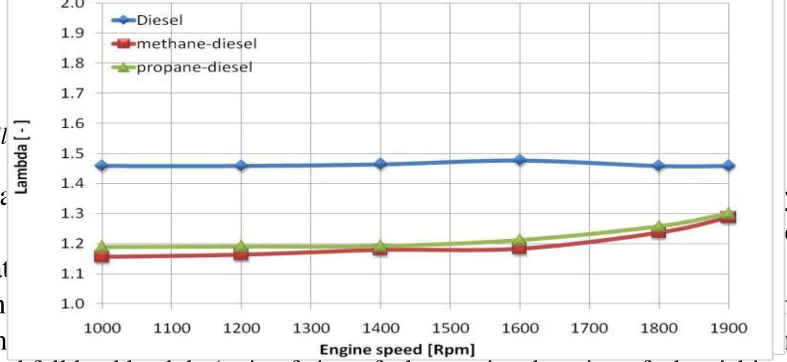

Figure 10 – Full load full load lambda (ratio of air-to-fuel operational to air-to-fuel stoichiometric) – Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines.

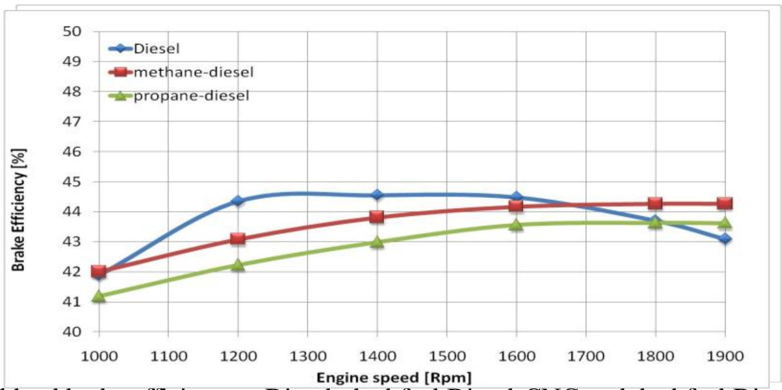

Figure 11 – Full load brake efficiency – Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines.

Fig. 11. Full load brake efficiency-Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines

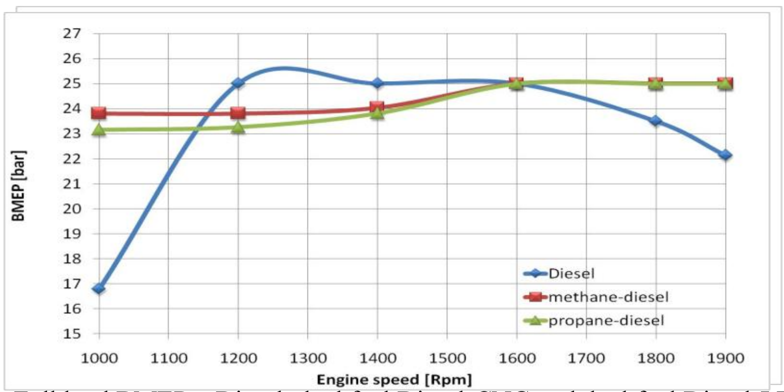

Figure 12 – Full load BMEP – Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines.

Fig. 12. Full load BMEP-Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines

In understanding Figs. 3 ∼ 5, we must remember that the turbocharger is the same with and without power turbine, and the power turbine is particularly negative when there is not too much of exhaust energy after the turbocharger turbine, especially at low speed. Clearly, without the opportunity to disengage and by-pass the power turbine, this device improves the engine efficiency in some areas but reduces the engine efficiency somewhere else. This is the reason why the power turbine with a fixed gear ratio has not been so successful and it is not that widespread so far. With the opportunity to disengage and by-pass the power turbine, noticeably it is possible to couple the better of the two operations, with and without.

2.2 Diesel-CNG and Diesel-LPG

As in the Diesel engine model, also in the CNG and the LPG models 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 or the LPG are neglected. Then, we suppose the same injection durations and the same combustion evolution will apply for the Diesel and the CNG or the LPG. Figures 10 to 13 present 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.

The CNG engine has to run with a larger 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 lower lambda.

It is at higher speeds that the CNG fuel permits to improve all the performances in both fuel conversion efficiency and power output also thanks to the power turbine. At medium speeds, the Diesel engine performs better than the CNG engine.

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. Increasing 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 higher λ. 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 pollutants and carbon dioxide emissions.

The LPG engine has similar to the CNG performances. The LPG engine has to run with a higher λ than the Diesel to achieve the target BMEP of 25 bar. A minimum value of λ of 1.2 is used for the LPG engine, while a minimum value of 1.46 is used for the Diesel engine. This compensates for the much faster mixing of the LPG flashing after injected and the therefore much quicker combustion that permits lower λ.

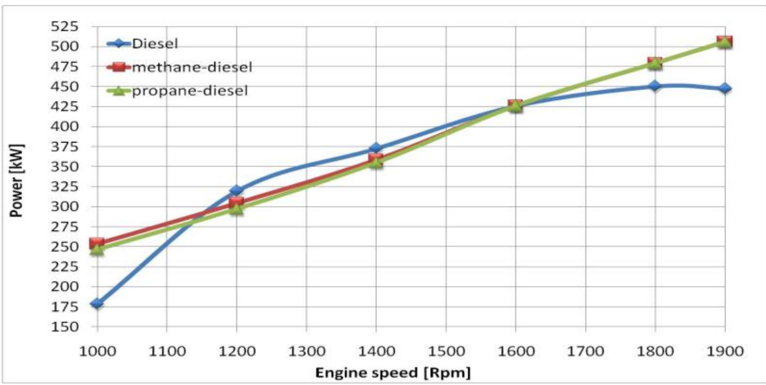

Figure 13 –Full load brake power – Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines.

Fig. 13. Full load brake power-Diesel, dual fuel Diesel-CNG and dual fuel Diesel-LPG engines

The LPG engine has similar to the CNG performances. The LPG engine has to run with a higher λ than the Diesel to achieve the target BMEP of 25 bar. A minimum value of λ of 1.2 is used for the LPG engine, while a minimum value of 1.46 is used for the Diesel engine. This compensates for the much faster mixing of the LPG flashing after injected and the therefore much quicker combustion that permits lower λ.

It is at higher speeds that also the LPG fuel permits to improve all the performances in both fuel conversion efficiency and power output also thanks to the power turbine. At low to medium speeds, the Diesel engine performs better than the LPG engine. While a stoichiometric mixture of Diesel and air has an energy content of 2.82 MJ/kg of mixture, a stoichiometric mixture of LPG and air has an energy content of 2.796 MJ/kg of mixture 2.2% lower. Increasing lambda, the fuel conversion efficiency reduces because of the increased heat losses.

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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 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.

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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

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 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 Litre 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 continuously 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.

Operating the engine dual fuel with a pilot Diesel and a main injection of an alternative fuel, the power turbine works better at higher speeds and loads with the alternative fuel permitting lower λ.

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 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 λ 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 LPG-Diesel internal combustion engine also 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 power outputs thanks to the much closer to stoichiometric operation with LPG. The benefits vs. the Diesel in terms of energy security and the environment are reduced vs. the CNG but still significant.

3.0.1 Abbreviations

BDC	bottom dead centre;
CVT	continuously variable transmission;
DI	direct injection;
EGR	exhaust gas recirculation;
EVO	exhaust valve opening;
EVC	exhaust valve closure;
IVO	intake valve opening;
IVC	intake valve closure;
LHV	lower heating value;
TC	turbocharged;
TDC	top dead centre.

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