On the possibility to decrease the fuel consumption for a heavy-duty truck Diesel engine using the Turbo-compound method: a case study

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Abstract The reduction of Diesel internal combustion engines emissions is one of the major concerns of the engines manufacturers. Despite the fact that the efficiency of the gas post-treatment systems has been significantly improved, decreasing the smoke and the soot from the cylinder inside remains a main research goal. This work is proposing a theoretical study on these pollutants formation for different kinds of direct injection methods. By dividing the in-cylinder injection the heat release characteristic could be modified, leading to different temperature and pressure levels. Using exhaust gas recirculation (EGR) the reduction of the gas temperatures might also be decreased, limiting NO_x formation. To evaluate the level of the cylinder gas emissions formation a two-step procedure could be followed. First, by using a numerical calculation system the heat release characteristic can be highlighted concerning a Diesel engine with stratified injection; then, using an experimental relationship applying a large data base, the amount of the gas emissions can be subsequently provided. The authors propose some combinations between injection characteristics and EGR used fractions which could generate successfully results speaking in terms of NO_x, soot and smoke formation.

1 Introduction

Diesel engine operation is very far to become replaceable related to the heavy-duty automotive and railway transportation, despite all the actual divergences regarding the exhaust emissions, especially those of NOx, therefore being criticized and almost banned in certain application zones, as far as the frequently occurring scandals are affecting the largest manufacturing companies.

For Diesel engines, approximately 30% of the fuel energy is wasted together with the exhaust gas, therefore, the higher the gas temperature is, the bigger the potential of the energy recovery should become. Following the main purpose of dropping the fuel consumption along with the basic gas emissions levels (CO2 and NOx), several methods have been proposed and tested, based on the use of the Rankine and Organic Rankine

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Cycles (RC and ORC), of the thermal-electric generators or of a supplementary turbine (Turbo-compound)[0].

The use of certain systems to recover the waste heat of the exhaust gas has become a long traditional matter, since the 1950's in the aviation industry and today various manufacturers, like Scania or Volvo are continuously using them when designing turbocharged engines [1]. By applying these systems, Volvo is considering that a 3% fuel decrease could be reached referring to the long distance transportation domain.

A great impulse to proceed to the energy recovery from the thermal engines was generated by the newest regulations brought to the Formula 1 racing cars [2]. For these ones two types of energy recovery systems have to be implemented in the breaking system, regarding both the transmission and the supercharging group (MGU-K and MGU-H). The operation of a racing car engine is characterized by highly unsteady regimes, varying from full load down to the braking zone, emphasizing in this way the reliability of these energy recovery systems.

The Waste Heat Recovery (WHR) from the exhaust gas by applying the turbo-compound method can be achieved using two different designing modes (see Fig. 1 with the turbo-compounding versions) [3].

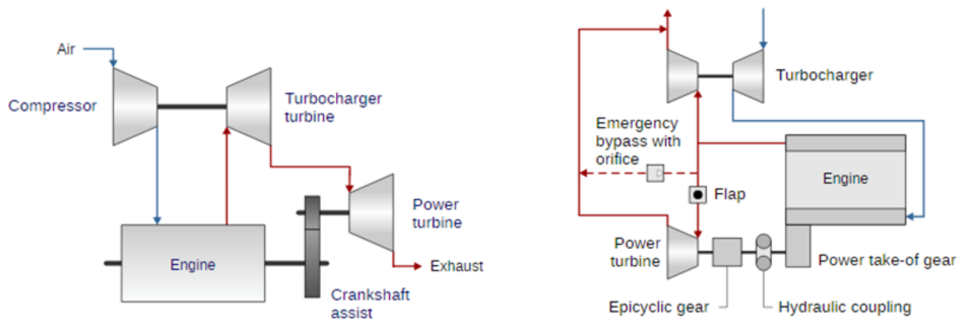

Fig. 1. Turbocompound versions [**3**]

According to the first design solution, so-called "in series", the power turbine is located downstream the supercharging turbine having therefore a lower level of the gas thermal entalpy. Speaking about the second release, with the "parallel" operation of the turbines, the power turbine uses a higher temperature level of the gas, but a divided amount of the gas flow, with the direct consequence of modifying the turbocharging regime.

The possibility of the supercharging system optimization has been evaluated by a number of studies in which various simulation software systems dedicated to the internal combustion engines have been applied [4,5]. The results converge to a fuel economy of about 2-3%.

2 Choosing the Engine and Modelling the Equipped Truck

For the theoretical study an engine designed by Detroit Diesel together with Daimler AG's Heavy-Duty Engine has been chosen [6]. This engine is equipping heavy-duty tractor heads with a capacity up to 22 t. Its features are: a number of 6-in-line cylinders with a total swept volume of 14.8 l, a compression ratio of 18, the stroke/bore ratio is 163/139 mm/mm and has one supplementary power turbine. Its performances are: maximal rating power of 340 kW at 1625 rpm and maximal rating torque of 2101 Nm at 975 rpm. These represent the maximal operating values and for the analyzed regimes lower values should be necessary.

The characteristics of the Freightliner Cascadia truck model [7] are: total weight of 23 t, head frontal surface of 7.8 m^2,being also considered an aerodynamic coefficient Ca=0.75. To calculate the necessary power of the vehicle in order to be driven on a horizontal route at constant speed, wheels friction and aerodynamic resistance are taken into account. Friction coefficient has a complex calculation formula, as following:

$$f = f_0 + f_{01} * v + f_{02} * v^2 + f_{04} * v^4 \tag{1}$$

where the used coefficients associated with the different speed exponents depend on the type of the contact between the wheel and the road [8]. Following a reasonable trust level of the precision the constant value of 0.016 for the "f" coefficient was adopted. The friction force with the road is proportional with the normal weight force of the track through the friction coefficient. A supplementary force with the slope (with the α angle) of the road is also added:

$$F_p = F_{frecare} + F_{panta} = f * G_a * \cos(\alpha) + G_a * \sin(\alpha) = G_a * (f * \cos(\alpha) + \sin(\alpha)) \tag{2}$$

in which Fp means the necessary propulsion force [N] and Gt is the weight force of the truck [N]. For a horizontal road (α=0), the slope component of the sum is null. Opposing the vehicle motion, a resistant force of the air is also to be considered and it is a direct correspondence with the square value of the relative speed between the advancing vehicle and the air (a supplementary effect of the wind blowing speed and direction is also taken). Normally, this calculation of the air resistance force [N] is provided by a semi-empirical formula:

$$F_a = \frac{1}{2} * \rho_a * v^2 * C_a * A \tag{3}$$

where ρa is the air density [kg/m3], v is the vehicle advancing speed [m/s], Ca is the coefficient of the total aerodynamic force [-] and A is the maximal cross section of the vehicle [m2]. The total propulsion power P [W] necessary for the truck in order to move at constant speed v [m/s] under the combined resistance force Fr [N] (from the air and from the road) is as following:

$$P = F_r * v = (F_p + F_a) * v \tag{4}$$

For the proposed model, corresponding to a highway cruising top-speed of 90 km/h, the engine necessary power should be 170 kW. The engine could provide this power starting from 900 rpm, as a result of the way in which the truck transmission is designed. Thus, the speed value of 1400 rpm is in good correspondence with the truck horizontal speed of 90 km/h. From the point of view of the engine operating domain, the proposed analysis refers to the use of the supplementary turbine system (turbo-compound) for a power range greater than 170 kW (see Fig. 2).

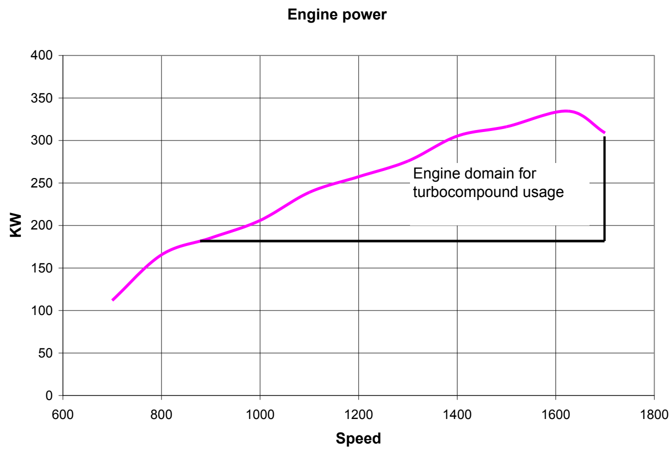

Fig. 2. Engine characteristic with turbocompound domain

3 Engine modelling

In order to describe the operation mode of the supplementary turbine together with the engine and its supercharging system it is necessary a complete description of the engine as an assembly of all its cylinders (3). The in-cylinder combustion model is based on Chmela model [8], included in AMESIM software, to deliver the way to estimate the cylinder heat release. The kinetic energy of the fuel spray is dispersed through the cylinder air and the combustion begins after the autoignition delay is consumed, delay which is calculated using an Arrhenius type formula. The combustion model takes into account also the effect given by the recirculated exhaust gas.

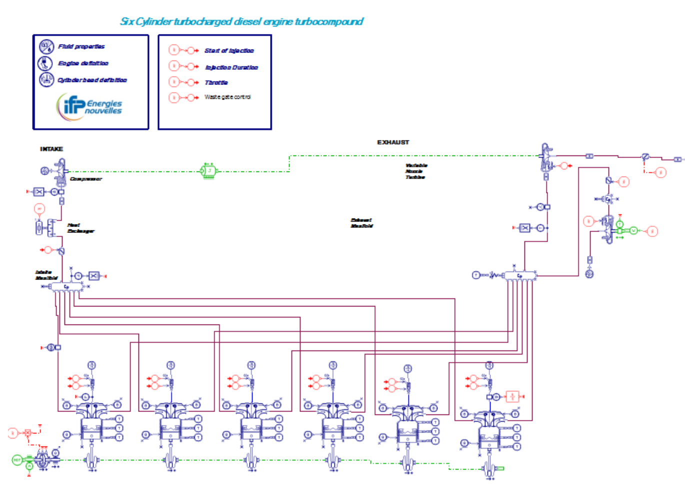

Fig. 3. The engine model

Modeling the turbocharging system was possible by using the functional characteristics of the both elements of the group, the turbine and the compressor. The characteristic of the turbine is plotted in Fig. 4. Its values are adjusted so that they could be grouped in families of curves.

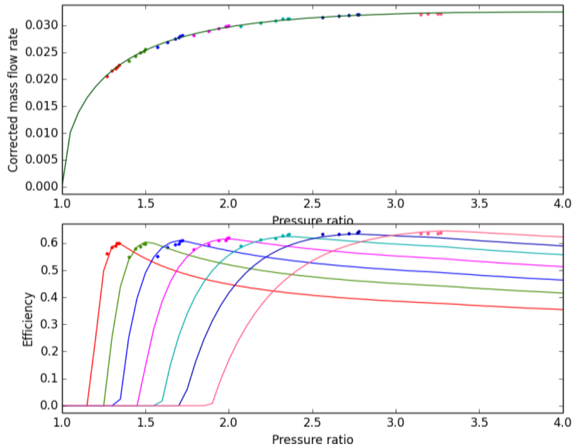

Fig. 4. Turbine characteristics

The families of the points correspond to the square root of the (n/T) fraction, with a resulting unit of [(rpm/K)^0.5], where n is the turbine speed [rpm] and T is the gas absolute temperature [K] and the gas mass flow is multiplied by the ratio between the square root of the absolute temperature and the gas pressure, having a correspondent unit of [(kg/s)·(K^0.5)/kPa]. For the compressor, the program uses also adjusted universally characteristics (see Fig. 5).

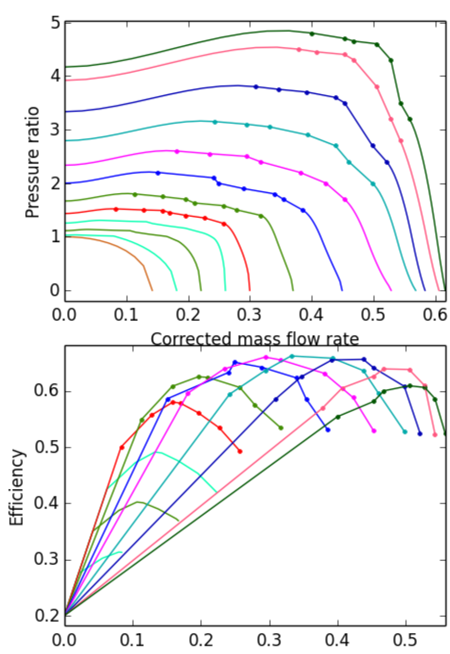

Fig. 5 Compressor chart

The curves describe the compressor behaviour at constant speed n, expressed by a reduced $n/(T_{up}/T_{st})$ form, in rpm units, where T_{st}[K] and T_{up}[K] mark the borders of the temperature increase during the compression process and the gas mass flow is respectively expressed by a non-dimensional correction given by its reduction with an expression of: $[(p_{up}/p_{st})\cdot(T_{up}/T_{st})$ ^0.5], in kg/s units, with the same remark concerning the upper limit p_{up}[kPa] and the lower limit p_{st}[kPa] of the pressure increase. Horizontal axis is marked by the these corrected values of the gas mass flow, and the vertical axis is plotting the values of the compression rate and those of the isentropic efficiency.

Related to the power turbine, the chosen model is presented in Fig. 6. From the exhaust gallery of the engine the gas is flowing toward the power turbine through a controlled flap, allowing the variation of the transferred gas quantity (see the second designing release from Fig. 3). Thus, diminishing the gas flow expansion through the main supercharging turbine, the compressor speed becomes lower, a lower air quantity is therefore admitted into the engine, affecting the rated power and the efficiency of the engine.

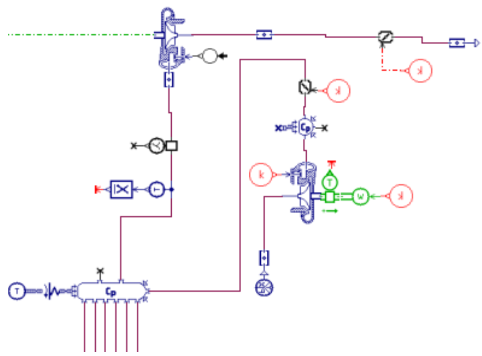

Fig. 6. Power turbine

4 Assembly-model calibration

The model has been initially calibrated so that the engine could obtain the performances from its technical specifications. Thus, at full load regime, expressed by a rating power of 340 kW at 1650 rpm, the in-cylinder pressure diagram indicates a top firing pressure of 226 bar and the cycle maximum temperature is above 2400 K (see Fig. 7).

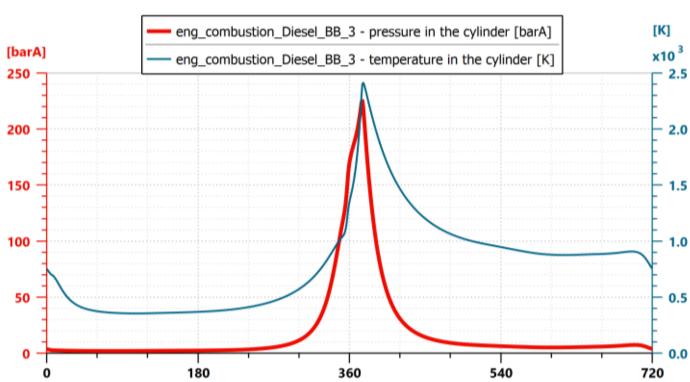

Fig. 7. The indicated diagram at nominal power

Regarding the gas exchange system, the inlet gallery pressure is 2.7 bar, the inlet temperature is 330 K, the exhaust pressure is 4 bar and the corresponding exhaust temperature is about 900 K. The turbocharging group speed is 95500 rpm.

5 Truck operating regimes

The analysis of the engine waste heat recovery system for the tested truck presumed that in order to drive it at constant cruising speed of 90 km/h, the engine has to provide the necessary power of 170 kW, at 1400 rpm. Starting from this regime the energy recovery appears interesting once that the gain of the power is always useful when driving overloaded. For the above mentioned regime, the indicated pressure diagram is plotted in Fig. 8. Referring to this, the cycle maximal pressure is 190 bar and the cycle maximal temperature is 2200 K. The correspondent relative air-fuel ratio is 2.02 and the speed of the turbo-compressor reaches 67000 rpm.

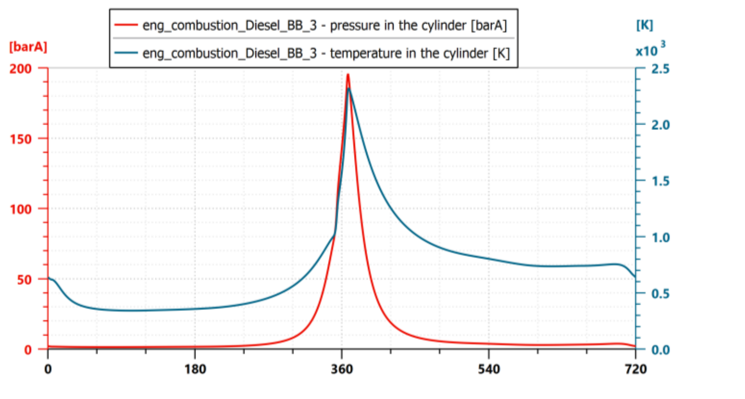

Fig. 8. The indicated diagram related to Pe=170 kW

For the first analysis 5 different cases of the throttle opening were considered, within the range from 20% up to 100% (from almost closed to wide opened). For the 80000 rpm value of the power turbine constant speed the exhaust gas flow operated by this one is highlighted in Fig. 9.

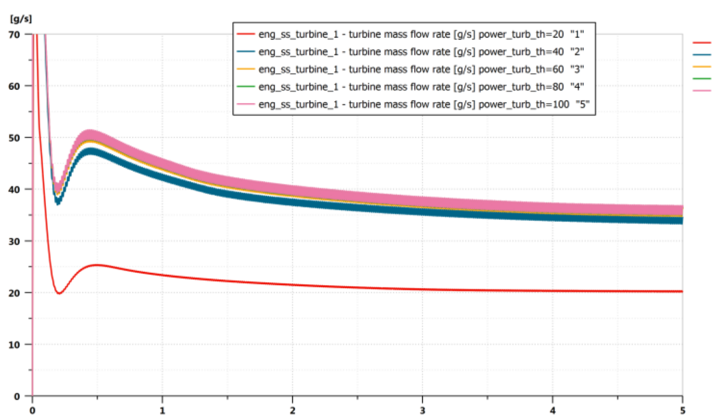

Fig. 9. Exhaust gas flow operated by the power turbine

The exhaust gas flow passing through the turbocharging group will decrease and its speed will consequently decrease from 59000 rpm down to 49000 rpm (see Fig. 10). In the same consequent way the turbocharging pressure drops from 1.8 bar to 1.4 bar and the relative air-fuel ratio diminishes from 1.75 to 1.53.

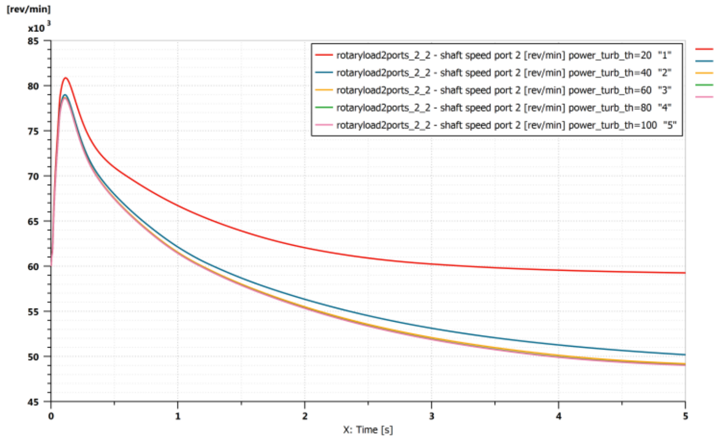

Fig. 10. Turbocompressor speed

Under these conditions the rating power of the engine remains constant, of about 170 kW, corresponding to the same fuel consumption of 0.1377 g/cycle, equivalent to a break specific fuel consumption (BSFC) of 201 g/kWh.

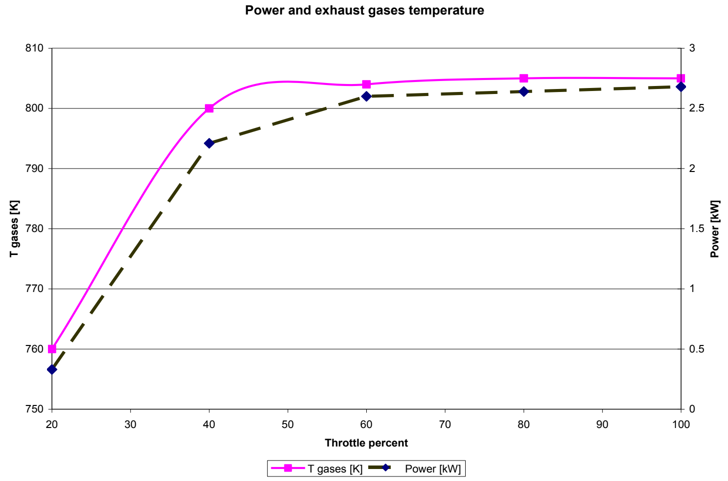

Fig. 11. Turbine power and exhaust gas temperature at 80000 rpm

As a conclusion, for this regime presumed to characterize a cruising driving of the truck with a constant maximum speed of 90 km/h, the saving of the power is about 2.5 kW.

6 Energy recovery at high loads

The highest level of the energy to be converted into mechanical work is associated to the full load domain. The operation of the truck engine corresponding to this domain will generally cover transitory situations, related to slope climbing or traffic outrunning. 6 different operational regimes, all over the standard cruising one were analyzed, in which cases the truck is in different ways overloaded. The parameters for these regimes are listed in Table 1:

Table 1. Overloading performance regimes

No.	n [rpm] Engine speed	P_e [kW] Engine rated power	n_{TS} [rpm] Turbo-compressor speed	p_s [bar] Turbo-charging pressure	λ [-] Relative air-fuel ratio	Tge [K] Exhaust gas temperature
1	1200	220	68402	1.83	1.52	801
2	1400	220	73000	1.91	1.8	765
3	1400	250	76800	2.04	1.69	792
4	1600	220	78300	1.98	2.02	752
5	1600	250	81800	2.12	1.89	780
6	1600	300	86700	2.32	1.74	819

The analysis of the possibility to recover the exhaust gas energy has been performed preserving the engine rated power, the gas flow towards the supplementary turbine being restricted by the valve. The speed of this power turbine remains at a constant value of 80000 rpm. The obtaining results regarding the following parameters are listed in Table 2 .

Table 2. Overloading regime performances using the power turbine

No	n [rpm] Engine speed	Pe [kW] Engine rated power	PTs2 [kW] Second turbine power	nTS [rpm] Turbo-compressor speed	λ [-] Relative air-fuel ratio	Tge [K] Exhaust gas temperature
1	1200	220	0.5	61000	1.32	850
2	1400	220	2.54	60300	1.43	837
3	1400	250	4.6	60970	1.27	890
4	1600	220	4.9	62353	1.5	831
5	1600	250	5.6	66450	1.44	862
6	1600	300	7	71270	1.33	908

The presented results are linked to the power turbine characteristics and to the mode in which the connectivity between the engine and the turbocharging group is expressed. Keeping the power turbine speed at constant value would make easier the design of the module to transfer the energy saving to the presumed user.

7 Conclusions

The work is based on the modelling, using the AMESIM numerical code, of a truck Diesel engine when adding a supplementary power turbine, operated by the exhaust gas flow from the outlet gallery through a flap controlling the gas flowing section.

Adding this supplementary element into the assembly design has major influence on the turbocharging system in order to ensure a significant amount of power for the new turbine. A basic hypothesis consists in preserving the engine generated power corresponding to a given fuel consumption and also in the fact that the exhaust gas quantity transferred to the new turbocharging group should not influence its efficiency.

This goal has been reached because when reducing the speed of the turbocharging group the relative air-fuel ratio decreases, but resources to maintain unmodified the engine efficiency still exist. In order to get significant rated power values it is necessary that the system should operate at high engine loads, the results indicating a maximum gain of power of 7 kW, or a decrease of the break specific fuel consumption by 2%. These savings have been obtained at certain operating regimes, an extended analysis of the system use for the rest of the operating domain being necessary.

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