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DEVELOPMENT OF A VIRTUAL CI ENGINE MODEL FOR ELECTRIC TURBO COMPOUND APPLICATIONS

Fernando Cunha Pimentel Ulhôa

Hairton Júnior José da Silveira

Sergio de Morais Hanriot

Pedro Américo Almeida Magalhães Júnior

ffernando.ulhoa@gmail.com

hairtonjjsilveira@gmail.com

hanriot@pucminas.br

paamjr@gmail.com

Pontifical Catholic University of Minas Gerais, Post-Graduation Program in Mechanical Engineering. Rua Dom José Gaspar 500, 30535-490 – Coração Eucarístico, Belo Horizonte, Minas Gerais, Brazil.

Abstract. *The quest for increasing the efficiency of propulsion units has become even larger as technology advances and environmental legislation become more demanding. More and more, the use of a turbocharger becomes indispensable in new engines. Thus, the possibility to couple an electric motor to the turbocharger to recover part of the energy contained in the exhaust gases is becoming very attractive, as demonstrated by several studies and by the current application in the Formula 1 championship. This study presents a theoretical review of electric turbo compound applications and its benefits and also the development of a virtual model of a mid-size (6.7 liter) Diesel engine. The focus is its validation together with the original base engine in order to use it as platform for a future electric turbo compound development. The developed model is able to fully meet the Brazilian standards and in the future can be upgraded up to EURO VI step C with the use of more advanced components. A final comparison between the results of the virtual model and data obtained in dynamometer indicated an average deviation of 2.8%.The set was developed through the Gamma Technologies's platform GT SUITE, which is a one dimensional (1D) engine code.*

Keywords: *Internal combustion engines, Computational Simulation, Electric Turbo Compound.*

1 INTRODUCTION

Diesel engines emissions are gaining increasing attention over the past with the growing awareness of the preservation of the environment. With this growing concern, over the years increasingly stringent emission controls are launched, posing major challenges for the development of engines and vehicles. Figure 1 below shows the evolution of the emission limit established by European standards.

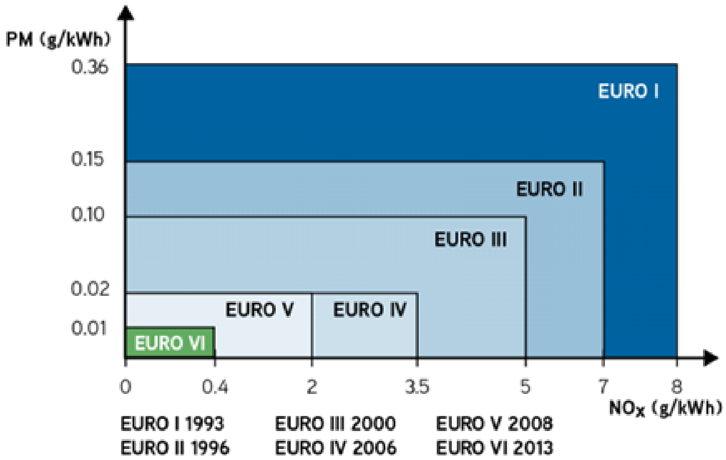

Figure 1. European emissions standards over the years.

Although the EURO VI emissions level has been in place for some time in Europe, in Brazil these standards fit close to the EURO III for agricultural and road machinery (PROCONVE MAR I) and EURO V for light and heavy duty vehicles (PROCONVE L6 and P7 respectively). Even though the regulations have long been outdated - the EURO III package was launched in Europe in 2000 - Brazil is a pioneer in South America to implement such control for agricultural and road machinery, the PROCONVE MAR I (ANFAVEA, 2015). Along with environmental concerns, rising fuel prices became a major motivation for the development of increasingly efficient propulsion units. Thus, secondary energy sources, such as contained in exhaust gases and the heat absorbed by the coolant, have become popular alternatives.

The recovery of energy from exhaust gases is very promising since the energy present in them represents about 30% of the potential energy of the fuel. The energy recovered from the exhaust can be used to power the engine directly or be used to power other auxiliaries such as lighting, air conditioning or audio system, thereby reducing parasitic loads on the engine and improving fuel efficiency. Among the methods developed for the redirection of the exhaust, turbo compound systems, although they do not offer as great yield as the Rankine cycle, have stood out because they present great possibilities of application. This devices have a low cost of implementation (compared to others Methods) and are also lightweight and compact systems (Teo Sheng J., Pesiridis A., Rajoo S., 2013).

A Turbo Compound system consists of a turbine positioned after the primary turbocharger of the combustion engine. The purpose of this second turbine is to take advantage of the flow and temperature of the exhaust gases to move it and thus direct such energy for other purposes. Typically, these systems are coupled to electric generators or gear systems for power transmission.

2 TURBO COMPOUNDING

A turbo compound engine is an internal combustion engine that has a power-turbine to recover energy from the exhaust gases. The turbine can be mechanically coupled to a gear and crankshaft that can provide additional power to the engine or be connected to a generator and produces electricity, which can power on board auxiliaries or stored in a battery. There are different ways of categorizing turbo compound engines such as series or parallel configuration, mechanical or electrical turbo compounding, among others. All of these setups can be observed on Figure 2 following.

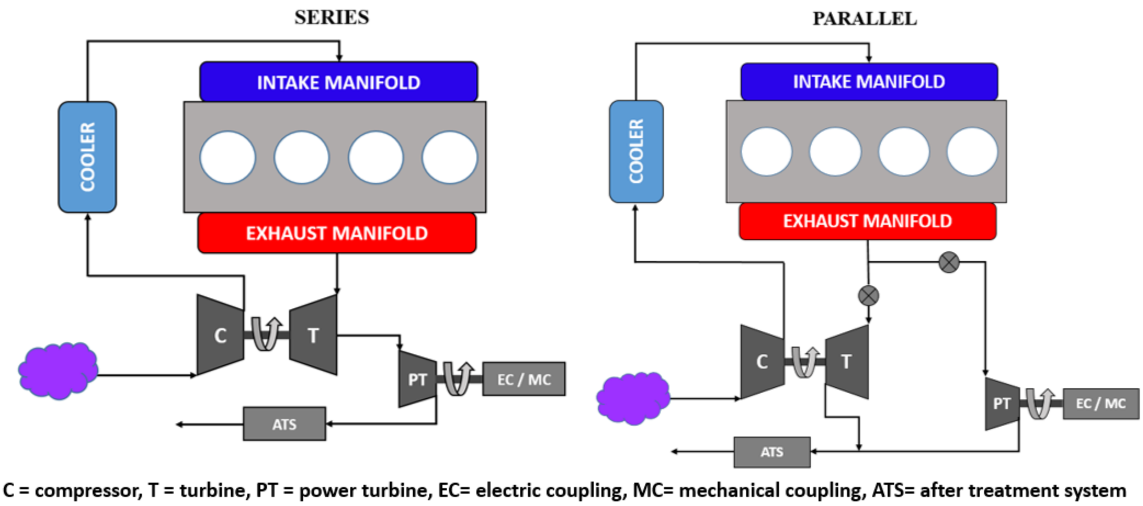

Figure 2. Turbo compound general layouts.

The mounts presented are only two bases from which several variations can be created. In a circuit like these can be added also EGR systems, WG and VGT turbines as the power turbine and single or two-stage turbo compounding. Although there are so many ways to categorize turbo compound engines, as shown, a more effective way of evaluating such systems is to perform a characterization of the rate of expansion of the power turbine. In this way we have the low pressure and high pressure turbo compound.

Low-pressure turbo compound (LPT) is a conventional turbo compound in which an extra turbine is used after the turbocharger and this second turbine operates at low-pressure ratios. The recovered power of the power turbine can be linked to the engine mechanically or converted to the electricity by a generator (Hiereth H., Prenninger P., 2007). LP turbo compounding usually has two stages of turbine, so then it can work without impose a massive rise of backpressure. High-pressure turbo compound (HPT) employs a single turbine that operates at higher pressure ratios and produces more power than the required compressor power. The excedent power from the turbine is directed to the engine or an electric generator (Hopmann U., Algrain M. C., 2003).

2.1 Mechanical Turbo Compounding

Mechanical turbo compound is a system that uses a second turbine in the exhaust to further obtain usable energy from the exhaust gas. The power is then transferred from the turbine shaft through a cascade to the crankshaft, thus, providing more torque and consequently more power to the engine. Studies show that mechanical turbo compound systems has advantages, such as high power density, fuel consumption optimization in heavy loading application, good engine response and well facilitated exhaust gas recirculation (EGR) due to high exhaust manifold pressure (Yin Y. et al., 2016).

Despite the simple explanation, implementing a mechanical turbo compounding in an internal combustion engine does have its challenges. Reliability and cost are some of the biggest concerns which create huge challenges in the design of gear train, coupling system and the power-turbine itself. The mechanical turbo compounding system also adds complexity and weight to the original engine configuration, which has to be weighted carefully for positive improvements. Another issue is that under low engine loading, there might be very little or negative gain from the turbo-compounding system.

2.2 Electrical Turbo Compounding

An electrical turbo compound system is very similar to the mechanical one, especially on the energy recovery method. Nonetheless, instead of being mechanically coupled to the engine, the power-turbine drives a generator. This generator produces electricity that can be guided to the vehicle electrical demand or stored in batteries. The electricity produced by the generator can be used to feed auxiliary power needs such as power steering and air compressors systems.

One great advantage of electrical turbo compounding is that it offers better output flexibility, once the power can be used to directly power auxiliary equipment or be stored in a battery and be used to power the auxiliaries when necessary to avoid engine idling for driver needs in hybrid application. This can increase fuel savings up to 10% depending on the idling reduction (Greszler A., 2008). The turbine power output and speed can also be controlled independently regardless of engine speed and load, opening a potential for better performance and emissions control.

3 TECHNOLOGY POTENTIAL

Modern Diesel engines can achieve efficiencies near 42%. For instance, efficiencies of typical spark-ignition engines ranges from 15 to 32%. A large amount of the fuel energy is lost as waste heat to the environment, in particular through the exhaust gases, which is responsible for about one-third of this energy. Exhaust systems of light duty and passenger vehicles operate at gas temperatures from 500ºC to 900ºC with most falling in the range of 600ºC to 700ºC, and most heavy duty engines produce exhaust gases in the range of 450ºC to 650C. This provides a significant opportunity for waste heat recover (WHR) to generate energy for various applications. Due to irreversibility, ambient condition, operational ranges, and other factors, only a percentage of the total exhaust energy is available for utilization. The exhaust exergy is more than 30% of its energy value at exhaust temperatures of around 600ºC and higher, but decreases along with the reduction of exhaust temperature. Exhaust exergy is low at low engine loads (following the pattern of driving cycles for example), however, it gets higher near the engine's peak efficiency zone (Aghaali H., Ångström H., 2015).

Simulations realized by Zhao et al. (2014, 2016, 2017) shown that an electrical turbo compounding system can provide a reduction of 8-9% overall BSFC with a turbocharger with relatively high efficiencies. Hountalas et al. (2007) achieved the turbo compound engine transient performance improvement by adjusting the power turbine rotating speed. The largest fuel reduction was up to 4% with the proposed electrical turbo compound. Briggs et al. (2014) studied the impact of power turbine sizing on the performance of a diesel electric hybrid bus. It was observed up to 2.4% reduction in BSFC was obtained over London's bus drive cycle. Mamat et al. (2015) fulfill a study on turbo compound with low expansion ratio power turbine. The researched 1-liter turbo compound gasoline engine obtained up to 2.4% of fuel savings at low speed conditions. Teo Sheng et al. (2013) showed that a mechanical turbo compound in a Diesel engine can reduce the BSCF up to 5,7% depending on the efficiency of the power turbine. The same study presents increased peak power up tu 10% and overall thermal efficiency improvement by 3-5%.

4 VIRTUAL MODEL DEVELOPMENT

For the virtual model, a standard 13-point acquisition along the engine's work range was defined. As it is an engine applied in urban vehicles, the working range is comprised between 850rpm and 2500rpm. In order to increase the quality of the tests, two more points of acquisition were added, one at each end of the curve, totaling 15 points of static analysis.

4.1 Governing equations

By using a one dimensional engine code, the governing equations are mass conservation, energy and momentum, solved along the flow's path. Equations of mass (1) and energy (2) are solved for each volume and the momentum equation (3) solved for each boundary between each volume.

$$\dot{m}_i = \dot{m}_o + \frac{dm}{dt}_{VC} \tag{1}$$

$$\dot{E} = \dot{Q} - \dot{W} = \sum_o \dot{m}_o \left[h + \frac{v^2}{2} + gz\right]_o - \sum_i \dot{m}_i \left[h + \frac{v^2}{2} + gz\right]_i \tag{2}$$

$$\sum \vec{F}_{EXT} = m\vec{a} \Rightarrow \sum d\forall \vec{f}_{ext} = \rho d\forall \frac{D\vec{V}}{Dt} \tag{3}$$

In the case of combustion and heat exchange, the DI Wiebe equation (4) and Woschini's correlation (5) were applied respectively.

$$\frac{Q(\varphi)}{Q_{TOT}} = 1 - exp\left[\ln(1 - \eta_{FUEL})\left(\frac{\varphi - \varphi_{SOC}}{\Delta\,\varphi_C}\right)^{M+1}\right] \tag{4}$$

One approach used in engine simulation modelling is to estimate the mass fraction burned as a function of engine position using the Wiebe function. The Wiebe function curve has a characteristic S-shaped curve and is commonly used to characterize the combustion process. The mass fraction burned profile grows from zero, where zero mass fraction burn indicates the start of combustion, and then tends exponentially to one indicating the end of combustion. The difference between those two ends is known as the duration of combustion.

$$v_C = \left[6.18 + 0.417\left(\frac{\pi\, R_{SWIRL}\, B\, \frac{rpm}{60}}{V_m}\right)\right] + 3.24\times10^{-3}\frac{V_D T_R}{P_R V_R}(P - P_m) \tag{5}$$

The dominant mode of heat transfer on the gas side of a combustion chamber is forced convection with a small radiation component. Woschni assumed that nitrogen is predominant in burned and unburned mixtures and that the transport properties vary as $k \propto T^{0.75}$ and $\mu \propto T^{0.62}$ for nitrogen. Moreover, by considering the ideal-gas law (ρ = p/RT), he rewrote the equation and assumed the Prandtl number equals to 0.74 and included it and the gas constant in the parameter C (already simplified in Equation 5). Woschni used the cylinder bore as the characteristic length and expressed the average cylinder gas velocity where the term $P - P_m$ is used for considering the effects of combustion on the increase in the gas velocity, and the subscript r refers to a reference state (start of combustion).

For the validation of the virtual model a standard comparison of power and torque, expressed by Equations (6) and (7) respectively, was selected.

$$\dot{W} = \frac{{}^{w}/_{V_d} \times A_p \times \bar{U}_p}{4} \tag{6}$$

Where w is the work of one cycle which can also be interpreted as the integral of the internal pressure along a volume ($\int P\, dv$), V_d is the engine's displacement volume (m³), A_p is the piston face area of all pistons in m² and $\bar{U}_p$ is the average piston speed (m/s). For this equation we have the final result in watts (W).

$$\tau = \frac{bmep \times V_d}{4\pi} \tag{7}$$

Where *bmep* is the brake mean effective pressure which can be calculated by the specific work divided by the specific volume (${}^{w_b}/_{v_{BDC} - v_{TDC}}$). As already shown, V_d is the engine's displacement volume (m³).

4.2 Engine Selection

The engine selected for this study was an 6.7 liter tier III on-road Diesel engine developed by FPT Industrial. The base of the engine consists of turbocharged intercooled 6-cylinder in-line diesel. The most important factor taken into account for this choice was its marketplace applicability, because this family of engines are available in Mechanical Injection and Common Rail systems, varying between 92kW and 419kW. Currently, engines of this family are applied to maritime, off-road, power generation and on-road vehicles. Therefore, choose a power unit of this platform guarantees a wide applicability of the virtual model and ensures a market for future technologies developed in this model.

This line of engines also features the most modern emission control systems such as iEGR (internal Exhaust Gas Recirculation), iEGR+ and SCR (light and heavy). In Brazil, emission standards equivalent to EURO V (PROCONVE P7 and L6) for urban vehicles and tier III (PROCONVE MAR I) for agricultural and off-road machinery (ANFAVEA, 2015) are in force. Such exhaust gas treatment systems are used up to EURO VI step C (ERECA, 2016), which guarantees to the developed model a very long useful life in Brazil. Table 1 below presents the main data regarding the engine selected for this study extracted from FPT Industrial’s datasheet (FPT INDUSTRIAL, 2017).

Table 1. Engine specifications

Configuration	OHV, 6 cylinders, 4-stroke, liquid cooling
Bore (mm)	104
Stroke (mm)	132
Displacement (cm^3)	6700
Compression ratio	17:1
Maximum power	240 kW@2500 rpm
Maximum torque	1140 N.m@1500 rpm
Fuel injection system	Common Rail

As well as the basic construction data to feed the virtual model, a reference curve, was used in order to complete the boundary conditions and to guarantee a means of comparing the real with the virtual (FPT INDUSTRIAL, 2017).

4.3 Engine Model

The virtual model of the engine has physical characteristics extremely similar to the physical one. All of its components were developed separately in order to achieve the highest possible fidelity to the original. On each piece, all parameters were set according to the manufacturer's specification sheet. After the geometric configuration of the model, the parameters of combustion (pre-mixing and burning), heat transfer, dynamic factors (such as swirl and tumble), NOx generation mechanisms, and others were inserted (GAMMA TECHNOLOGIES, 2015). Figure 3 below presents the complete engine model created in GT SUITE.

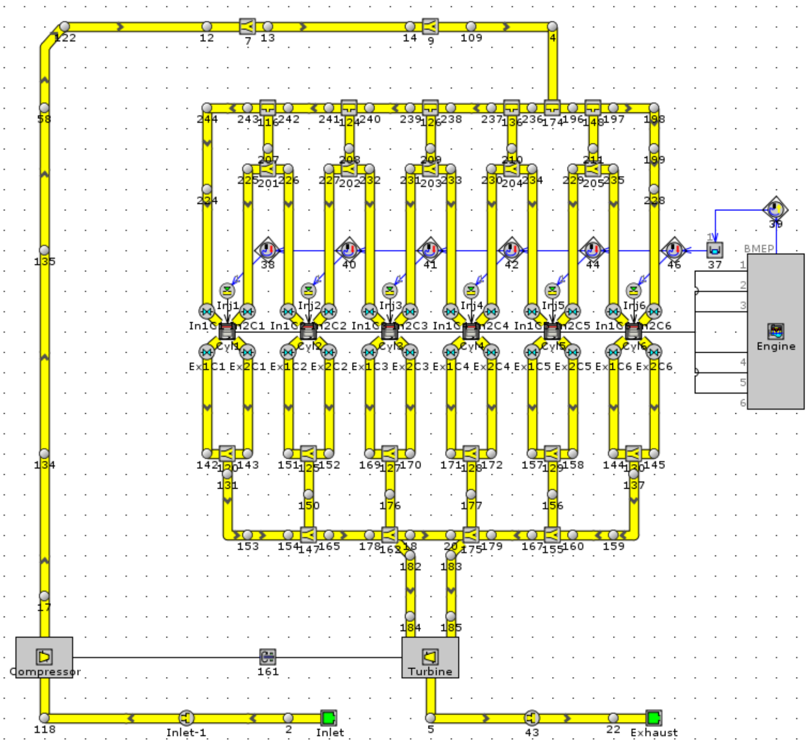

Figure 3. GT SUITE 1D engine model.

4.3.1 *Engine Model Assembly*

The model construction started by the engine intake manifold, which is shown in Figure 4 below.

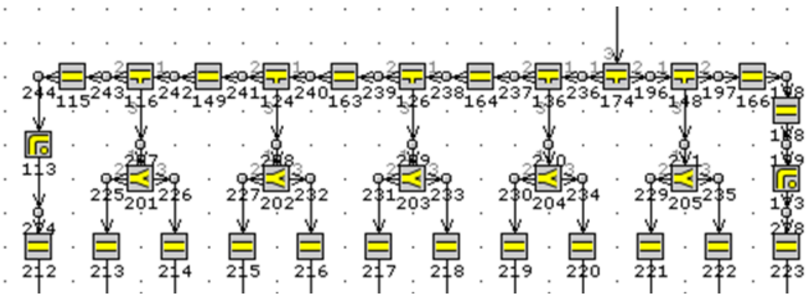

Figure 4. Virtual model intake manifold.

As can be seen, the air enters the system through a duct connected to the gasket 174. The inlet conduit for this system comes from the intercooler, which will be presented later. Just like in the real engine, this is not a symmetrical manifold so the inlet is away from the center of the part. After this intake, the air is distributed to the six cylinders of the engine and at its lower end over another division to supply the valves individually. This piece is made of cast aluminum, but receives a finish through machining. Thus, its coefficient of friction was neglected.

The next step was to develop the structure of the engine that would receive the intake manifold. In this way, the second developed block grouped cylinders, valves and the general characteristics of the engine. Figure 5 below shows the main block of the finished model.

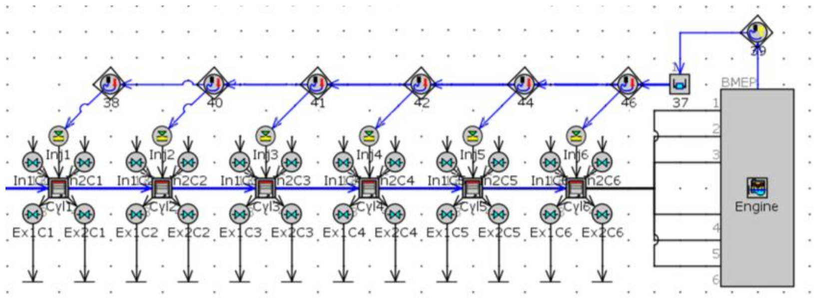

Figure 5. Virtual model base structure.

In the center the cylinders were arranged with their respective valves and injectors, which also have their injection control system separately. The flow and discharge coefficients were calculated for the valves and the cam profile of the camshaft was measured manually. From these data, it was possible to trace the opening curve of the valves and with the coefficients of discharge and flow, the program can calculate the flow rate and pressure of the air that enters and leaves the cylinders of the engine.

As already mentioned, the DI Wiebe model was used to determine the burn profile in two zones (burned and unburned mass). As the limits of Brazilian emissions are not so rigorous, the use of complex combustion strategies is not necessary to serve them. In various national applications, the emission limits are easily served with fully diffusive injection regimes and application of light after-treatment systems, like light EGR. For this study, only one Wiebe profile was used to calculate the mass burn fraction for a full diffusive injection.

Unlike a spark combustion engine, where the start of combustion is controlled by the spark plug pulse, in compression combustion this set point is not as well defined. By analyzing a diesel combustion, the starting point, therefore known and controlled, is just the beginning of the fuel injection. As indicated in Figure 6, after the injection has begun, there is generally a time until the combustion actually begins. During this period, the atomization of the fuel, evaporation, mixing with the air and some pre-chemical reactions occurs. This period is called τ_{id} and can be represented by Equation (8) below.

$$\tau_{id} = C_1 p^{C_2} \exp(C_3/T) \tag{8}$$

This approximation was obtained experimentally at a constant volume of fuel with a cetane number greater than 50. A temperature range between 590K and 780K was considered and the constant values as C1 = 0.44ms, C2 = -1.19 and C3 = 4650K leaving the pressure to be the only entry.

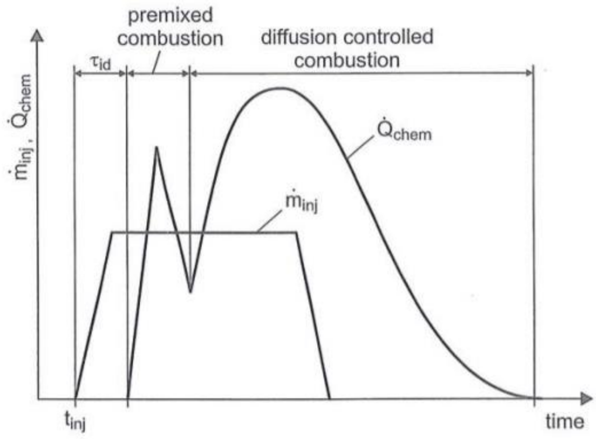

Figure 6. Single DI Wiebe burn profile.

The fuel premixing factors and burning duration were initially set to the standard value of the system, however, after some tests and a comparison with the temperatures and pressures found empirically (dynamometer data), these values were slightly modified to favor a more precise adjustment. Dynamic factors such as Swirl and Tumble were defined as default due to lack of previous data. Finally, a standard NOx generation mechanism was used, since a very precise emission control is not taken into account for this step.

The NOx generation model used was adapted from Ericson et al. (2006). In this study, the NO formation was modelled using the Zeldovich mechanism equilibrium approach. The equilibrium approach assumes that the residence time in the flame is short; therefore the NO formation takes place in the burned zone only.

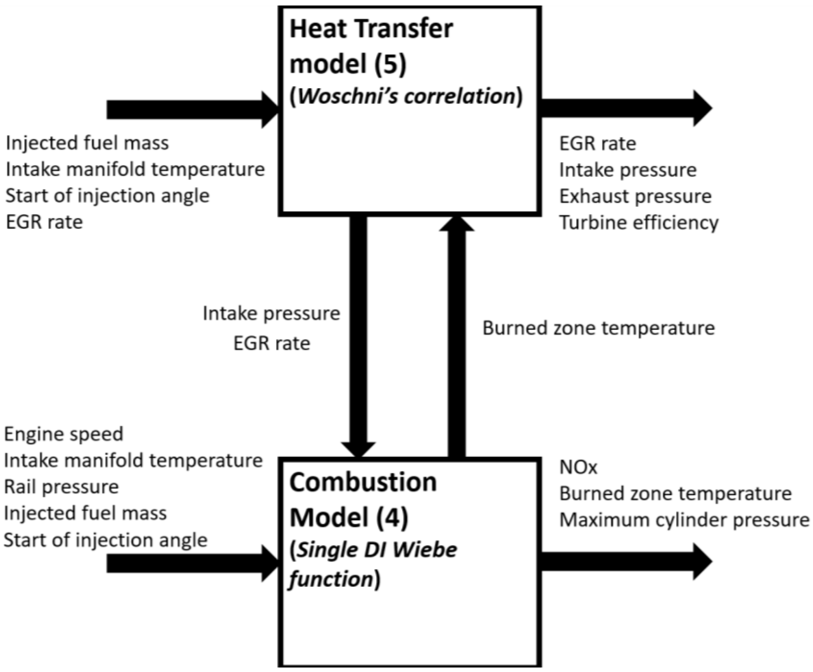

Figure 7. NOx prediction model.

The model consists of two parts. The combustion model describes the heat release, pressure and temperature inside the cylinder and NOx formation. The gas exchange (heat transfer) model generates input to the combustion model, inlet manifold pressure and EGR rate. External inputs are inlet manifold temperature, speed, EGR valve and VGT actuator signals, injected fuel mass, injection timing and rail pressure (of the common rail injection system). A large number of output signals are possible. Typical outputs are inlet and exhaust manifold pressure, EGR rate, NO concentration, peak cylinder pressure and exhaust temperature.

The final part of the model to be built was the set comprised of exhaust manifold, turbocharger and intercooler. It was decided to create such a large block in order to avoid any inconsistency, since these three have great dependence and synchronism. First, the exhaust manifold was designed, this time symmetrical and cast iron, and attributed a coefficient of friction of 0.25, according to the literature (Novaski O., 2013). Moving on to the turbocharger a Holset fixed geometry and waste gate was used for development. The turbine and compressor pressure ratios were taken from the dynamometer tests while material data, friction and moment of inertia of the parts were purchased from manufacturer's catalogs. The intercooler was dimensioned from the Woschni GT model to reproduce the temperatures found in the previous physical tests. Figure 8 below shows the third block of the finished model.

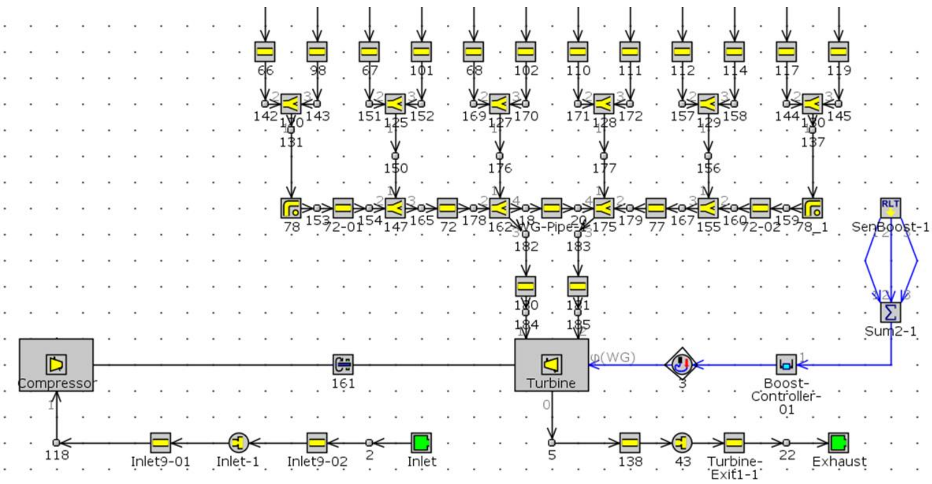

Figure 8. Virtual model exhaust manifold and turbocharger.

At the end of the development of each block, they were assembled into a single design to form the complete engine system, shown earlier in this chapter.

5 RESULTS AND DISCUSSIONS

After performing several simulations and analysis of the proposed virtual model, it was possible to achieve consistent results. The system was created aiming a reproduction of the original engine, thus, its output data should be as close as possible to the engine reference obtained in dynamometer. For the validation of the developed model, a correlation with the data of the real engine was carried out using information from an average of three tests performed in dynamometer. Following, a power curve and a torque curve were generated to be compared with those obtained in the GT-SUITE. Figure 9 following presents the comparison between the virtual model and the real engine.

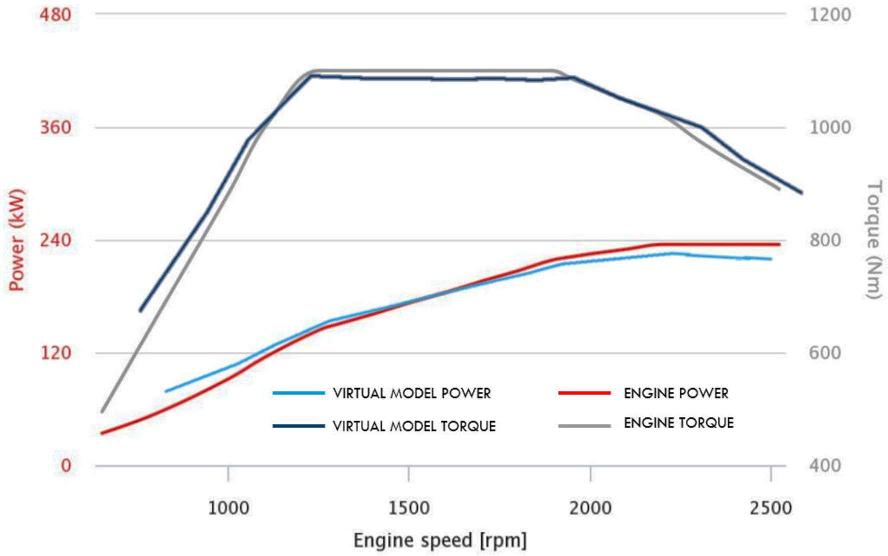

Figure 9. Power and Torque comparison.

At lower range, from low idle to about 1200rpm, the difference between the modeled engine and the real engine was the maximum on the entire test range, keeping on average 4.8%. At the higher end, the average differences were smaller. The modeled engine deviates from the real engine by about 1.15% of the original engine from 1250 to 1900rpm. Overall, the average difference between the modeled engine and the real engine is about 2.8%.

Although the simulation was performed with relatively simple hardware (a fixed geometry turbocharger, waste gate and a 100% diffusive combustion strategy was used), it is possible to advance this model to more current standards. As it already happens in Europe and the United States, this platform is able to achieve regulations up to EURO VI step C with the use of twin scroll turbochargers, VGT, multipoint injection strategies, low soot high performance lubricants and after treatment systems as the high efficiency SCR. But for the time being, the system presented here is fully capable of meeting the Brazilian demands.

6 CONCLUSIONS

In order to compose a Mechanical Engineering master dissertation this project aimed the development of a virtual model of a turbocharged intercooled six cylinders diesel engine faithful to the original engine for application in further studies. The results obtained, with an average deviation of 2.8% of the dynamometer measured data, are satisfactory and acceptable for the continuation of the work and open the precedent, with the GT-SUITE platform, to simulate electric and mechanical turbo compound systems.

As presented in the previous topic, the model created has the advantage of being able to receive several upgrades with advanced components (from pistons and injectors to a more robust electronics) to be able to comply with the more rigorous regulations. Therefore, its useful life following the Brazilian regulations can be dramatically increased making it possible to compare, study and develop technologies compatible with the most advanced countries.

ACKNOWLEDGEMENTS

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DEFINITIONS & ABBREVIATIONS

$\boldsymbol{\varphi}$ – Crankshaft angle of combustion;

$\boldsymbol{\varphi_{SOC}}$ – Crankshaft angle for start of combustion;

$\Delta\ \boldsymbol{\varphi_C}$ – Combustion angle duration;

$\boldsymbol{A_p}$ – Piston face area [m²];

B – Bore;

BDC – Bottom dead center;

bmep – Brake mean effective pressure;

BSFC – Brake specific fuel consumption;

EGR – Exhaust gas recirculation;

iEGR – Internal exhaust gas recirculation;

iEGR+ – Internal exhaust gas recirculation +;

HPT – High-pressure turbo compound;

LPT – Low-pressure turbo compound;

$\boldsymbol{P}$ – Engine cycle pressure [bar];

$\boldsymbol{P_m}$ – Engine cycle mean pressure [bar];

$\boldsymbol{P_r}$ – Reference pressure [bar];

$\boldsymbol{R_{SWIRL}}$ – Swirl ratio;

SCR – Selective catalytic reduction;

$\boldsymbol{T}$ – Cylinder temperature [K];

$\boldsymbol{T_r}$ – Reference temperature [K];

TDC – Top dead center;

$\overline{\boldsymbol{U_p}}$ – Average piston velocity [m/s];

$\boldsymbol{v_c}$ – Piston velocity [m/s];

v_d – Displaced volume [m³];

v_m – Piston average velocity [m/s];

v_r – Reference volume [mm³];

VGT – Variable-geometry turbocharger;

WG – Waste gate;

WHR – Waste heat recover;

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