A phenomenologically based airline model of a 2 MW Gas Engine

VALENCIA, G. E. [a*], CONSUEGRA, F. E. [b], OSORIO, M[c]

a. Universidad del Atlántico, Área Metropolitana de Barranquilla

b. Universidad del Norte, Área Metropolitana de Barranquilla

c. Universidad Pontificia Bolivariana, Medellin

**Corresponding author, guillermoevalencia@mail.uniatlántico.edu.co*

Abstract

With the continuous advance and technological growth of society, the demand for energy has increased, more specifically the consumption of electrical power. This has led to the development of renewable energy sources such as wind power or solar energy. Despite their numerous advantages, such as environmental and economic benefits, at an industrial level, higher reliability and generating capacity energy sources are preferred. Because of this, nowadays many industrial sectors prefer fossil fuel-based energy generation, frequently using turbines and internal combustion engines as a primary energy source. The choice of one or other primary energy generation option depends on how variable the electricity demand is, as it causes continuous change in generator load. If a constant energy demand and economic feasibility study are required, a gas turbine can be chosen, despite having lower thermal efficiency than internal combustion engines. Otherwise, an engine is a better choice whether operating on diesel or natural gas as a fuel, because its high robustness allows it to adapt better to variable load rates. Considering the above, it is not surprising that in Colombia, a high percentage of industries uses generation engines to self-generate a part of its electric consumption. Considering that, usually, these generation engines must supply electrical power to industrial plants in a 24-7 regime, it is not feasible to keep them out of operation, neither in unexpected operation regimes for long periods of time. This, together with the manufacturer's restrictions and the laborious nature of making changes within the processing and control unit of an equipment, make it necessary to carry out a simulation of the system with its respective results, which can be transferred to the real system later. Therefore, the generator-engine assembly has been defined as a set of process systems, where the operational behavior of the equipment can be simulated using mathematical equations, obtaining a phenomenologically based semi-physical model that can be used to perform experiments in simulations. Finally, this work focuses on the implementation of the methodology for combining phenomenologically based semi-physical models to obtain a dynamic of the air line of a 2 MW Jenbacher natural gas internal combustion engine, focusing on modeling of mean values that involves the study of some engine parameters such as the intake manifold temperature and pressure, the mass flow through the throttle and turbo-bypass valve in the engine, the electric power and the gas emission.

Keywords: *Mean Value Model, Gas Engine, Phenomenological Semiphysical model*

1. Introduction

Since its inception, Internal combustion engines have been object of academic interest and widely studied, so it is not surprising that from the first proposals of the Otto cycle for the spark ignition engines and diesel engines that was made for compression ignition, many researchers have been deepening in the thermodynamic study of these thermal machines, which allows to explain the hydraulic, thermal and chemical behavior of both the spark and compression ignition engines [1]. From these approaches, a large number of authors have developed models that aim to describe the phenomenology of combustion engines, to improve their design and development, by using both the cylinder-by-cylinder focus, and mean value modeling [2].

One of the characteristics of the first scope is the detailed study of the energy provided by the fuel input, where the interest is devoted more specifically to the behavior of the rate of heat release when the fuel ignition is carried out. There are different proposals for modeling this phenomenon, where the commonest are zero-dimensional and thermodynamic approaches, which resolve in time the equations associated with the corresponding control volume. Therefore they do not consider spatial variations in the control fluid volume [2]–[4].

Another characteristic of the cylinder-by-cylinder modeling is that it can be used to predict and to diagnose in detail the engine combustion process, and only some data is needed such as the behavior of heat release and the combustion chamber pressure. This formulation have been used for predictive purpose in diesel engines such as those of Payri [3] and Rakopoulos [5], while the diagnostic point of view results are presented by authors such as Armas [6] and Martin [7]. The predictive purpose, despite being a powerful tool, requires a previous engine characterization, and if it is not available the results obtained from the model will be far from reality, so a previous combustion diagnostics is necessary.

Although the first class of models described above is highly used, for control and fault detection applications in internal combustion engines, the second class, the mean value models, are chosen [2]. This type of modeling, despite not a high complexity compared to its counterpart previously described, generates excellent results [2], [8]–[10]. Mean value models, coincidentally, emerged along with the growth of automation and control in the industry during the 1990s, although engine control had already been attempted in previous decades [11]. Authors such as Hendricks [9], [10], [12] promoted the proposal and development of these models to simplify the control and observation of engines since the different existing forms of engine control at that time were based solely on models of quasi-static engines and operation maps [9]. This orientation in the modeling caused high wait times while the model adapted to the current engine conditions and different types of engines, in addition to being very sensitive to sensors errors, entailing sometimes undesirable dynamic behaviors.

In 1991, to improve the control strategies applied in the spark ignition engines, a mean value model was implemented in microprocessors for the application of control and regulation strategies already known as "speed-throttle control," "Speed-density control" and "mass air flow (MAF)" [10]. In 1992, based on previous work, the limitations of the control strategies implemented in internal combustion engines in the 1980s were highlighted [13], in a work that showed that the most common barriers were that this strategy did not have the capability of taking into account the non-linearities inherent to the combustion engines processes. Hendricks, based on the previously proposed models [9], [10], [14], implemented observers for the correct control of variables such as the fuel-air ratio. Subsequently, through different works and publications, it has been demonstrated with more experimental evidence that this kind of model observer applications had a high potential [12], [15].

Authors such as Müller [16], following the research line framed by Hendricks [9], [10], [13], [14], proposed a mean value model for spark ignition engines. This time it was used for turbocharged engines, so the dynamics of the intake air and the combustion products expulsion changes significantly compared to the proposal used for natural aspiration engines. Müller describes the phenomenology associated with the compressor and the turbine in a form, which, despite its compactness, produced excellent results in comparison with experimental data [16]. One year after the previous work, Müller, in cooperation with Fons [17], proposed a model that, in addition to approaching the phenomena associated with turbocharged ignition engines, added EGR (Exhaust Gas Recirculation) consideration. This work, like the previous one, was validated experimentally.

Another author that stands out in the field of mean value modeling in engines is Chevalier [18], [19]. In the first work referenced, Chevalier, in order to justify the effectiveness and use of the MVEM (Mean Value Engine Model) built a predictive combustion model, much more complex than the one previously mentioned in the framework of this paper (the complex model of Chevalier [18] is a one-dimensional model that considers simplifications of the general Navier Stokes equation). This adjusted MVEM, in comparison with the more complex model, obtained excellent outcome when used to represent mechanical losses and temperature rates in the intake manifold, despite its apparent simplicity with respect to the robust one-dimensional predictive model. For the development of MVEM. Also, Chevalier, based on his previous work [18], demonstrates the applicability of the MVEM itself in activities associated with the control of the air/fuel ratio of spark ignition engines [19].

In more recent years, authors such as Vasu [20] proposed, in addition to the implementation of an MVEM, to attach to the latter the ability to detect faults in the engine. In the particular case of Vasu, he developed a fault detection methodology based on the Dempster-Shafer theorem along with tests of multiple hypotheses. Finally, Na [21], despite addressing again the challenge of controlling the air-fuel ratio of an spark ignition engine through the combination of MVEM's and sliding observers, implemented them with the notorious difference that although there are uncertainties at the input of the observer, the system can efficiently deal with them without losing robustness in the observation.

Taking in account all this previous research, it can be noticed that continuous improvement in topics associated with the modeling and control of internal combustion engines is needed, particularly the spark ignition engines. In the following sections the development and validation of a model for an airline spark ignition generation engine, obtained by means of a methodology oriented to obtain mean value semi-physical models.

2. Methodology

Since this work consists in obtain equations that describe the systems and phenomena (Phenomenological Base Models) required for future observers implementation in different representative variables of the engine performance, the need arises for a clear and concise methodology that allows it. To this end, the methods used and described by Álvarez are used here [22]. This methodology consists of 10 steps, which will be described as follows [22]: Step 1: Describe verbally and by means of a flow chart of processes that complement each other. Step 2: Set a detail level for the model, according to its use: What questions will the model answer?. Step 3: Define as many process systems (PS) within the process to be modeled, as required by the detail level, and represent the relationship of all SdePs in a block diagram. Step 4: Apply the principle of conservation of mass and energy to each process system. The set of equations obtained in this step are called "Dinamic Balance Equatins" (DBEs) Step 5: Select from the DBEs those with valuable information to meet the objective of the model. Step 6: Define, for the essential DBEs the parameters, variables and known constants of each PS. Step 7: Find constitutive equations that allow the calculation of the most significative parameters in each process system. Step 8: Verify the model degrees of freedom. Step 9: Obtain the computational model or solution of the mathematical model. Step 10: Validate the model for different conditions and evaluate its performance.

3. Model Development

Below is the application of the methodology for obtaining a Phenomenological Base Semi-physical Models to the airline of a 2 MW natural gas generation engine, with nominal data as shown in Table 1 [23]. It is necessary to notice that only the steps from first to ninth are considered in this section, the tenth level is discussed in section 4 four.

Table 1. Physical Engine Parameters

Displacement (lts)	Compression Ratio	Number of Cylinders	Stroke (mm)	Bore (mm)	Max Torque kN/m	Max Power (kW)	Nominal Velocity (RPM)
74.852	10.5	12	220	190	60.66	1820	1500

Step 1: Describe verbally and by a Flow Chart of Processes the process to model. Initially (see Fig. 1a), air suction (1. Air) is mixed in the mixer, as shown in the general flow diagram in Figure 1a, at a temperature between 30 °C to 40 °C, relative humidity between 75% to 85% and an atmospheric pressure around 1.0074 bar, with natural gas (2. Fuel), whose composition is 97.97% CH_4, 1,5% N_2,

0.1626% CO_2, 0.2559% C_2H_6, 0.0516% C_3H_8 , 0.0207% C_4H_{10} (i-butane), 0.0083% C_4H_{10} (n-butane), 0.0076% C_5H_{12} (i-pentane), 0.0021% C_5H_{12} (n-pentane) and 0.0132% C_6H_{14} (n-hexane) with a pressure line comprised between 1152 bar at 1211 bar, and uncorrected volumetric ratio of 110 L/s to 140 L/s in order to obtain an optimal flammable gas-air mixture with a lambda value or air/fuel ratio. Actual amount of air with respect to the minimum amount of air necessary for a complete combustion, comprised between 1.4 to 1.8.

Subsequently, the gas-air mixture leaves the gas mixer (3, 7) and get into the wheels of the compressors of the two turbo-compressors that operate in parallel, which suck the mixture increasing both its pressure and temperature, to values between 3 bar and 5 bar, and could not recorded in the process.

Next, the currents (6) and (10) join and come into the engine's suction chamber through the mixer, mixer/water heat exchanger, to lower its temperature and then go to the check valve, also called throttle valve, which allows to adjust, according to its position, the flow of the mixture to the suction chamber of the crankcase, point in the process (16) where the mixture enters the intake manifold and is distributed to the 12 cylinders, with a mixing temperature between 60 °C to 70 °C and loading pressures between 2.6 bar to 4.6 bar. The engine generates warnings when the mixing temperature exceeds 71.1 °C and fails when it exceeds 75 °C.

The mixture flow can be regulated with the throttle valve and the turbo bypass valve. The throttle valve takes a percentage opening that depends on the operating engine mode, which could be 80% for island mode, independent operation of the network, or 98% in synchronism, operating in parallel with the system. The turbo bypass valve takes values around 15% to 50% independent of the mode of operation. The turbo bypass valve recirculates the mixture (14) to the output of the mixer (4 and 8), allowing a regulation of the flow supplied to the equipment and therefore regulating the electrical power generated by the engine which is between 1000 kW to 1979 kW. Before entering the mixture into the cylinders, combustion is done in a pre-chamber to guarantee the reaction homogeneity; this combustion is complemented if required with a new injection of secondary fuel, generating exhaust gases (31 to 44) in each cylinder at a temperature between 580ºC to 650ºC.

Finally, both exhaust gas streams (21, 26) are mixed in a single outlet (27), with a concentration of the gases that at the minimum load of 1000 kW with lambda of 1.79 and maximum capacity of 1982 kW with lambda of 1.97, is comprised between 9.45% to 10.52% of O_2, 731 mg/m^3 to 588 mg/m^3 of CO, 461 mg/m^3 to 468 mg/m^3 of NOx, 317 mg/m^3 to 368 mg/m^3 of NO_2, 95 mg/m^3 to 65 mg/m^3 of NO.

The engine operates nominally at 1500 rpm, coupled with a three-phase electric generator that at a frequency of 60 Hz, the power factor of 0.9%, delivers a reactive power of 911 Kvar, an electric potential of 1975 kW and an apparent power of 2177 kva , with an average voltage between lines of 13,264 V.

a) b)

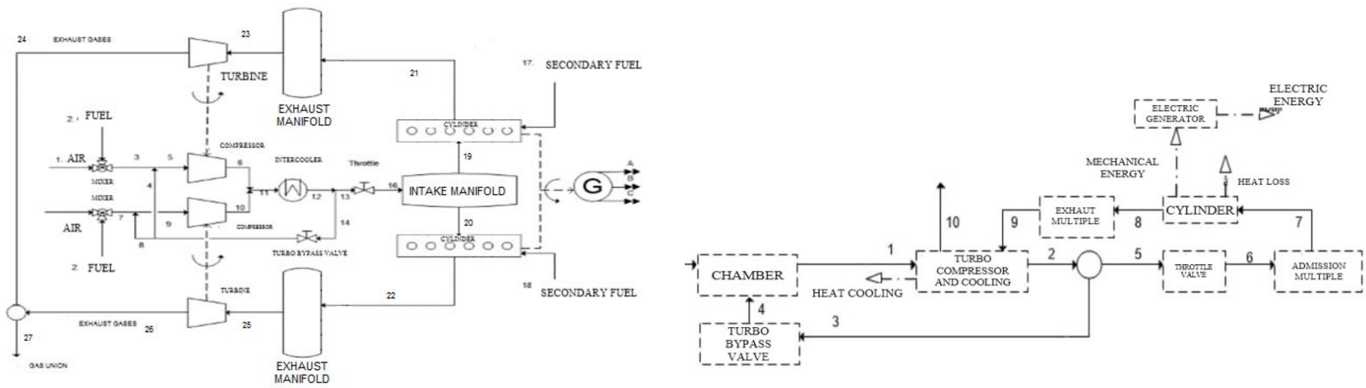


Fig. 1. a) General flow diagram engine system generator. b) Subsystem processes diagram.

Step 2: Set a Detail Level for the Model, according to its use: What questions will the model answer?

A particularity of the spark ignition engines fueled by natural gas is that the power they deliver depends on the mixture density, which depending on its value, positive or adverse effects on the engine performance can be caused. The volumetric efficiency is an indicator of the cylinders filling during the renewal of the engine load and has been defined as the ratio between the mass flow of air entering the bottle and the mass flow that would be displaced in ideal conditions by the engine [24]. Therefore, the two questions that the model will answer is: how variables such as mixing temperature, the pressures of the intake manifold, the opening of the turbo bypass are related with electrically generated power and nonstandard volumetric gas flow? And also how these last two variables are associated to the engine emissions?

The macroscopic approach used by Guzzela [25] will be used to answer these questions, given that it is more suitable for our application and adapts better to the instrumentation available on the equipment.

Step 3: Define as many Process Systems (PS) on the Process to be modeled as required by the detail level and represent the relationship of all PS in a block diagram. In the macroscopic model of the turbocharged natural gas engine, each equipment is represented as a different sub-system of the fixed boundary, where the mass and energy interactions of each processing subsystem are highlighted, as shown in Fig. 1b, in a general for the engine.

Step 4: Apply the principle of conservation of mass and energy to each process system. Taking into account the objectives defined for the model, the subsystems of interest in the engine would be all those shown in Figure 1b related to the engine intake system. Mass and energy conservation balances was developed only for the subsystems of interest considering the following considerations: the camera as shown in Fig. 2a and the manifolds are considered as reservoirs of thermal energy, with output properties equal to those of the control volume. And finally, the two turbocharger systems and the cooling system are unified in a single system, which will be modeled as a single stage of exhaust expansion and compression in the intake line. The camera consists of two inputs (0 and 4) and one output (1), resulting in the mass balance as shown in equation (1).

a) b)

Fig. 2. Components of the gas engine, a) Mixing Chamber, b) Turbocharger

$$\frac{dm_{cam}(t)}{dt} = \dot{m}_0(t) + \dot{m}_4(t) - \dot{m}_1(t) \tag{1}$$

Finally, because the transfer of heat is neglected and there is no work done within this volume of a control, the energy balance is expressed as equation (2).

$$\frac{dT_1(t)}{dt} = \frac{\dot{m}_0(t)C_{p0}(t)T_0(t) + \dot{m}_4(t)C_{p4}(t)T_4(t) - \dot{m}_1(t)C_{p1}(t)T_1(t)}{m_{cam}(t)C_{v1}(t)} \tag{2}$$

Turbocharger

In the mass balance of the turbocharger shown in Fig. 2b, you have to enter a mass flow $\mathbf{\dot{m}_1(t)}$ and a single mass flow comes out $\mathbf{\dot{m}_2(t)}$, therefore the dynamics of the mass in the compressor $\mathbf{m_{com}(t)}$ is given by equation (3).

$$\frac{dm_{comp}(t)}{dt} = \dot{m}_1(t) - \dot{m}_2(t) \tag{3}$$

Regarding the energy balance in this equipment, the energy consumed by the compressor is described $\mathbf{\dot{W}_{com}(t)}$ depending on the energy difference of the fluid as follows in equation (4)[25].

$$\dot{W}_{comp}(t) = \frac{\dot{m}_1(t)}{\eta_{comp}} \frac{kR_1(t)\left(T_{comp}(t) - T_1(t)\right)}{k-1} \tag{4}$$

The dynamics of the mass in the turbine $\mathbf{m_{turbina}(t)}$, an similar analysis to that developed on the compressor is carried out, as shown in equation (5), where $\mathbf{\dot{m}_9(t)}$ is the exhaust mass flow of the exhaust manifold and $\mathbf{\dot{m}_{10}(t)}$ the mass flow of the exhaust gases [25].

$$\frac{dm_{turb}(t)}{dt} = \dot{m}_9(t) - \dot{m}_{10}(t) \tag{5}$$

Finally, the energy balance of the turbine allows determining the energy produced by the turbine $\mathbf{\dot{W}_{turbina}(t)}$, which is represented in equation (6).

$$\dot{W}_{turb}(t) = \dot{m}_9(t) C_{pturb}(t)\left(T_9(t) - T_{10}(t)\right) \tag{6}$$

Step 5: Select from the DBEs those with valuable information to meet the objective of the model. Since this system assumes a total delivery of the energy from the turbine to the compressor, it is necessary to consider the equations (7) and (8).

$$\dot{W}_{comp}(t) = \dot{W}_{turb}(t) \tag{7}$$

$$\frac{\dot{m}_1(t)}{\eta_{comp}} \frac{kR_1(t)\left(T_{comp}(t) - T_1(t)\right)}{k-1} = \dot{m}_9(t) C_{pturb}(t)\left(T_9(t) - T_{10}(t)\right) \tag{8}$$

Step 6: Define for the essential EDBs, the parameters, variables and known constants of each Process System. The variables for the essential EDBs previously raised are the mass flows of each process system ($\dot{m}_1(t)$, $\dot{m}_2(t)$, $\dot{m}_9(t)$ y $\dot{m}_{10}(t)$ for the turbocharger) and the energy flows associated with different thermodynamic phenomena (Mass flow temperatures, output work and heats of, input and output, in this case $T_1(t)$, $T_2(t)$, $T_4(t)$, $T_7(t)$, $T_8(t)$, $T_9(t)$ and $T_{10}(t)$ respect to the flow temperatures and T_{com} and T_{tur} for the energy flows). The constants only focus on thermodynamic properties of the flows in question, such as the specific heats and their ideal gas constants.

Step 7: Finding constitutive equations that allow calculating the highest number of parameters in each process system. Because the framework of these paper focuses on developing a phenomenological semi-physical mean value engine model, additional equations are required to those raised in the previous steps of the methodology. For the implementation of these equations, the line of considering them according to their corresponding subsystem is followed:

Turbocharger

For this subsystem, relations for the temperatures and pressures of both the compressor and the turbine are needed. Besides, it is important to take into account the efficiency of isentropic state changes of those variables, as shown below in equation (9) and (10)[25].

$$\frac{T_2(t)}{T_1(t)} = 1 + \frac{1}{\eta}\left(\left(\frac{P_2(t)}{P_1(t)}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{9}$$

$$\frac{T_9(t)}{T_{10}(t)} = 1 + \frac{1}{\eta}\left(\left(\frac{P_9(t)}{P_{10}(t)}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{10}$$

Turbo by-pass Valve

It is considered that this subsystem accumulate mass, and therefore it is only able to instantly modify the flow of gases that passes through it. This flow ($\dot{m}_4(t)$) is obtained from the equation (11), where CV_{valtb} is the flow coefficient of the valve, $P_3(t)$ and $T_3(t)$ are the input pressure and temperature, S_{gtb} is the specific gravity of the gas mixture and $\Delta P_{tb}(t)$ it is the pressure change generated by the valve [26].

$$\dot{m}_4(t) = \frac{CV_{valtb} P_3(t) S_g}{0.00976} \sqrt{\frac{\Delta P_{tb}(t)}{T_3(t) P_3(t) S_{gtb}}} \frac{0.45}{3600} \tag{11}$$

It should be noted that in this expression the pressure and temperature units are in psia and Rankine degree respectively. For the calculation of CV_{valtb}, the polynomial expression shown in equation (12) was used.

$$CV_{valtb} = 15.82 + 0.7317 AP_{tb} + 0.0191 (AP_{tb})^2 \tag{12}$$

Throttle valve

As in the previous subsystem, this one is not able to accumulate mass, therefore only its flow ($\dot{m}_6(t)$), can be calculated, which is obtained similarly to the turbo bypass valve as shown in equation (13) where CV_{valth} is the valve's flow coefficient, $\rho_{6(t)}$ is the density of the gas mixture, $\Delta P_{th}(t)$ is the pressure change generated by the valve and S_{gth} is the specific gravity of the gas mixture [25].

$$\dot{m}_6(t) = CV_{valth} \sqrt{\frac{\Delta P_{th}(t)}{S_{gth}} \frac{\rho_{6(t)}}{3600}} \tag{13}$$

For the calculation of CV_{valth}, the equation (14) was used.

$$CV_{valth} = 1492.84355 (\Delta P_{th}(t))^{-0.496} \tag{14}$$

Intake manifold

Unlike the previous items, this one does have the capacity to contain mass, but due to the framework of the present work, this is not of vital relevance for the same. Variables that interest this subsystem are pressure and temperature, so for the case of the pressure in the intake manifold ($P_7(t)$) is calculated as equation (15), where K_{adm} is the coefficient of losses by sudden expansion in the intake manifold, $V_6(t)$ is the velocity of the fluid in the inlet pipe to the engine, V_{ad} is the available volume in the manifold, and $\rho_6(t)$ and $\rho_7(t)$ are the mass input and output flows [25].

$$P_7(t)=\rho_7(t)\left(\frac{P_6(t)}{\rho_6(t)}+\frac{(V_6(t))^2}{2}-K_{adm}\frac{(V_6(t))^2}{2}\right) \tag{15}$$

In addition, the outlet temperature is expressed as shown in equation (16), where $\forall_{adm}$ and m_{adm} are the volume and mass of gas in the intake manifold respectively.

$$T_7(t)=\frac{P_7(t)\forall_{adm}}{m_{adm}R_7(t)} \tag{16}$$

For the mass in the manifold was used the equation (17).

$$m_{adm}=\forall_{adm}\rho_7(t) \tag{17}$$

Step 8: Verify the degrees of freedom of the model. It was verified that the number of equations coincides with the number of unknowns of the model, in addition to relating the total number of unknowns raised as an object of study.

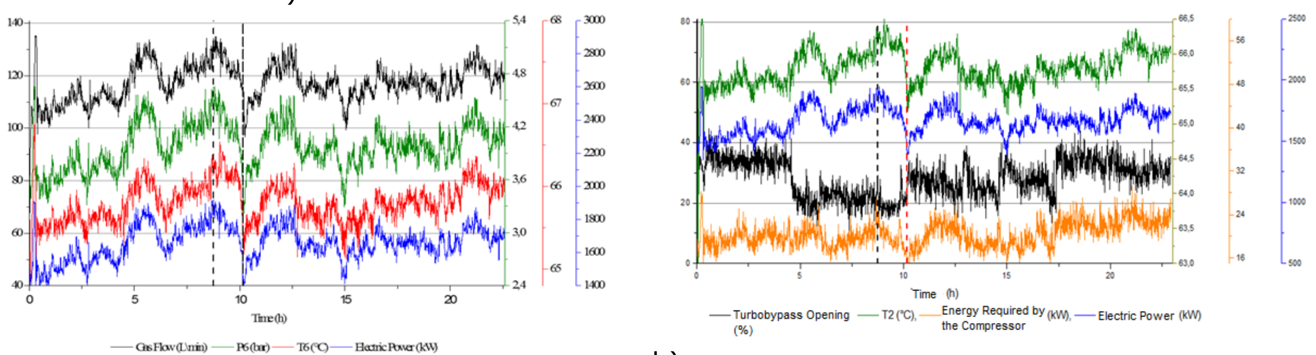
Step 9: Obtain the Computational model or solution of the mathematical model. For the calculation of simultaneous differential and algebraic equations, Matlab 2014b® was used, using the "ODE" commands, which in turn are based on high-order Runge-Kutta methods.

4. Results

Step 10: Validate the model for different conditions and evaluate its performance. Fig. 3a shows the variables Gas Flow, Pressure in 6 and Temperature in 6. This figure shows a clear proportional relationship between these variables, which is not surprising since this is evidenced in [27]. This information also could be confirmed via data analysis, for example near to 8.8 hours a 7.34% increase in pressure 6 generates a proportional 1% increase in temperature 6, 7.25% increase in electrical power and 0.25% decrease in gas flow. Although this reference does not say that the volumetric flow of gas is proportionally linear concerning the rest of the variables presented in the figure, it had been associated with the Lambda air excess coefficient, which if it is enunciated in [27].

In addition, Fig. 3b shows the turbo-bypass valve opening percentage, the temperature at 2 and the work required by the compressor. By looking more closely, it can also be noticed a proportional relationship between the temperature at 2 and the energy needed for the compressor. This last relation is not entirely illogical since if the previous approach of the model is detailed, it can be seen that there is a directly proportional between the discharge temperature of the compressor and the exit temperature of the intercooler. Finally regarding the percentage of opening of the turbo-bypass valve, so that its effects on the other variables are significantly observable it is necessary that it takes values much higher than 30% opening.

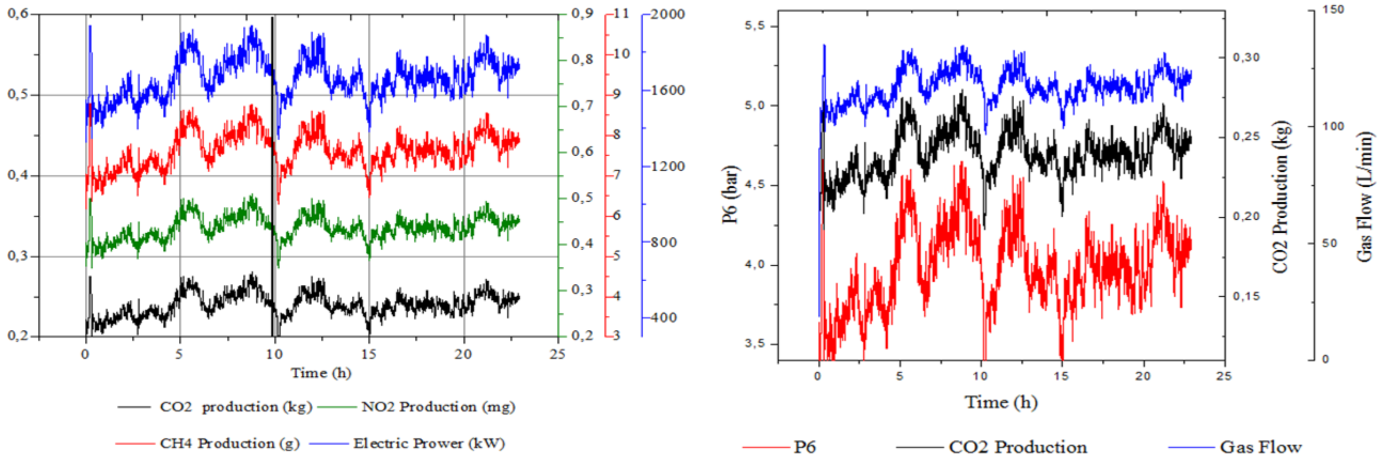
a)



b)

Fig. 3. a) Non-Standart Gas Volumetric Flow, Pressure at 6 and Temperature in 6 vs. Electric Power, b) Turbo-bypass Opening Percentage, Temperature in 2 and Energy required by the Compressor Vs. Electric power.

On the other hand, Fig. 4a shows the electric power in contrast with conventionally generated emissions such as CO_2, NO_2 and CH_4 mass generated. In times close to 10 hours of operation before an increase of 0.85% of the electrical power produced there are increases of 1.3% in the generation of CO_2 and NO_2 and a rise of 0.96% in the production of CH_4. This proportional behavior can be observed throughout the entire graph, which confirms that strategies aimed to reducing the consumption of fossil fuels, or replace them, are the most suitable to ensure the environmental sustainability of the process.



a)

b)

Fig. 4. a) CO_2 produced mass, NO_2 produced mass, CH_4 produced mass vs. Generated Electric Power. b) CO_2 produced mass, Pressure 6 Vs. Non-Standard Gas Volumetric Flow

Finally, Fig. 4b shows the gas flow in contrast with CO_2 emissions and Pressure in 6. Again a clear proportional relationship between the variables is noticed. In operational hours close to 10 hours of operation before a decrease of 0.25% of the nonstandard volumetric gas flow, there are increases of 7.34% in pressure 6 and 1.3 % rise in the generation of CO_2. This suggests that despite the control of the gas flow and the mixing conditions of the gas with air (pressure 6), the contaminants generation is not entirely mitigated and, on the contrary, poorly regulated.

5. Conclusions

According to second step of the applied methodology, a proportional relation was found between the generated electrical power, the inlet pressure, the inlet temperature of the air/fuel mixture and the volumetric gas flow. Regarding the position of the turbo bypass valve, this variable affect the electric power generated, but not in the same proportions such as the natural gas flow, the inlet pressure, and the temperature of the air/fuel mixture on the admission, and therefore, the gas emission composition.

Finally, for the second question proposed in the second step related to the engine emissions, it was found a proportional relationship between the operational variables such as the mixing temperature, the pressures of the intake manifold, the opening of the turbo bypass, and the engine power to the generation of pollutants. As a result, operating conditions of high gas demand and low recirculation percentages in the turbobypass valve were presented where the concentration of CO_2, NO_2, and CH_4 emissions increased by around 20% with respect to the nominal operating condition. This strengthens the idea that in the future other environmental friendly types of fuels should be evaluated, or a possible slow but necessary replacement of both consumer level as an industrial level.

The obtained model give goods results when modeling the intake line of the gas generation engine to predic the gas emissions, and the termal operation conditions inside the systems. Despite this good performance of the engine model, it is recommended in the future based on what has been developed in this work, to generate online strategies of mitigation and control of pollutants in this type of internal combustion engines.

REFERENCES

[1] K. Wark, *Advanced thermodynamics for engineers*. New York: McGraw-Hill Education, 1995.

[2] J. Karlsson and J. Fredriksson, "Cylinder-by-Cylinder Engine Models Vs Mean Value Engine Models for use in Powertrain Control Applications," *Society*, no. 724, 1998.

[3] F. Payri, P. Olmeda, J. Martín, and A. García, "A complete 0D thermodynamic predictive model for direct injection diesel engines," *Appl. Energy*, vol. 88, no. 12, pp. 4632–4641, 2011.

[4] M. Lapuerta, O. Armas, and J. J. Hernández, "Diagnosis of DI Diesel combustion from in-cylinder pressure signal by estimation of mean thermodynamic properties of the gas," *Appl. Therm. Eng.*, vol. 19, no. 5, pp. 513–529, 1999.

[5] C. . Rakopoulos, D. . Rakopoulos, E. . Giakoumis, and D. . Kyritsis, "Validation and sensitivity analysis of a two zone Diesel engine model for combustion and emissions prediction," *Energy Convers. Manag.*, vol. 45, no. 9, pp. 1471–1495, 2004.

[6] O. Armas, "Diagnóstico experimental del proceso de combustión en motores Diesel de inyección directa," Universidad Politécnica de Valencia, 1998.

[7] J. Martín, "APORTACIÓN AL DIAGNÓSTICO DE LA COMBUSTIÓN EN MOTORES DIESEL DE INYECCIÓN DIRECTA," Universidad Politécnica de Valencia, 2007.

[8] M. Kao and J. J. Moskwa, "Turbocharged Diesel Engine Modeling for Nonlinear Engine Control and State Estimation," *J. Dyn. Syst. Meas. Control*, vol. 117, no. 1, pp. 20–30, Mar. 1995.

[9] E. Hendricks and S. C. Sorenson, "Mean Value Modelling of Spark Ignition Engines." SAE International , 1990.

[10] E. Hendricks and S. C. Sorenson, "SI Engine Controls and Mean Value Engine Modelling." SAE International , 1991.

[11] D. J. Dohner, "A Mathematical Engine Model for Development of Dynamic Engine Control." SAE International , 1980.

[12] E. Hendricks, "Engine Modelling for Control Applications : A Critical Survey," *Meccanica*, vol. 32, pp. 387–396, 1997.

[13] E. Hendricks, T. Vesterholm, and S. C. Sorenson, "Nonlinear, Closed Loop, SI Engine Control Observers," 1992.

[14] E. Hendricks and T. Vesterholm, "The Analysis of Mean Value SI Engine Models." SAE International , 1992.

[15] E. Hendricks and J. B. Luther, "Model and observer based control of internal combustion engines," *Proc. Int. Work. Model. Emiss. Control Automot. Engines (MECA), Fisciano, Italy*, pp. 9–21, 2001.

[16] M. Müller, E. Hendricks, and S. C. Sorenson, "Mean Value Modelling of Turbocharged Spark Ignition Engines." SAE International , 1998.

[17] M. Fons, M. Muller, A. Chevalier, C. Vigild, E. Hendricks, and S. C. Sorenson, "Mean Value Engine Modelling of an SI Engine with EGR." SAE International , 1999.

[18] A. Chevalier, M. Müller, and E. Hendricks, "On the Validity of Mean Value Engine Models During Transient Operation." SAE International , 2000.

[19] A. Chevalier, C. W. Vigild, and E. Hendricks, "Predicting the Port Air Mass Flow of SI Engines in Air/Fuel Ratio Control Applications." SAE International , 2000.

[20] J. Z. Vasu, A. K. Deb, and S. Mukhopadhyay, "MVEM-Based Fault Diagnosis of Automotive Engines Using Dempster–Shafer Theory and Multiple Hypotheses Testing," *Syst. Man, Cybern. Syst. IEEE Trans.*, vol. 45, no. 7, pp. 977–989, 2015.

[21] J. Na, G. Herrmann, C. Rames, R. Burke, and C. Brace, "Air-Fuel-Ratio Control of Engine System with Unknown Input Observer," pp. 0–5, 2016.

[22] H. Alvarez, R. Lamanna, P. Vega, and S. Revollar, "Metodología para la Obtención de Modelos Semifísicos de Base Fenomenológica Aplicada a una Sulfitadora de Jugo de Caña de Azúcar," *Rev. Iberoam. Automática e Informática Ind. RIAI*, vol. 6, no. 3, pp. 10–20, 2009.

[23] General Electric, "JMS 612 GS-N.L (Descripción y Operación)." General Electric, 2012.

[24] J. Heywood, *Internal combustion engine fundamentals*. McGraw-Hill Education, 1988.

[25] L. Guzzella and C. Onder, *Introduction to modeling and control of internal combustion engine*

systems. Springer Science & Business Media, 2009.
[26] WoodWard, "F-Series Actuator and Integrated Throttle Body." WoodWard, 2013.
[27] GE ENERGY, "JMS 612 GS-N.L Descripción/operación." 2012.