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ENCIT-2018-0742 COMPUTATIONAL SIMULATION OF A FOUR-STROKE ETHANOL-FUELED INTERNAL COMBUSTION ENGINE USING DIESEL-RK SOFTWARE

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Abstract. *This paper presents a theoretical analysis of the performance of an four-stroke internal combustion engine, used in vehicles marketed in the national territory. All engine parameters, necessary for computacional analysis, and experimental data obtained in dynamometer, were ceded by an automaker present in Brazil. These data refer to tests using ethanol as fuel. The computer simulations were performed using Diesel-RK® software, that is a Russian origin tool, and is free for academic use. The computational model was validated by the available experimental data, and the average error of all the simulations performed, was 10.6% to specific fuel consumption, 11.2% to air mass flow rate, and 2.21% to torque.*

Keywords: *Internal Combustion Engine, Computational Simulation, Ethanol*

1. INTRODUCTION

The process of designing an automobile engine is not finalized simply after the end of the selection of materials, components, assembly, simulations, etc. Among so many of the last steps to be followed, and one of the most costly for the automakers today, is the calibration phase, where several sectors are involved, including the engine set exchange. With the intention of reducing financial and time expenses, companies begin to invest in numerical simulations capable of characterizing the operation of combustion engines.

This characterization, however, is not simple. One way, perhaps, of approaching a more realistic model would be to analyze each of the events that occur during the cycle of an internal combustion engine, in order to consider the various peculiarities that occur in each event, introducing the losses and imperfections. Even so, in the case of an internal combustion engine, it would not be enough to analyze the events predicted in the thermodynamic cycle, since other phenomena occur in addition to the thermodynamically analyzed ones, such as losses related to the contacts between the rod, crankshaft and bearings.

Fortunately, the technological advances provide us with the possibility of using computational resources to create models that are much more complex and that can compute all the above-mentioned particularities (or even others that go beyond this text), without great difficulties, requiring only that they be well defined.

In the present work, the software DIESEL-RK®, of Russian origin, will be used as simulator. This tool uses an online server, where the data are processed, to return results from the input that is inserted. The version of the software used is considered free for academic purposes.

2. DIESEL-RK® SOFTWARE AND ENTRY DATA FOR COMPUTATIONAL MODEL

The Diesel-RK® software began to be developed between 1981 and 1982, in the department of internal combustion engines, of the Bauman Moscow State Technical University (BMSTU) in Russia. In the beginning it was a tool used to facilitate the research in the area, which had a special concern with the time necessary to perform the simulations (Kuleshov, 2004).

The creation of a model in Diesel-RK® is done in five steps. The first one is the definition of the stroke number of the cycle, followed by the definition of the type of fuel used and the way its feed is made, as shown in Fig 1.

The second window defines the number of cylinders, their arrangement and the type of cooling of the motor, as shown in Fig. 2.

Then, the geometric parameters of the motor are defined, these being the diameter, the stroke and the compression ratio. In addition to these parameters, should be inform the nominal motor rotation, which is nothing more than the maximum torque rotation under fully open throttle conditions. The window with these parameters is shown in Fig. 3.

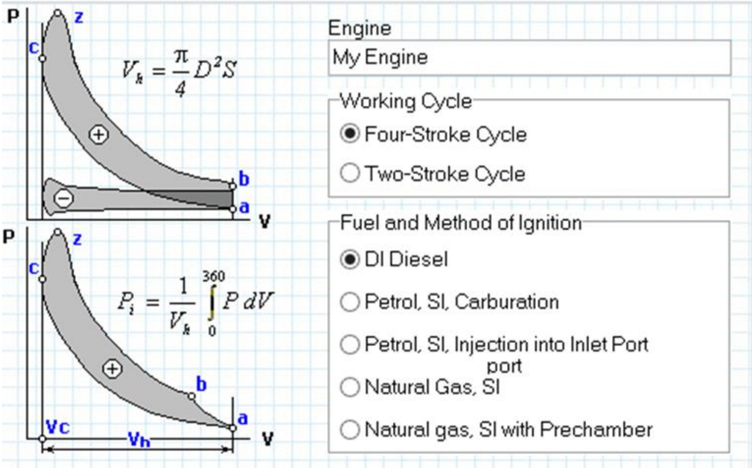


Figure 1. First project creation window in Diesel-RK® software.

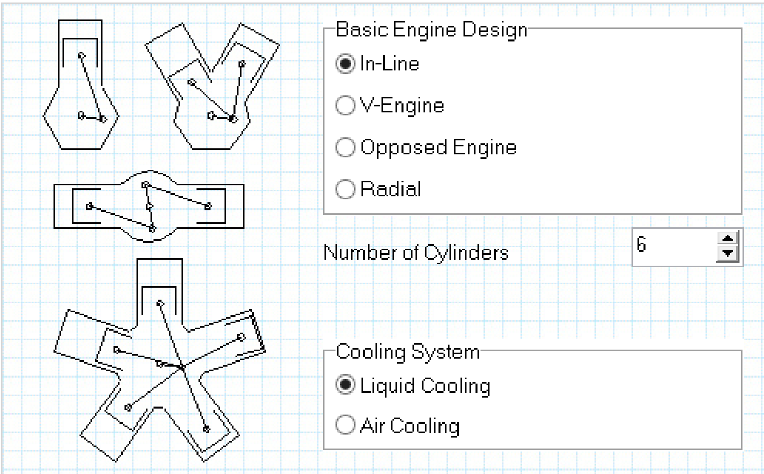


Figure 2. Second design window in Diesel-RK® software.

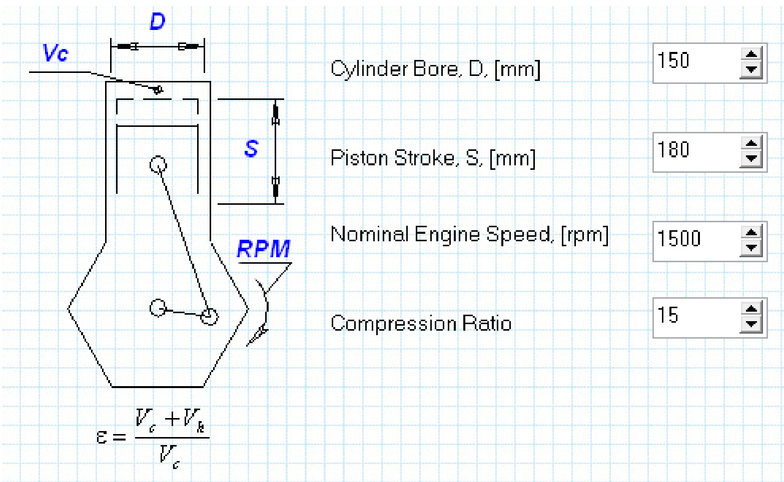


Figure 3. Third project creation window in Diesel-RK® software.

The fourth window defines the conditions of the standard environment and the application in which the engine will be submitted (overland and on the sea, submarine or aviation), as shown in Fig. 4. For applications other than overland and on the sea, it is necessary to insert some more information. In the case of engines for aeronautical use it is necessary to be informed of the height and speed of flight. For engines that will be used in underwater applications, it is necessary to know the depth at which the engine will be used.

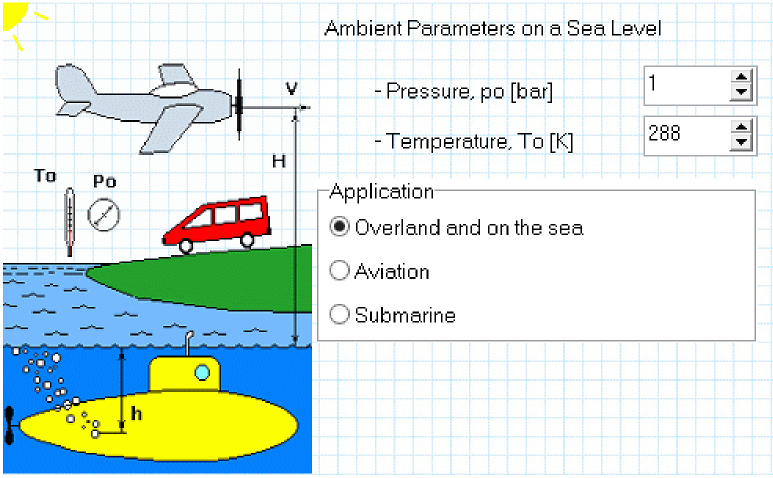


Figure 4. Fourth project creation window in Diesel-RK® software.

Finally, in the last window of creation of the model, it must be informed if the engine is supercharged or turbocharged, the number of valves on the cylinder head, and, in the case of diesel engines, the fuel injection pressure. If the engine is supercharged or turbocharged, it should be reported whether or not there is an intercooler. The window with these parameters is shown in Fig 5.

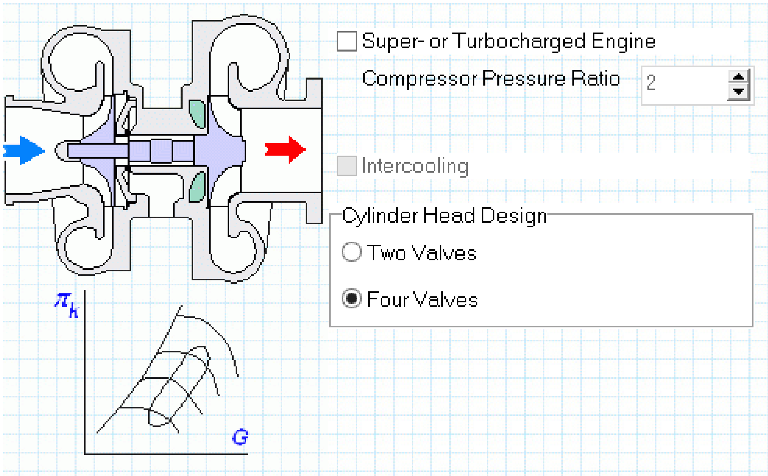
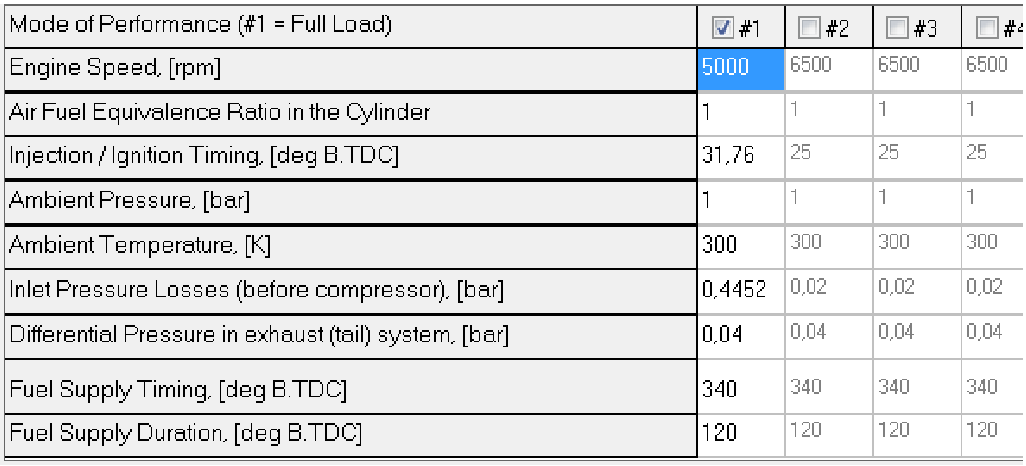


Figure 5. Fifth project creation window in Diesel-RK® software.

2.1 Control parameters of the computational model

Some of the parameters required for the simulations are not obtained analytically, but through the engine operating conditions. These data are related to the engine control, which are conditions imposed for its operation and can be easily changed by the test operator (in the case of experimental tests), or by the electronic control unit (in the case of the vehicle running on the road).

These electronically controlled parameters will be referred to here as control parameters and for each operating condition there is a value for these parameters. Figure 6 shows the window where these parameters are inserted, which are: the ignition point, the equivalence relation, the position of the throttle valve (which in this case is inserted indirectly, taking the ambient pressure and the fall of total pressure in the intake manifold), and the engine speed in revolutions per minute.



Mode of Performance (#1 = Full Load)	#1	#2	#3	#4
Engine Speed, [rpm]	5000	6500	6500	6500
Air Fuel Equivalence Ratio in the Cylinder	1	1	1	1
Injection / Ignition Timing, [deg B.TDC]	31,76	25	25	25
Ambient Pressure, [bar]	1	1	1	1
Ambient Temperature, [K]	300	300	300	300
Inlet Pressure Losses (before compressor), [bar]	0,4452	0,02	0,02	0,02
Differential Pressure in exhaust (tail) system, [bar]	0,04	0,04	0,04	0,04
Fuel Supply Timing, [deg B.TDC]	340	340	340	340
Fuel Supply Duration, [deg B.TDC]	120	120	120	120

Figure 6. Model Control Parameters Insertion window.

2.2 Thermal exchanges

The thermal exchanges considered in this work are only those performed with the cylinder, and are determined from the Woschni's correlation (1967). The coefficient of this correlation was used as the adjustment parameter of the model, since, as shown in Przybyla et al. (2013), the use of this parameter with a fixed value can generate errors greater than 100% in the global heat transfer coefficient. The Diesel-RK® allows this coefficient to be varied from 25 to 400. The location of the insert is shown in Fig 7.

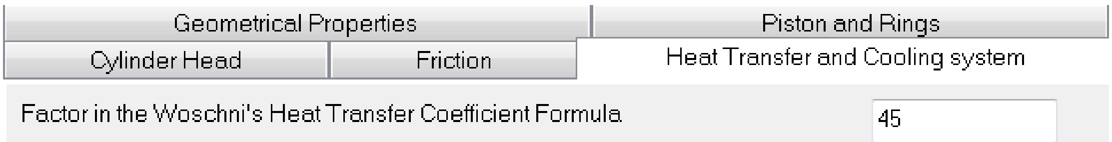


Figure 7. Woschni coefficient change window.

2.3 Fuel Parameters

Although the Diesel-RK® has an extensive fuel library, it does not contain the fuels used in Brazil. In this way, is needed to insert them into the software. Recalling that in this work the fuel used is ethanol.

Some of the parameters of the fuel were obtained experimentally (Calorific Heat and Density) in the Laboratory of Combustion, Propulsion and Energy (LCPE) of the Technological Institute of Aeronautics, and the others were found analytically through the equation presented by Lacava (2014). All the properties found and calculated for the fuel characterization in the software are summarized in Table 1.

Table 1. Parameters obtained for the equivalent fuel.

Equivalent Fuel	*C*	*H*	*O*
CxHyOz	1,736	5,472	1
COMPOSITION	-		
C [Mass Fraction]	0,4921		
H [Mass Fraction]	0,1302		
O [Mass Fraction]	0,3777		
Lower Calorific Power *MJ/kg* - Liquid fuel	24,4		
Fuel Density a 323 *K* in *kg/m3*	783		
Specific Heat of Vaporization *kJ/kg*	1019,22		
Thermal Capacity of Fuel at Injection Temperature *J/kgK*	2405,6		
Molecular mass *kg/kmol*	42,365		

2.4 Combustion Parameters

Diesel-RK® uses the model Wiebe's curve to characterize the combustion (Ferguson, 2001), and the definition of combustion end is fixed in the software as occurring when 99.9% of the total mass of the air-fuel mixture is consumed. However, it is necessary to know the duration of the combustion and the Wiebe form factor.

Fortunately, some of the parameters present in the experimental data used in this work refer to the mass burned fraction during combustion. In this way, it was possible, through an optimization method, to determine the duration of the combustion and the Wiebe form factor, for each of the operating conditions. Figure 8 shows the window for insertion of these parameters into the software, and Fig. 9 shows an example of a curve obtained for a given operating condition.

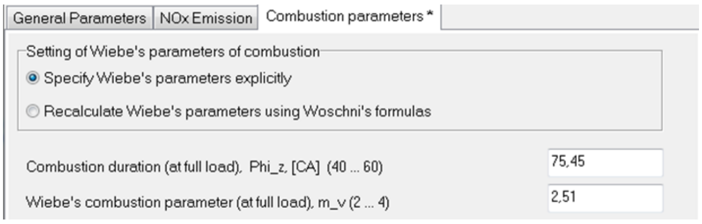


Figure 8. Input screen for Wiebe parameters.

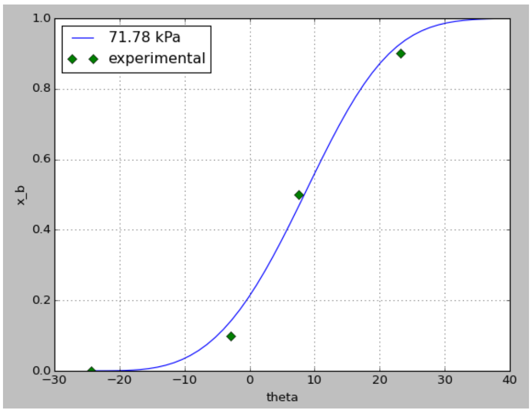


Figure 9. Regression for rotation of 3500 rpm and pressure in the intake manifold of 71.78 kPa.

3. EXPERIMENTAL APPARATUS

The experimental data used in this work were provided by an automaker present in the national territory, who prefers not to have their identity exposed. The tests were carried out within the premises of the company, using an AVL® DynoPerform 160 dynamometer and following the guidelines of the NBR ISO 1585 (ABNT, 1996).

During the test, the data required to create the computational model were obtained, such as: torque, intake manifold pressure, mass airflow, equivalence ratio, specific fuel consumption (sfc), and the mass burned fractions during the cycle. Figure 10 shows a photo of a test performed at the company's domains.

Figure 10. Experimental apparatus image.

4. RESULTS AND DISCUSSION

The data used refer to a four stroke engine, spark ignition of 1497 cm³ volumetric displacement, 3 cylinders, water cooling and using ethanol as fuel.

The computational model was adjusted from the experimental data assigned for the execution of this work, considering the environmental and operating conditions established during the tests. All control parameters and model creation are based on information provided by the company or taken from the experimental data.

The results presented in this work are in the form of percentages, for reasons of confidentiality of the company, that wishes to remain anonymous. Therefore, results such as the specific fuel consumption (sfc) and torque, which are usually presented in the form of *g/kWh* and *Nm*, respectively, will be presented as a percentage of a reference value, which cannot be reported either.

Figure 11 shows the comparison between the experimental and numerical results of Torque for a speed of 1750 rpm.

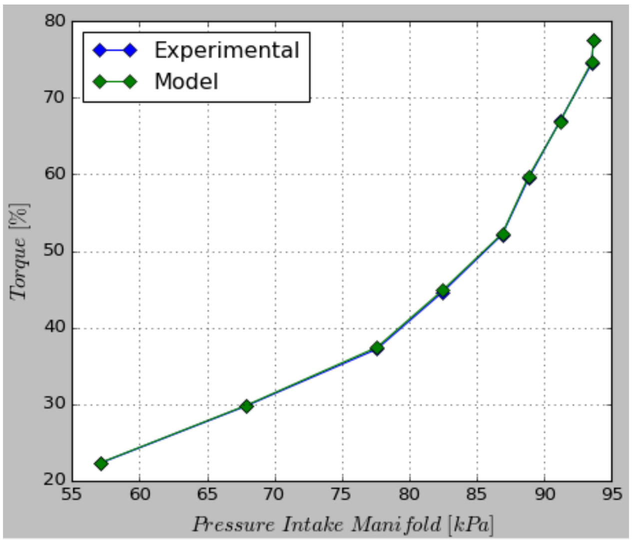


Figure 11. Torque results for the rotation of 1750 rpm.

Note that there is little difference between the experimental results and the model, and for this rotation, the mean deviation obtained is below 1%. This type of behavior is repeated for all the others engine speeds, so that torque results will no longer be displayed in this document as graphs.

Figures 12 and 13 show, respectively, the comparison between the experimental and numerical results of (sfc) and mass airflow, for the rotation of 1750 rpm.

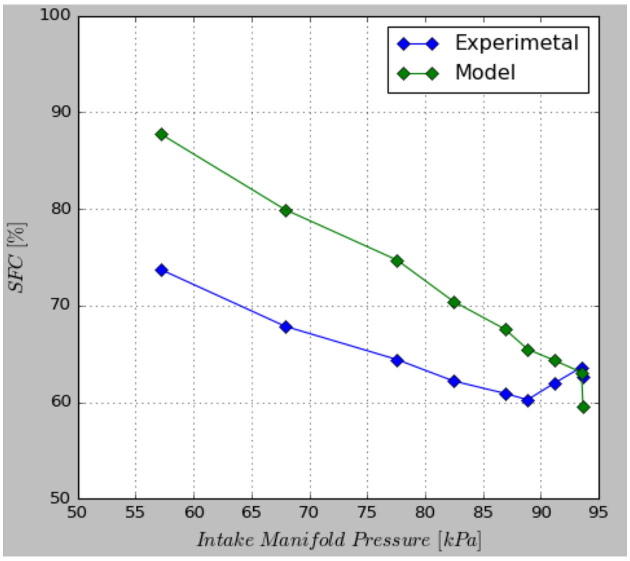


Figure 12. Results of (sfc) for the rotation of 1750 rpm.

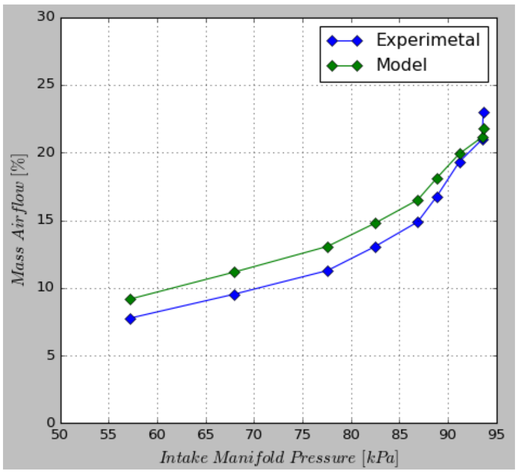


Figure 13. Results mass airflow for the rotation of 1750 rpm.

It is observed that the experimental results and the adjusted model have the same behavior, differing only by an additive deviation, both for the (sfc) and for the mass airflow. The deviations of the simulations for this rotation did not exceed 20% peaks of difference, being that in means these deviations are in the order of 10%, which is considered an acceptable value by the author.

Figures 14 and 15 show the same comparison of results, for a rotation of 2500 rpm.

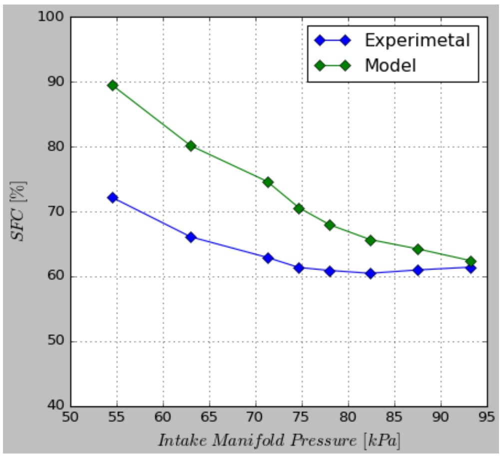


Figure 14. Results of (sfc) for the rotation of 2500 rpm.

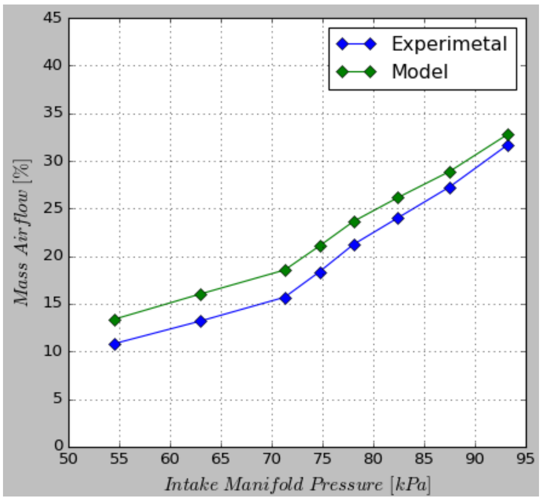


Figure 15. Results of mass airflow for the rotation of 2500 rpm.

As for the rotation of 1750 rpm, it is noted that the two results have the same behavior, again with an additive deviation. It is also perceptible that at low loads the model shows a greater distance from the experimental results; one of the possible causes of these deviations is the fact that a single friction model was used for the whole analysis, so that these losses may be overestimated at low loads. The average of the deviations found in this rotation is around 13% for the two grids analyzed.

Figures 16 and 17 show the same comparison of results, for a rotation of 3500 rpm.

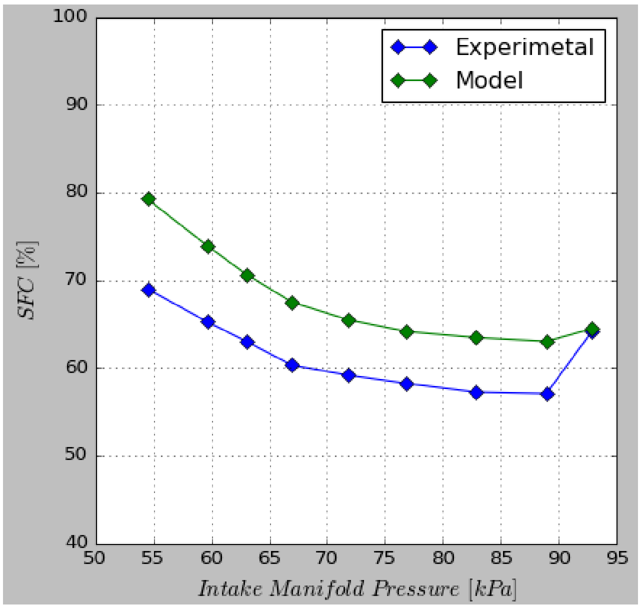


Figure 16. Results of (sfc) for the rotation of 3500 rpm.

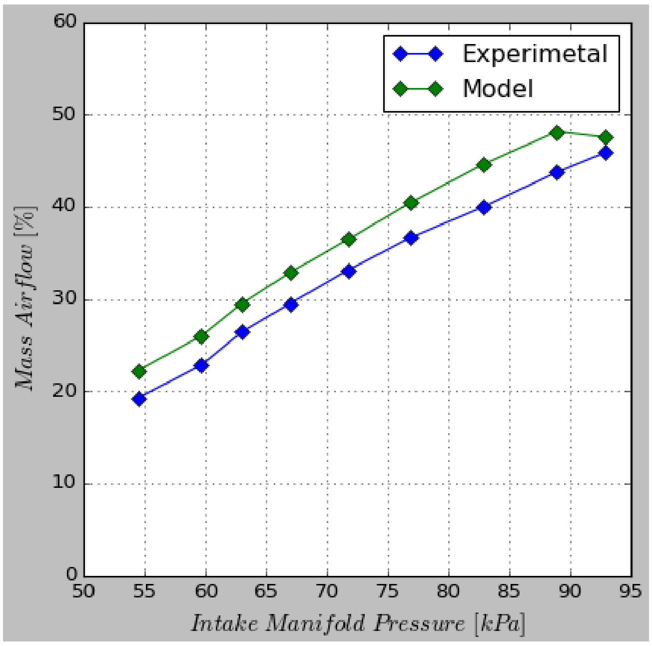


Figure 17. Results of mass airflow for the rotation of 3500 rpm.

For this rotation a more constant deviation is observed for the entire analyzed load range, and the behavior of these results remains the same. The mean deviation for this rotation is around 10%, for both parameters, as in the rotation of 1750 rpm, but these deviations have a smaller variation from one load to another, which can be considered a positive point.

Figures 18 and 19 show the same comparison of results, for a rotation of 5000 rpm.

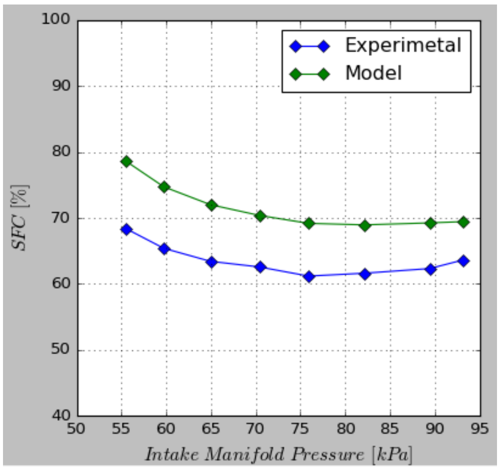


Figure 18. Results of (sfc) for the rotation of 5000 rpm.

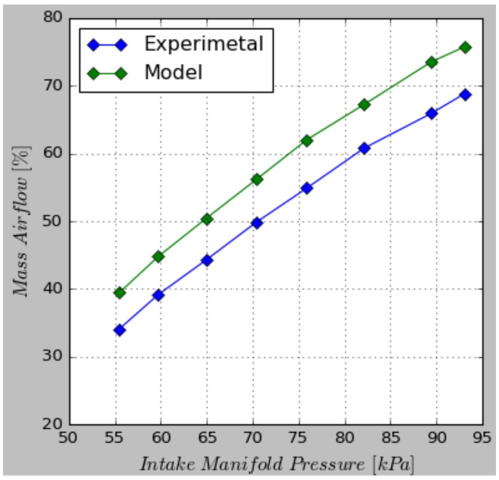


Figure 19. Results of mass airflow for the rotation of 5000 rpm.

As for the rotation of 3500 rpm, a constant deviation is observed for the entire analyzed load range, and the behavior of these results remains the same. The mean deviations for this rotation are around 13%, for both analyzed parameters, and the maximum deviations verified were of the order of 14.9%, that is, close to the mean for the analyzed range. Despite the average equal deviation, the author considers that these results are better than those obtained for the rotation of 2500 rpm, due to the fact that there is not a very large variation of the deviations from one load to the other.

Figures 20 and 21 show the same comparison of results, for a rotation of 6500 rpm.

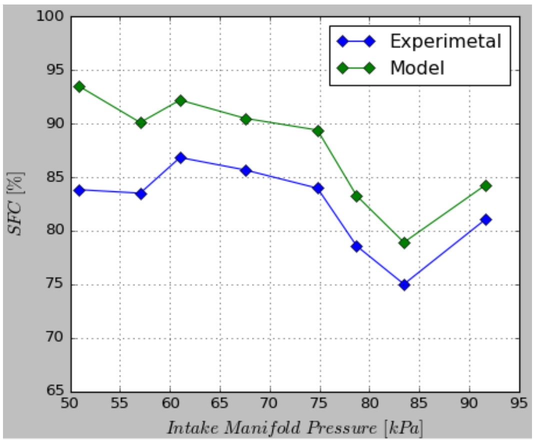
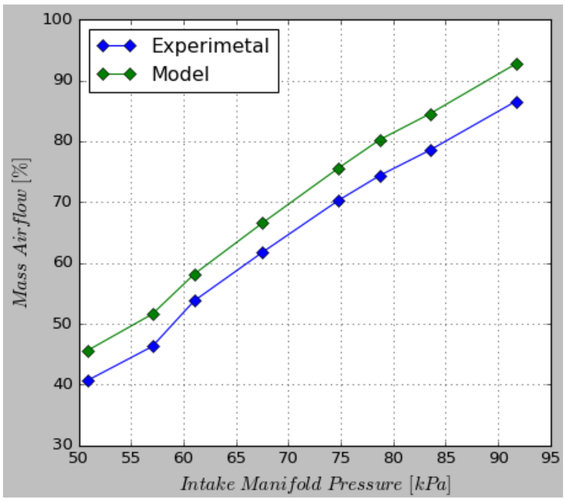


Figure 20. Results of (sfc) for the rotation of 6500 rpm.



21. Results of mass air flow for the rotation of 6500 rpm.

As for the previous results, it is noticed that the results of the model have the same behavior of the experimental results. For this rotation the mean deviations of (sfc) were 6.6%, and those of mass airflow around 8.7%, these results appear to be very satisfactory.

Figures 11 to 21 show the results of 5 engine speeds, which were selected in order to show the general behavior of the engine by its range of operation. However, the work analyzed a total of 14 speeds, with 121 different operating conditions, being that the mean deviation obtained for (sfc) is 10.6%, and for the mass air flow is 11.2 %. The mean deviations found for torque are 2.21%.

5. CONCLUSION

It is important to remember that the results are presented in the work in the form of percentages, because the disclosure of their absolute values would allow the company identity, which made available the experimental data, to be exposed. The results used are divided into 14 speeds from 1250 to 6900 rpm, with about 121 operating conditions. In the results of the adjusted model, an average deviation (sfc) of 10.6% was obtained, and for the air mass flow rate of 11.2%, the greatest differences were approximately 20% for both parameters, and even then only for a few isolated points.

The model presents better results with the increase of the rotation, however, it can be said that the computational model represents satisfactorily the motor analyzed.

The observed deviations are directly related to limitations in the calculation of the thermal changes and the friction, and more robust models for the determination of these quantities will be implemented in future works.

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