

## ENCIT-2018-0556 SIMULATION OF A GASOLINE POWERED FOUR-STROKE ENGINE USING THE DIESEL-RK SOFTWARE.

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**Abstract.** *This work presents a theoretical study of the performance of a 4-stroke gasoline internal combustion engine, used as propellant of some vehicles currently produced in Brazil. The experimental values were assigned by the responsible automaker, where they were obtained by means of bench dynamometer tests. The experimental values were entered in the free academic software Diesel-RK®. The results obtained by the software were compared with those obtained by means of experimental tests. The computational model was validated comparing these results obtained in the dynamometer considering the conditions of pressure and atmospheric temperature of 1 bar and 300 K, respectively. The adjusted model presented a difference of 13% and 14% for specific consumption and air mass flow respectively, and for the torque presented a difference of 2%, for the entire range of rotation and load studied, in relation to the experimental values*

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

### 1. INTRODUCTION

Internal combustion engines of the alternative type 4 times are widely used in diverse needs, mainly in the automotive area. The continuous evolution of the internal combustion engine, led by lowering the limits of consumption and polluting emissions, brings with it the need to design increasingly complex engines and sophisticated new technologies, able to manage all possible operating conditions. To meet the requirements of the standards and also of customers, surveys are carried out using current projects by computer resources, that allow the development of several engine simulations.

The simulation by means of engine models are critical as the aid for the creation of the control system of the respective subsystems. Currently the simulation of internal combustion engines alternative-type has been widely used by the factories and assembly plants with promising results, as it avoids some of the experimental trials, thereby reducing the cost and time of project. Doing the simulation of an engine enable the possibility of get results more quickly, rather than develop the entire project until you reach the stage of testing. It would take a long time until this engine being tested on test benches and the cost would be very high..

### 2. DEVELOPMENT OF COMPUTATIONAL MODEL

For the development of this work of computer simulation, it was used the version free academic program *Diesel-RK®*. This software was developed by Kuleshov, in 1982. Is currently in version 4.3.0.189. Developed in Moscow State Technical University, Bauman became a commercial version in 1991. Here are the main steps of development of a given model.

The Diesel-RK ® program starts with five windows to insert basic data from the desired, the following will be described. The first window has the function to select the work cycle, which can be 2 and 4 times, the type of fuel to be used and the method of injection of the engine to be designed. Fig. 1 shows the creation of a new project.

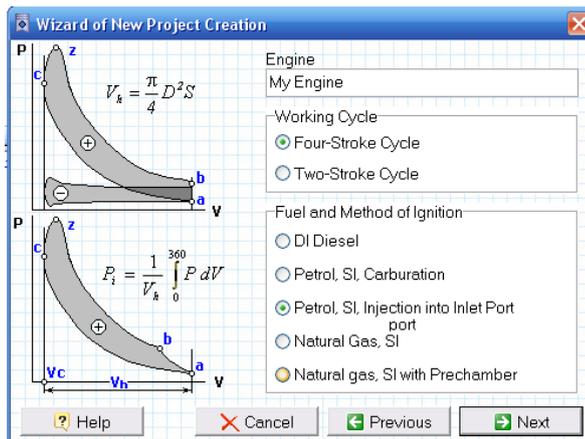


Figure 1. Creating a new project.

With the type parameters of the cycle and the fuel already chosen, the next step is to select in the basic design of the engine, if the same will have cylinders arranged in line, in "V", opposite cylinders, "motor boxer", or radial used in some aircraft. It is also necessary to supply the number of cylinders and the type of cooling system with liquid or air. Fig. 2 shows the cylinder layout models, the number of cylinders and the type of engine cooling (Kuleshov, 2008).

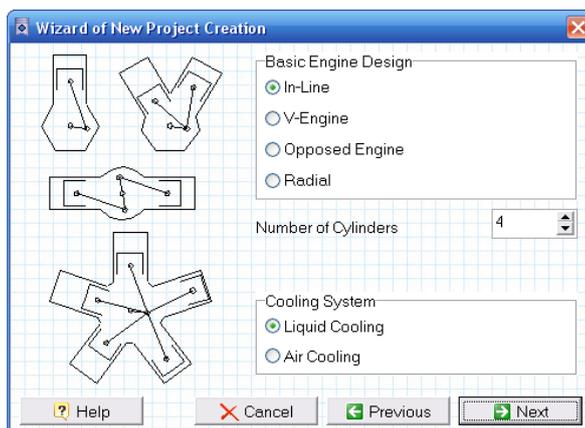


Figure 2. Models of cylinder layout and cooling type.

With the basic design of the preset engine, the next step is to insert the engine data, such as cylinder diameter, stroke, nominal rotation and compression rate, observing that the more accurate the data, the better the quality of the results obtained, thus making the model more reliable. Fig. 3 shows the interface with the fields to be filled with the engine data (Kuleshov, 2008).

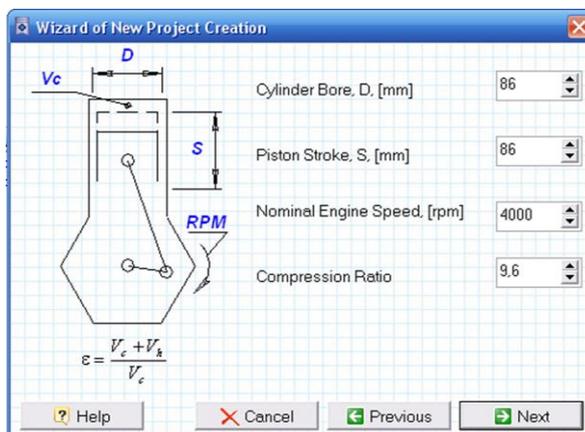


Figure 3. Interface to enter with engine data.

The program also allows you to select the environmental parameters in which you want to simulate the engine, such as temperature and pressure, and also allows the user to inform the application if the engine will work on land, sea, aviation and even submerged in the case of submarines (Kuleshov, 2008). Fig. 4 shows the fields of the model's environmental parameters.

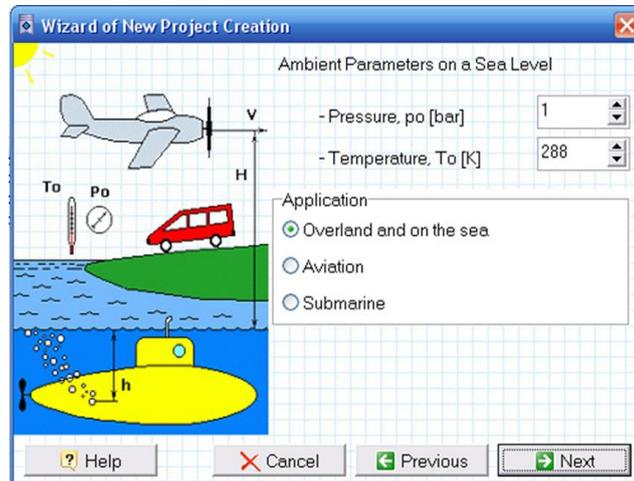


Figure 4. Environmental parameters of the model.

Finally the last window of the creation of a new model, allows the user to provide data if the engine will be turbo powered, also informing the pressure ratio of the turbine. Another important information to create a reliable model is to inform if the engine that is desired study will have two or four valves per cylinder (Kuleshov, 2008). Fig. 5 shows the fields mentioned above.

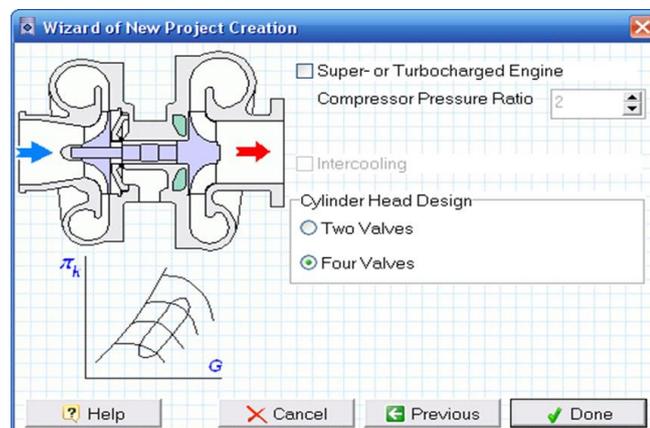


Figure 5. Interface of valve numbers per cylinder, and ratio of turbine pressure.

It can be noted that to create a model in Diesel-RK ® is not necessary a wide range of data, thus enabling a very fast model creation. But Diesel-RK ® allows us to further detail the insertion of data, thus making the model more and more reliable. The following will be shown and commented on some parameters that were used in this work beyond those cited.

## 2.1 Fuel parameters

The Diesel-RK ® software has a library containing several types of fuel used in piston engines, but it does not have a fuel equivalent to Brazilian gasoline (E27) which contains 27% in volume of anhydrous ethanol in its composition. Due to this fact, it was necessary to build a model of fuel that was characterized in the laboratory of combustion, propulsion and energy (LCPE) – (ITA), where the following values were obtained shown in the Tab. 1 for the equivalent fuel (Petrobras, 2015).

Table 1. Parameters obtained for the equivalent fuel.

COMPOSITION	-
<i>C</i> [Mass fraction]	0,7394
<i>H</i> [Mass fraction]	0,1491
<i>O</i> [Mass fraction]	0,1115
Lower calorific power KJ/kg-Liquid fuel	39000
Octane – IAD	90,00
Fuel density kg/m <sup>3</sup>	713
Specific heat of steaming kJ/kg	512
Thermal fuel capacity in J/KgK injection temperature	2200
Molecular mass kg/kmol	77,35
Fuel temperature K	323

## 2.2 Heat transfer coefficient

Diesel-RK ® allows the Woschni heat transfer coefficient to be varied from 25 to 400 using as gasoline fuel. Thus, the Woschni heat transfer coefficient was used to adjust the torque curve of the simulated values with the torque curve of the experimental values. This method was used to obtain the values of the heat transfer coefficient, because a fixed value for Woschni did not produce good results. There was the difficulty of adjusting the torque curves for the adjustment of the model. Fig. 6 shows the field for variation of the value of the Woschni heat transfer coefficient (Woschni, 1970).

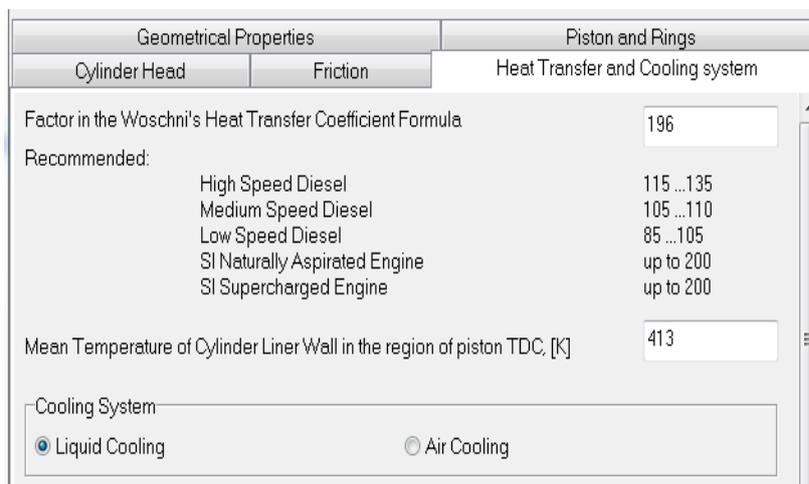


Figure 6. Woschni heat transfer coefficient.

## 2.3 Diesel-RK ® operating mode

The operating mode allows the choice of the simulation process, and can be the specific fuel mass of the air/fuel cycle ( $\lambda$ ), which was used in this work. The pressure in the collector can be calculated by Diesel-RK ® or directly imputed, which was the option chosen for that work. Fig. 7 shows the fields that have been inserted in this work: Rotation values (*RPM*); Ratio of air/fuel equivalence ( $\lambda$ ); Ignition Point ( $\theta_i$ ) and the pressure on the intake manifold for each rotation and its respective load. The values have to be adjusted again according to the experimental data given by the assembler before starting each simulation, totaling 121 different conditions.

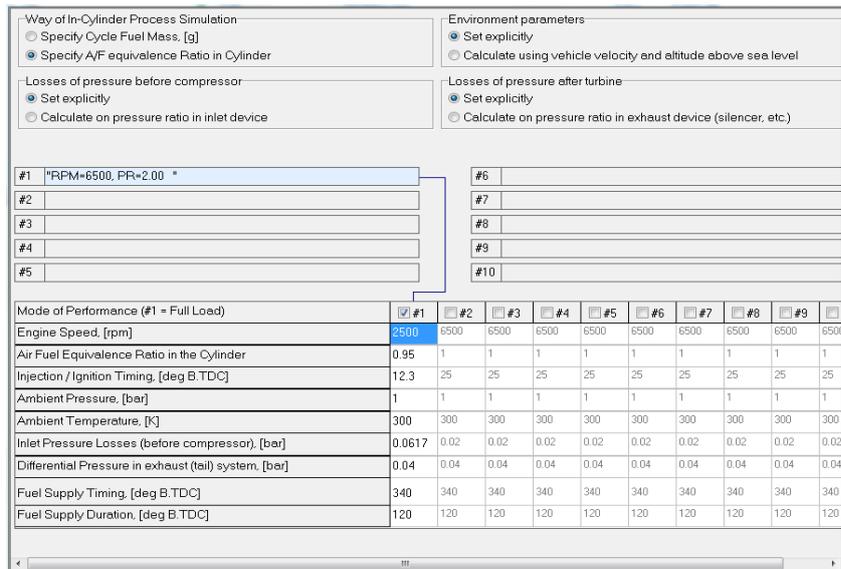


Figure 7. Diesel-RK ® operating mode.

### 3. RESULTS

As results are shown the torque curves, specific consumption and mass air flow of only five rotations 1250, 2000, 4000, 5500 and 6500 rpm among the fourteen studied. The computational model was adjusted with the experimental curves, taking as reference the torque curve. The variation in the value of the Woschni coefficient ( $C_w$ ) was made until the curves were perfectly adjusted to each other. In this way the simulated and experimental curves do not have significant differences as shown in Fig. 8. Due to this coincidence between the experimental and simulated curves is presented here only a result of torque.

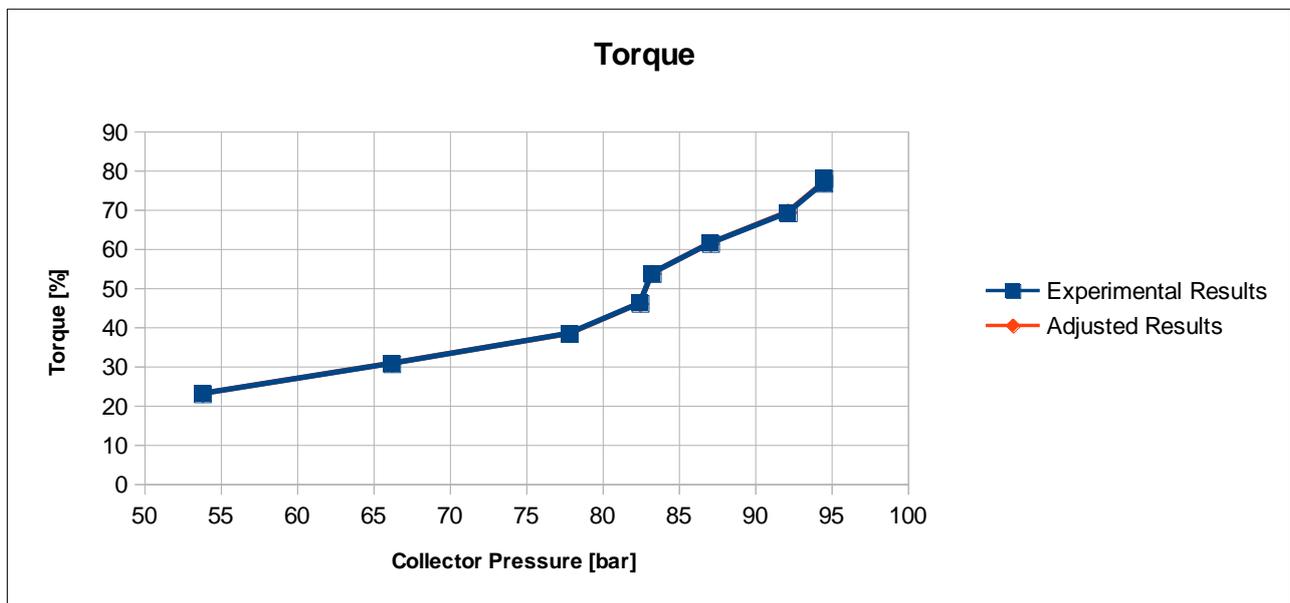


Figure 8. Comparison of torque curves by pressure on the experimental collector and obtained by Diesel-RK<sup>®</sup>.

With the torque curve adjusted, the behaviour of the specific consumption curves could be seen in Fig. 9, where the curve of the simulated results had a similar behavior to the experimental, having a maximum time deviation of approximately 29% and a medium deviation 16% for rotation of 1250 rpm.

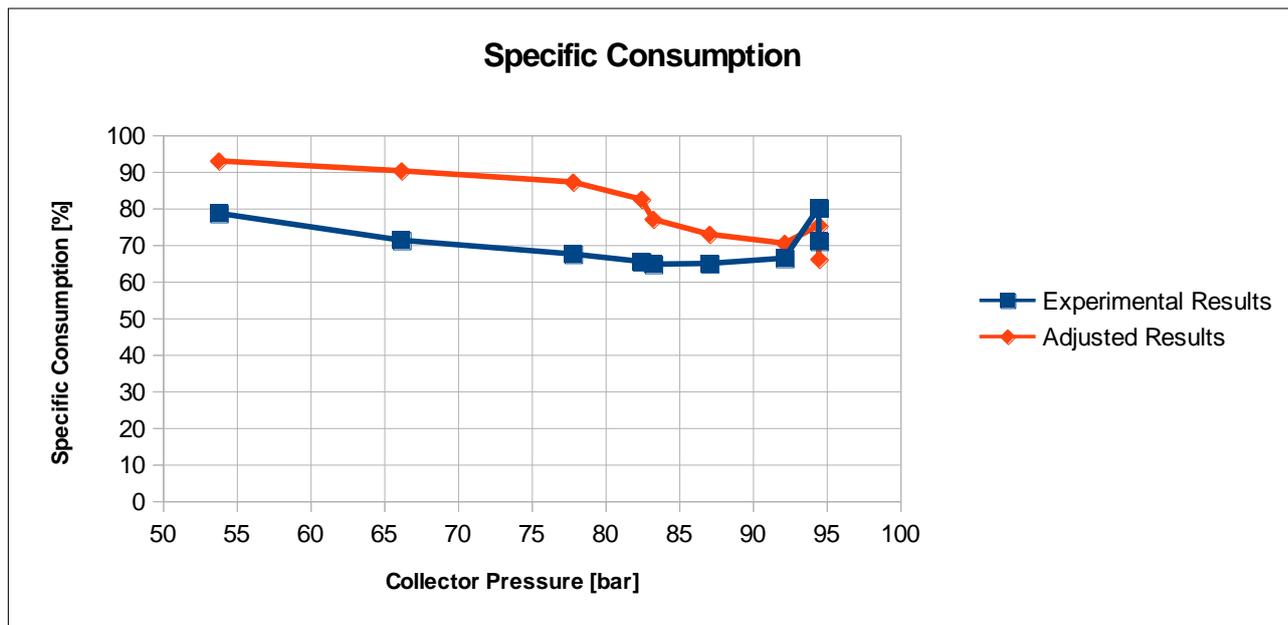


Figure 9. Comparison of the specific consumption curve by the pressure in the simulated and experimental collector, for rotation of 1250 rpm.

The mass flow curves of experimental and simulated air are shown in Fig. 10. It can be observed that the simulated curve presents a identical behavior to the experimental, having a maximum time deviation of 29.5%, and an average deviation of 16.4%, which can be considered satisfactory for this rotation of 1250 rpm.

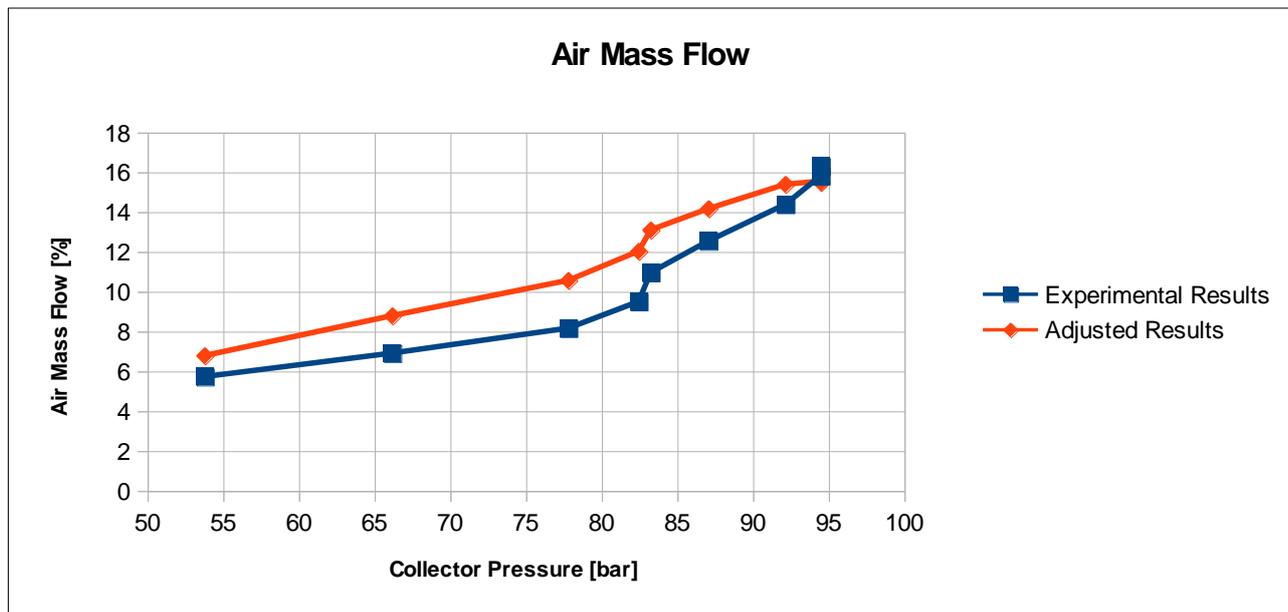


Figure 10. Comparison of experimental and simulated curves with the mass flow of air with the pressure in the collector, for rotation of 1250 rpm.

For the 2000 rpm rotation can be seen in Fig. 11, a maximum time deviation of 31%, and a minimum deviation of approximately 1% for specific consumption. However, it can be noted that the behavior of the simulated curve is similar to the experimental, having an average consumption deviation specific to that 16% rotation.

Fig. 12 shows the behavior of flow mass curves for this same rotation. The simulated curve shows a similar behavior to the experimental, having a maximum time deviation of 32% and an average of 17%.

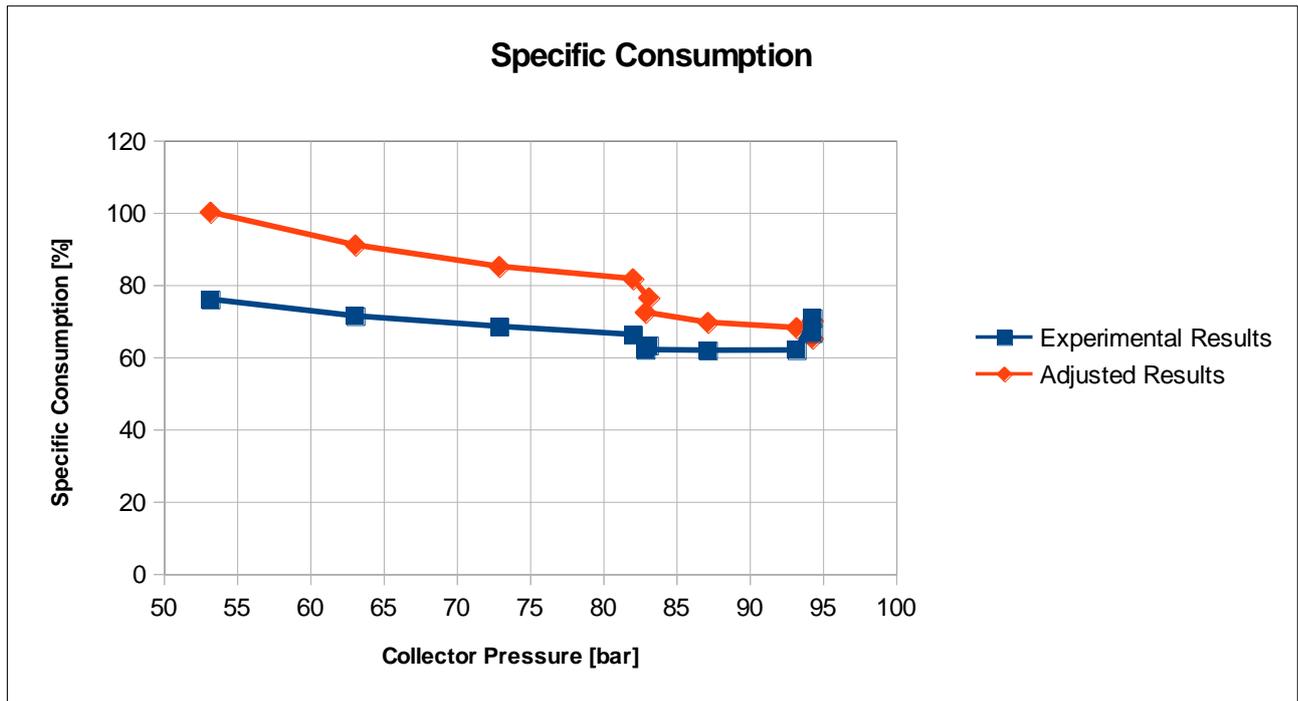


Figure 11. Comparison of the specific consumption curve by the pressure in the simulated and experimental collector, for rotation of 2000 rpm.

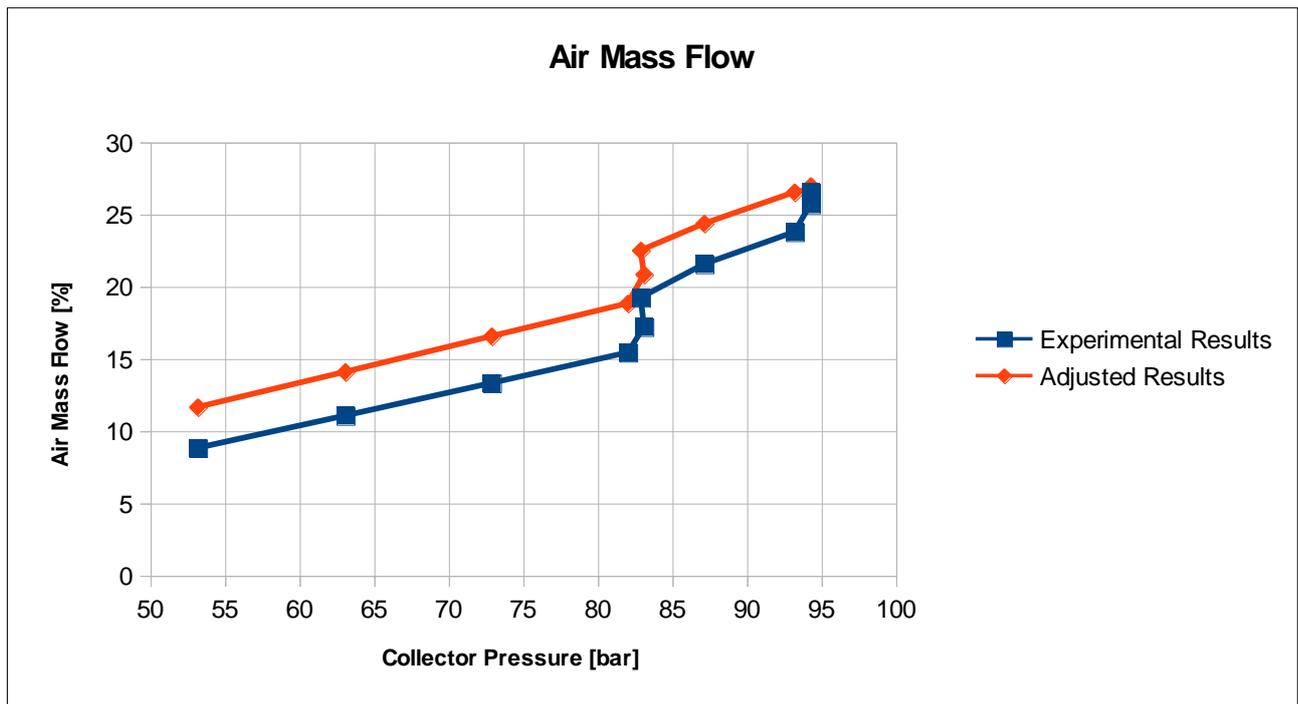


Figure 12. Comparison of the experimental and simulated curves of the mass flow of air by the pressure in the collector, for rotation of 2000 rpm.

Fig. 13 shows the behavior of the simulated curve with the experimental fuel consumption for 4000 rpm. With the increase in rotation it can be noted that the simulated curve has an even better behavior compared to those mentioned earlier. For this rotation the maximum punctual deviation was 15% and an average of 11%.

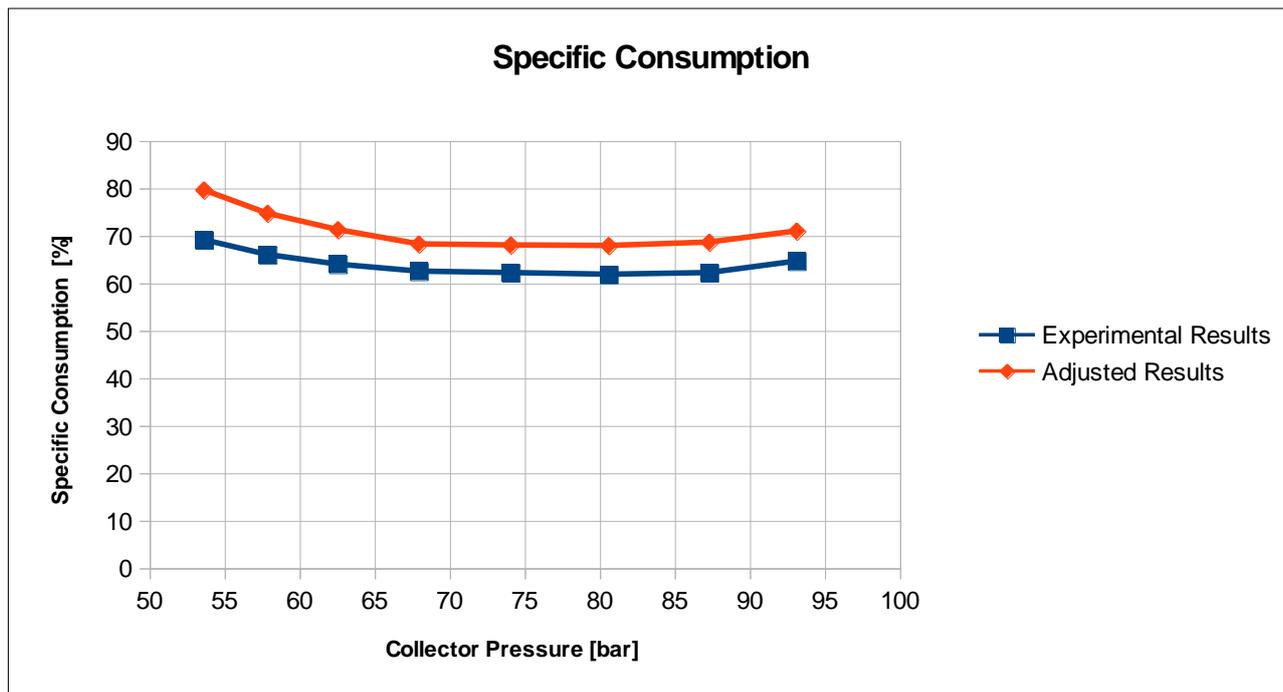


Figure 13. Comparison of the specific consumption curve by the pressure in the simulated and experimental collector, for rotation of 4000 rpm.

Figure 14 shows the behavior of the air mass flow curve for that same rotation. It can be noted that the simulated air mass flow curve also has better behavior with increased rotation, having a maximum time deviation of 15.4%, and an average of 11.6%.

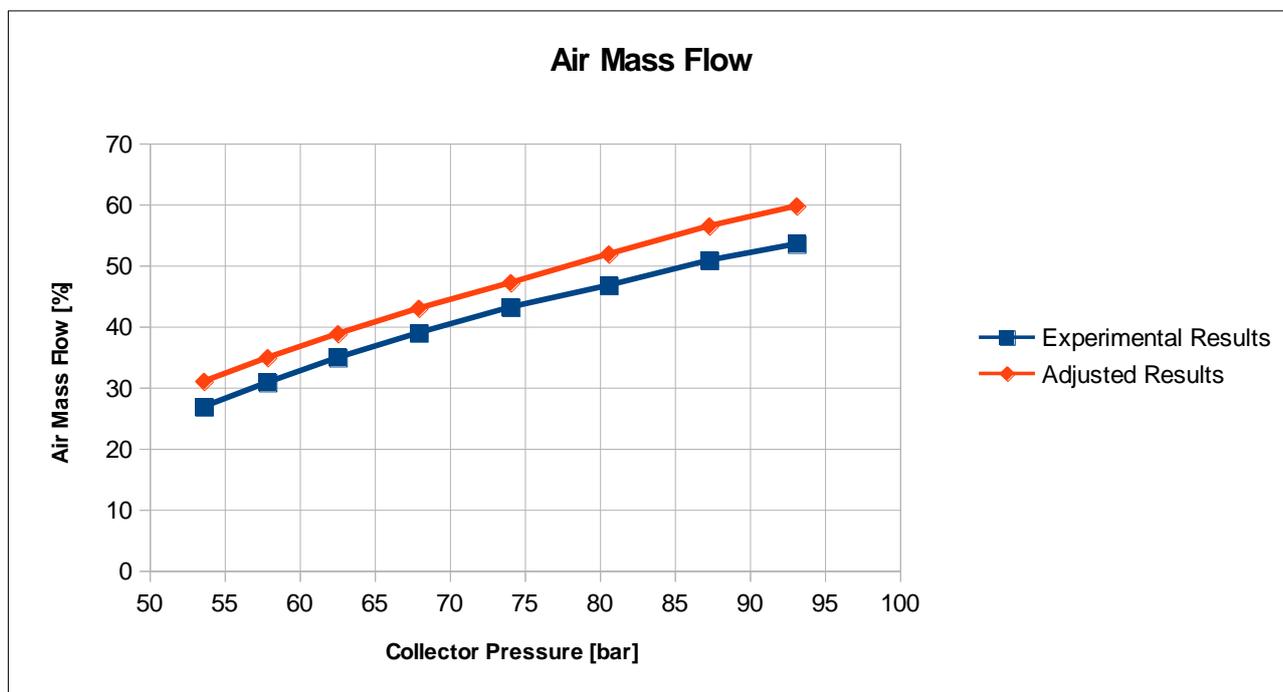


Figure 14. Comparison of experimental and simulated curves with the mass flow of air with the pressure in the collector, for rotation of 4000 rpm.

Fig. 15 shows the behaviour of the specific fuel consumption curve for the rotation of 5500 rpm. The simulated curve presents good behavior in relation to the experimental further improving the calibration of the model, because the maximum deviation for this rotation is not more than 14%, and the next medium by 10%.

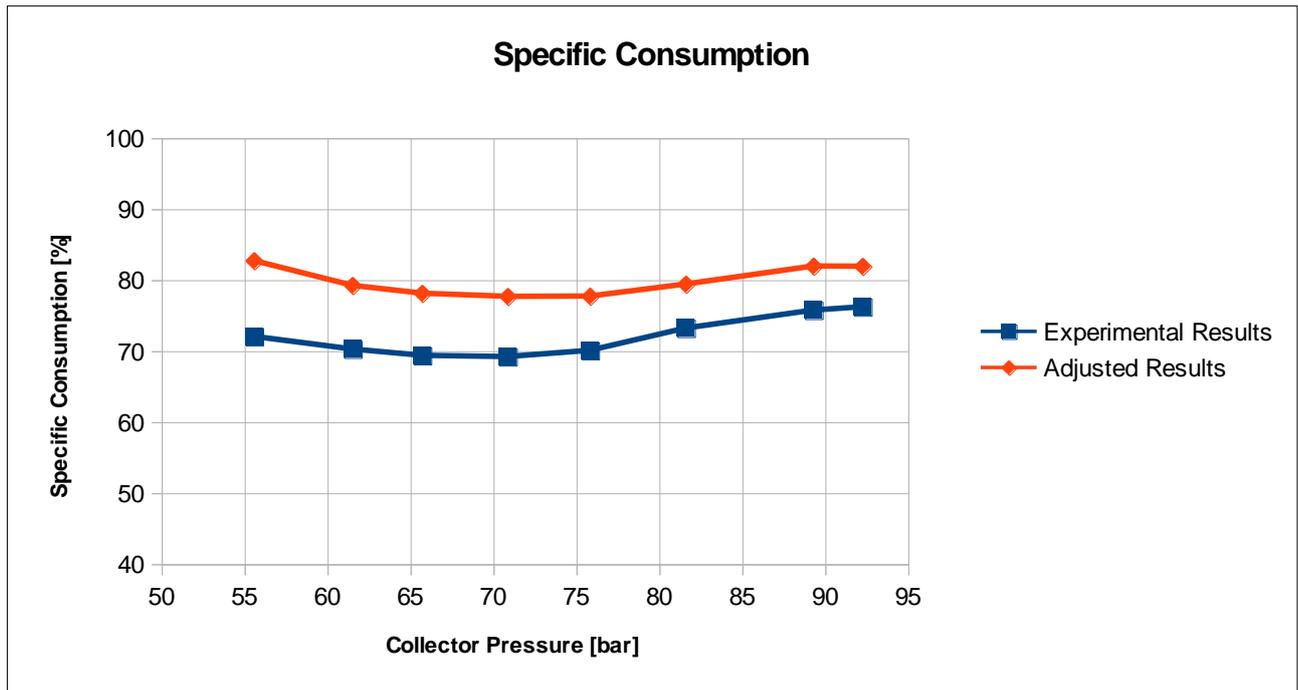


Figure 15. Comparison of the specific consumption curve by the pressure in the simulated and experimental collector, for rotation of 5500 rpm.

For the mass air flow, it can be observed that the behavior of the simulated curve in relation to the experimental curve also has a better behavior, thus obtaining a better adjustment with a maximum time deviation of 14%, and an average of 12%. Figure 16 shows the behavior of simulated and experimental curves with the mass flow of air to 5500 rpm.

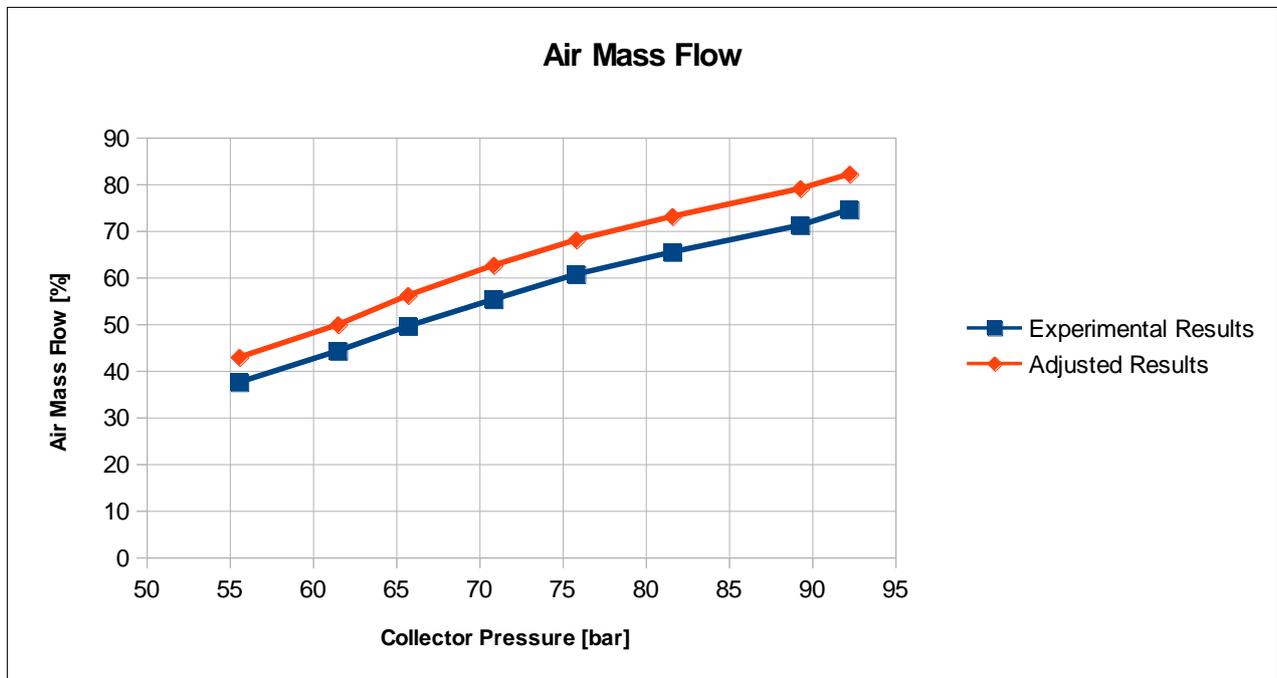


Figure 16. Comparison of experimental and simulated curves with the mass flow of air with the pressure in the collector, for rotation of 5500 rpm.

For the rotation of 6500 rpm, it can be observed in Fig. 17, the compartment of the specific consumption curves where the simulated curve has similar behavior to the experimental curve, having a maximum deviation of 3.3% and an average of 1.9%.

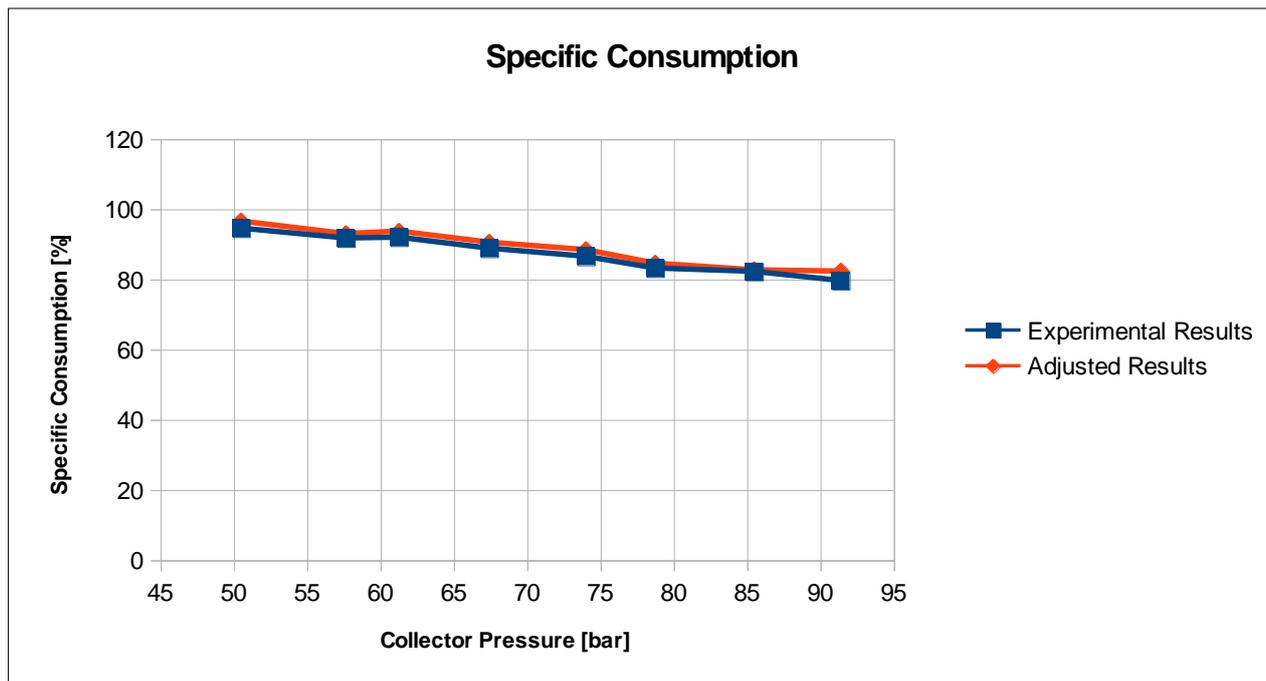


Figure 17. Comparison of the specific consumption curve by the pressure in the simulated and experimental collector, for rotation of 6500 rpm.

In Fig. 18 you can see the behaviour of the curves for mass air flow, where the maximum deviation does not exceed 5.9%. The average deviation was around 4.3%. It is noted that the behavior of the simulated curve is identical to the experimental curve.

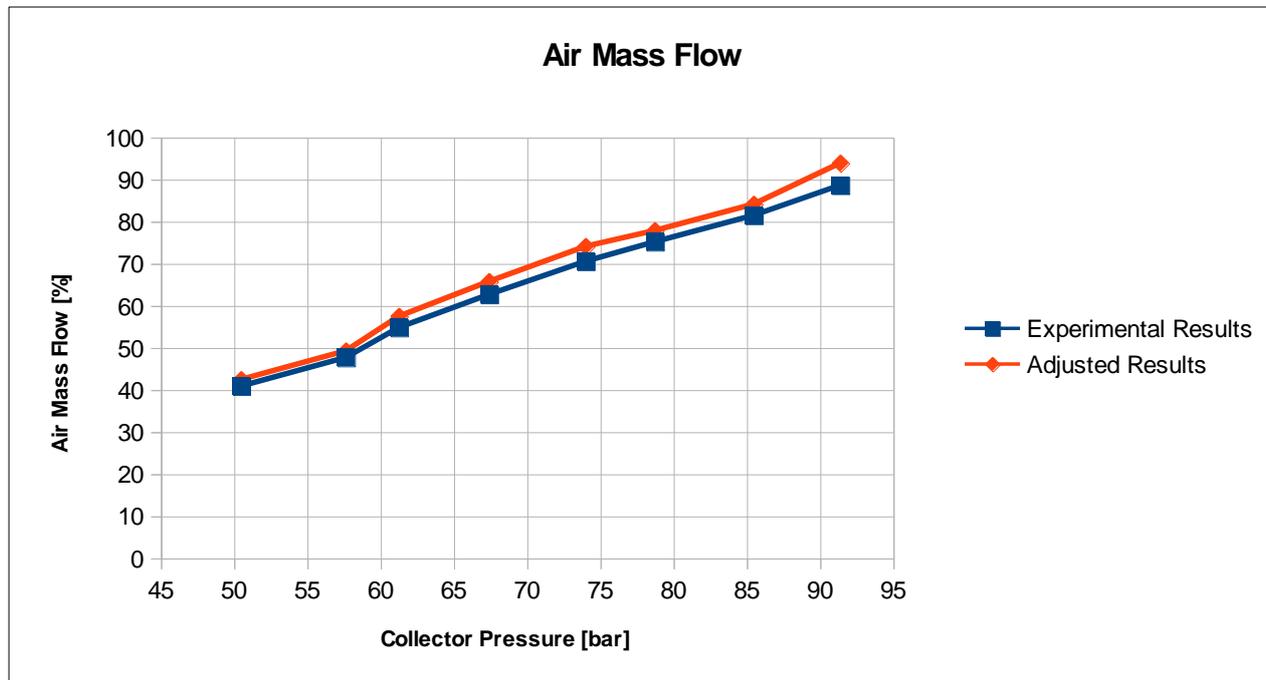


Figure 18. Comparison of the experimental and simulated curves with the mass flow of air with the pressure in the collector, for rotation of 6500 rpm.

#### 4. CONCLUSION

It can be observed that with the increase of rotation, the range of the deviation of the adjustment decreases, thus making the model more accurate. It is noted that the model has a higher deviation in low rotations and loads; This deviation is due to the thermal compensation inserted at the beginning of the simulation so that it was possible to adjust the model with the experimental.

This thermal compensation was inserted so that the model could be adjusted to the fourteen rotations with their respective loads. In this way, the adjustment of the model had an average deviation for specific consumption and mass air flow of 13% and 14% respectively, along the rotations, which is considered an acceptable deviation.

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