24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-1123
DESIGN AND ANALYSIS BY FINITE ELEMENTS OF A BASE FOR A RECIPROCATING COMPRESSOR

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Abstract. *In the study of dynamic systems, it is necessary to consider the behavior of structures subjected to mechanical vibrations, for the purpose of control of the oscillatory movement level or to predict the occurrence of resonance induced by the harmonic motion. Amongst the various equipment present in the industrial parks, compressors continue to be used due to their efficiency and versatility, particularly the reciprocating compressors, one of the oldest compressor models. In this context, the present work proposes to execute a project of a fixation support for the compressor, starting from an already existing reciprocating compressor base, in order to assure that the base does not have natural frequencies near the assembly's operating frequency, to avoid the phenomenon of resonance. The methodology used for the structure's development was a mathematical modelling and computer simulation using finite element method on a commercially available software and an analysis of the frequency response function between modal hammer and accelerometer in the compressor base with the purpose of evaluating and validating the efficiency of the applied methodology. Through the software, it was also possible to identify the natural frequencies and modes of vibration of various possible base models for the reciprocating compressor. Thus, it was possible to identify the project that fit best the operational needs. Finally, comparing the vibration levels to previous results to ensure the change lowered vibration levels during operating conditions.*

Keywords: *Modal Analysis, Structural Dynamics, Vibration.*

1. INTRODUCTION

The pair composed of motor and reciprocating compressor requires a base capable of supporting the action of static loads, time varying loads due to the vibration of the assembly's operation due to, for instance, an unbalance associated to a residual mass of one of the rotating pieces that generates a harmonic force on the system (Lee et al. 2004).

This work proposes to evaluate the use conditions of the already built compressor base through modal analysis using the finite element method of a commercial software, in which will be possible to evaluate the theoretical results of the system's natural frequencies. Afterwards, these results will be verified using an experimental approach to obtain the natural frequencies through a Frequency Response Function (FRF), by applying an impulsive force by means of an impact hammer and establishing the system's response with an accelerometer.

These results will determine if the structure is ill conditioned to withstand the dynamic loads this system is subjected to, in theory (project) and in practice (built physical model). The already built base, henceforth referred to as "old base", and the anchorage points of the compressor and motor are presented in Fig. 1.

The irregular project brings the necessity of a new structural project. Through modeling and selection between suggested base models with support to the motor and compressor, it is possible to identify their natural frequencies and select the most adequate structure for construction. Afterwards, it will be possible to propose improvements to the project by means of structural stiffeners, increasing the value of the natural frequencies, making sure the first natural frequencies are occur above the assembly's operating frequency, in order to avoid problems related to resonance (VanLaningham and Wood, 1979).

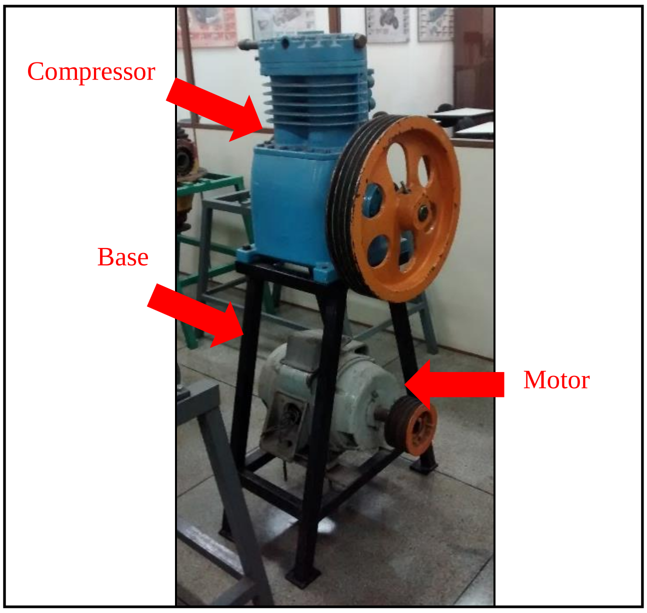


Figure 1. Motor, reciprocating compressor and old base assembly.

In this study, a Bitzer Open Block Reciprocating Compressor VIIs and a DELCO KI508E6 electric motor, with 15 HP, 1760 RPM (29.3 Hz), 220 V three-phase power-supply and starting current of 42 A were used. The pulley ration between the axes is 1.6:5, which results in an output rotational speed to the compressor of 352 RPM (9.38 Hz). To design and simulate the compressor base model, the commercial software Solidworks and its Simulation tool were used in order to identify natural frequency values and test modifications and reinforcements to the structure.

2. MODAL ANALYSIS

Among the several techniques used for the dynamic study of a structure, we highlight the modal analysis. The estimation of the dynamic characteristics is a requirement to evaluate the operating behavior of the structure as well as the safety indices.

The modal analysis allows for the study of the dynamic behavior of the model by means of a matrix formulation and the model parameters can be obtained through an analytical or experimental approach. The modeling of the system is based on three basic simplifications: the structure is linear, time-invariant and observable (Formenti, 1977; Snoeys, 1992). If these conditions are satisfied, the real structure can be represented by a discreet numerical model capable of describing the dynamic characteristics of the structure with a reduced data set. The formulation leads to a numerical model that allows establishing a relation between the structure's input and output from the modal parameters of the model.

The various aspects of the technique, as well as its applications, have been widely researched. Currently, there is a vast literature that covers the main theoretical as well as the practical aspects related to the modal analysis testing (Ewins et al. 1981; Allemang, 1982; Rost, et al. 1985; Leuridan et al. 1990; Snoeys et al. 1992; Ewins, 1984; Heylen et al. 1995; Maia et al. 1997; Allemang, 1999).

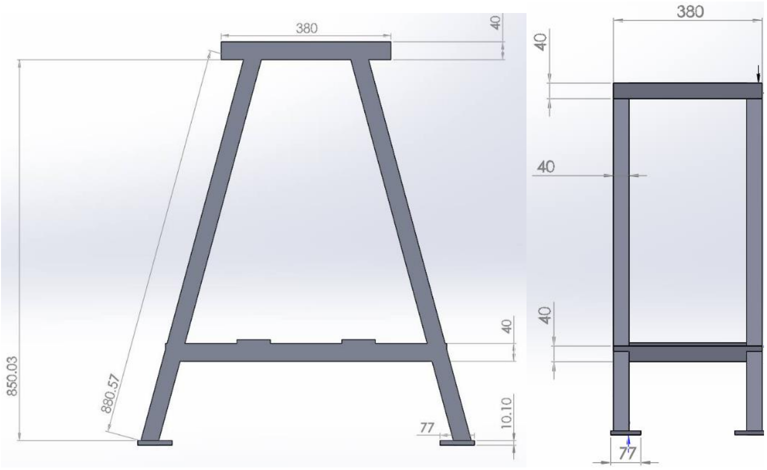
In the experimental modal analysis, the input-output relation is calculated from the excitation and responses captured, respectively, in the excitation points and the measurement points, previously selected. In this case, we obtain a set of complex functions, in which each function represents a transfer function Hij (s) between the excitation force applied on point j and the response measured on point i of the structure. The transfer functions can also be evaluated on the frequency domain, being called Frequency Response Functions (FRF(s)) then, or on the time domain, being called Impulse Response Functions (IRF(s)). The modal parameters of the system are estimated from the FRF(s) measured (Goyder, 1980; Lembregts, 1988), as well as from their time domain equivalents (Juang e Pappa, 1985).

Due to the necessity of knowing the input and the output of the system to obtain the respective FRF(s), these tests are usually conducted in laboratories, in well controlled environments with artificially generated excitations, using an impact hammer or an electromagnetic shaker. It is noted that, in this work, the formulation and mathematical foundations that compose the numerical and experimental techniques of the modal analysis were omitted, further information can be obtained in the works of Ewins, (1984) and Maia et al. (1997).

3. NUMERICAL AND EXPERIMENTAL PROCEDURE

3.1 Numeric model

With the objective of verifying the necessity of a new base, and the precision of the proposed methodology, the old base was numerically modeled using the commercial software Solidworks, Fig. 2 shows the dimensions of the model.



a) **b)**

Figure 2. Old base dimensions (values in mm): (a) Front view; (b) Side view.

For the numerical analysis, were considered the following:

- Base composed of 40mmx40mm x 2mm bars;
- AISI 1020 steel (Young's modulus 2x1011 N/m2, Poisson's ratio 0,29 and density 7900kg/m3);
- Estimated load from compressor: 250kg, and motor: 120kg;
- Base feet, as observed from the real model, are not perfectly aligned with the surface, therefore the feet indicated by 1 and 2 in Fig. 3 were considered with a free condition;
- Points 3 and 4 indicate the locations where it was possible to obtain, respectively, the input and output signals for model validation, using an impact hammer and an accelerometer, which will be commented on further.

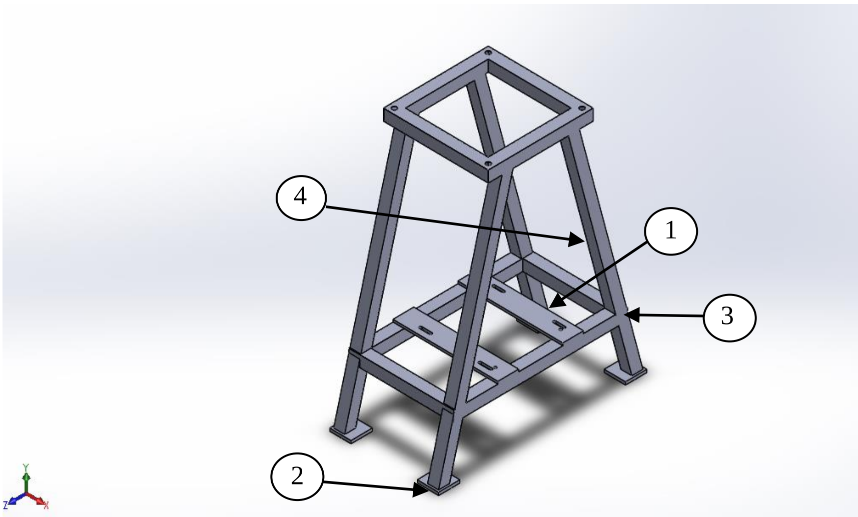
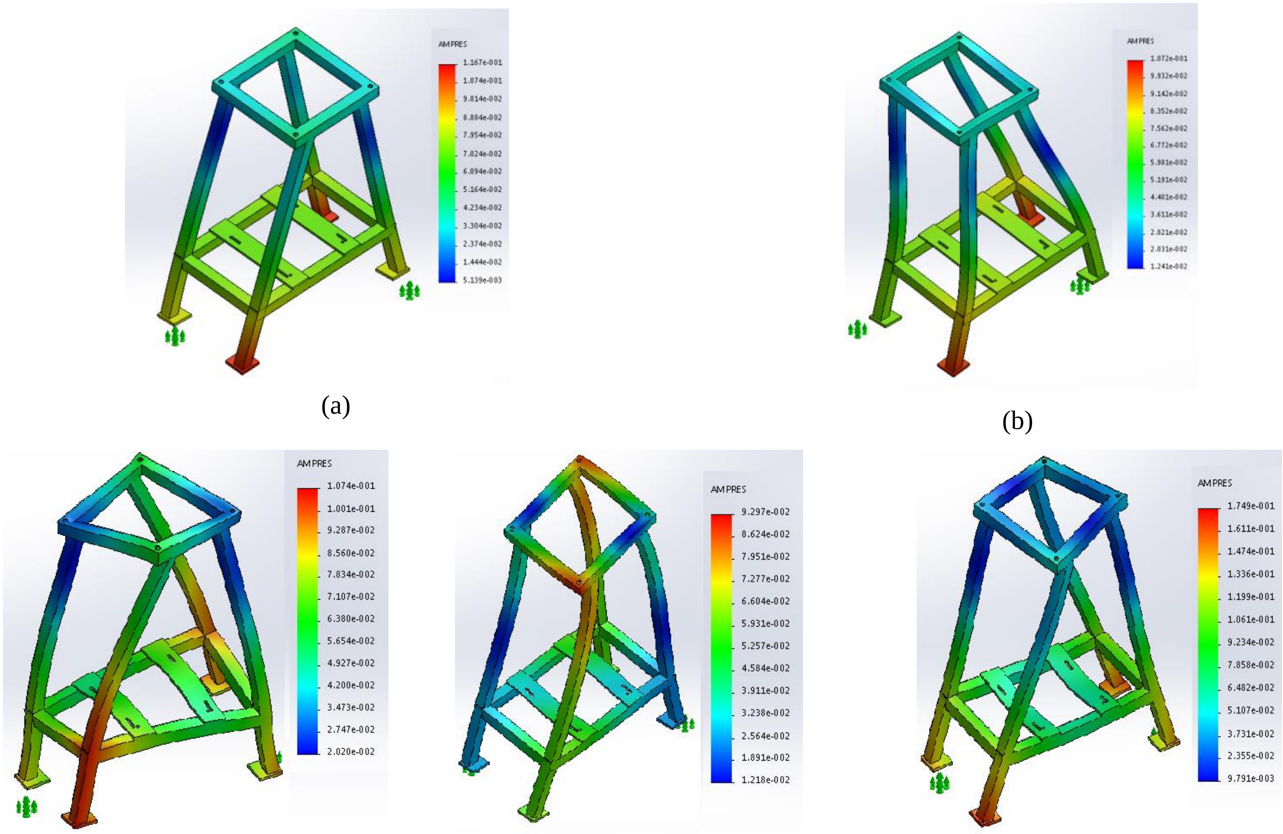


Figure 3. Computer model of the old base.

Under the described conditions, the natural frequencies were obtained using the *Simulation* tool of the Solidworks software, having used an equilateral triangle based mesh with sides of 10 mm, with 72094 elements and 142458 nodes. This procedure, besides being used to plan the experiment, was also used to obtain the modal results, natural frequencies and mode shapes, for the analysis of the five first modes and compare results with those obtained experimentally. Figures. 4 (a)-(e) show the obtained mode shapes for the obtained frequencies of 7.25 Hz for the first mode, 25.014 Hz for the second mode, 29.32 Hz for the third mode, 40.338 Hz for the fourth mode and 42.803 Hz for the fifth mode, respectively.



(a) (b)

(c) (d) (e)

Figure 4. Mode shapes from 1st to 5th mode respectively in (a), (b), (c), (d) and (e).

3.2 Experimental model

To obtain the real values of natural frequencies, the following equipment were used:

- PCB Piezotronics 086D20 Impact Hammer – 25 mV/N load cell;
- PCB Piezotronics accelerometer – Sensitivity 10,86 mV/g;
- Signal processing software SignalCalc *ACE*, Data Physics QUATTRO acquisition system with 4 inputs and 2 outputs.

During the experiment, the accelerometer was positioned on point 4, and the impulse excitation was applied with the impact hammer on point 3, as shown in Fig. 3. It was decided to perform a simplified test, in order to obtain only the first natural frequencies for posterior confrontation with numerically obtained results. It is noted that, in this present work, data related to mode shapes were not compared, as the measurements were made using only 2 points in the structure. Figure 5 shows the Frequency Response Function (FRF) obtained between points 3 and 4, as input and output, respectively.

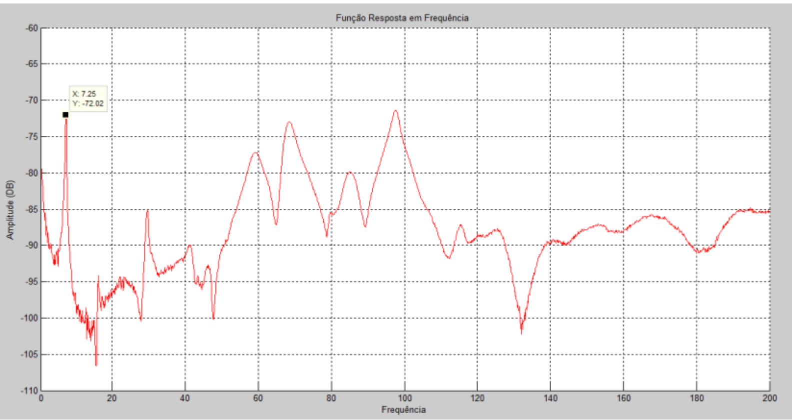


Figure 5. FRF between points 3 e 4.

In Figure 5, we note that the first natural frequency occurs at 7.44 Hz, as expected from the results obtained with the numerical model, in which the first frequency is of 7.25 Hz, with a difference of 0.19 Hz or 2.7%. We believe this deviation é due to the discretization of the frequency domain, as the results were obtained using a frequency range of 0 to 200 Hz with 1600 spectral lines.

It is important to note that the results obtained by comparing the numerical and experimental models were considered satisfactory, in a way that the numerical model presented itself as representative of the real model, only noting that mode shape data was not compared. However, we believe the first mode found in the numerical model is equivalent to the experimental results.

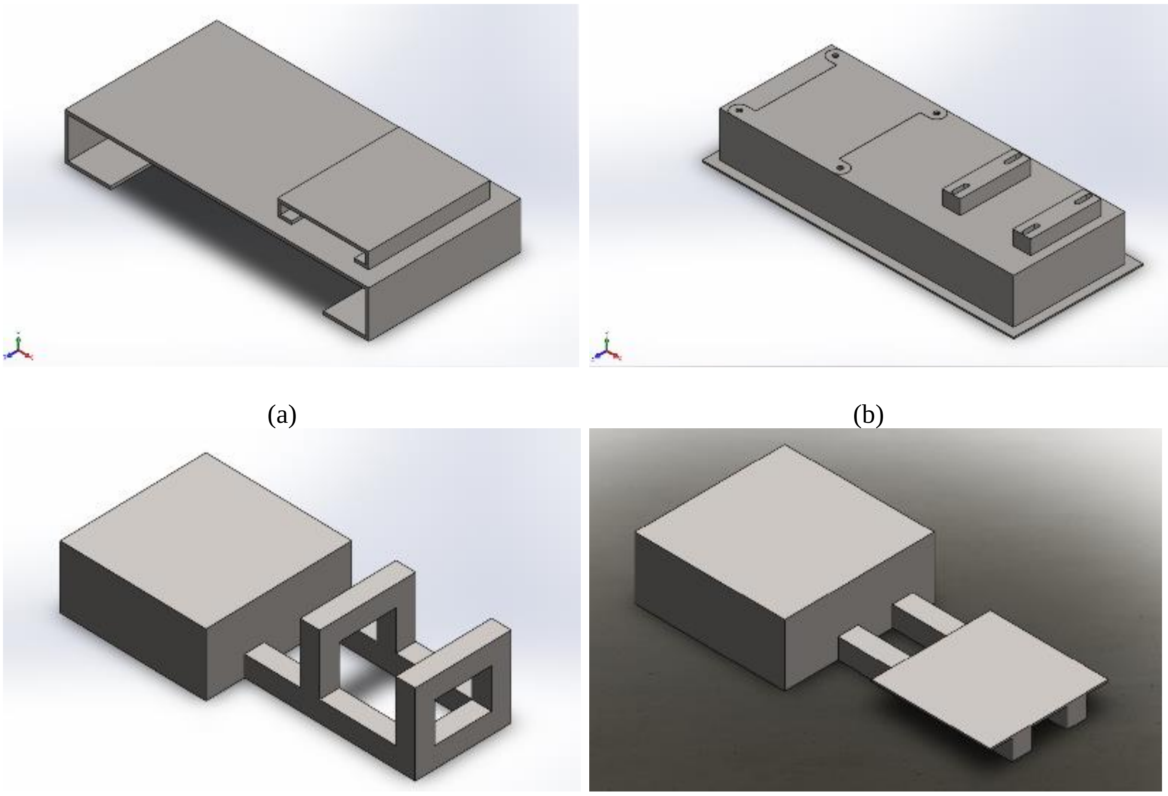
The next section will present proposed improvements to the compressor base, in order to assure that it's natural frequencies are above the motor operating frequency of 29.3 Hz.

4. PROPOSED BASE MODEL

In order to identify the most adequate model for construction, four numerical models were developed, as shows in Fig. 6, as suggested new bases. Afterwards, modal analyses were performed to compare natural frequencies and their respective mode shapes.

The simulations were performed using AISI 1020 steel, sliding surface condition on the bottom boundary, with the same loads from the compressor and motor stated previously. The meshes were configured with equilateral triangles with 10 mm sides, which resulted in:

- Model 1: 107447 elements e 191669 nodes;
- Model 2: 68027 elements e 132152 nodes;
- Model 3: 54969 elements e 95874 nodes;
- Model 4: 42457 elements e 78010 nodes.



(a) (b) (c) (d)

Figure 6. Numerical models suggested for construction: (a) Model 1, (b) Model 2, (c) Model 3 and (d) Model 4.

The selection criteria of the new base must consider that the frequencies excited by the motor's axis acceleration must not coincide with the base's natural frequencies until it achieves its steady state. This means that the selected base must have a first natural frequency above the operation frequency of the assembly. This is due to the unbalance caused by residual mass that creates a harmonic excitation with the motor's frequency.

Table 1 displays the obtained natural frequencies for the proposed models. Observing the results, only model 1 had the first natural frequency below the operation speed of the assembly. Considering Models 2, 3 and 4 have close values for the first natural frequency, Model 4 was chosen due to manufacturing constraints and available materials.

Table 1. Natural frequencies of the numerical models.

Mode	Model 1 (Hz)	Model 2 (Hz)	Model 3 (Hz)	Model 4 (Hz)
1	13,94	30,92	36,42	30,71
2	48,28	62,61	41,31	39,19
3	49,64	71,32	80,47	67,47
4	87,06	78,34	83,01	80,56
5	114,00	89,07	118,20	84,02

After selection, improvements to the project were suggested, in order to increase the stiffness and, by consequence, increase the natural frequencies of the base. Figure 7 shows reinforcements applied to the numerical model.

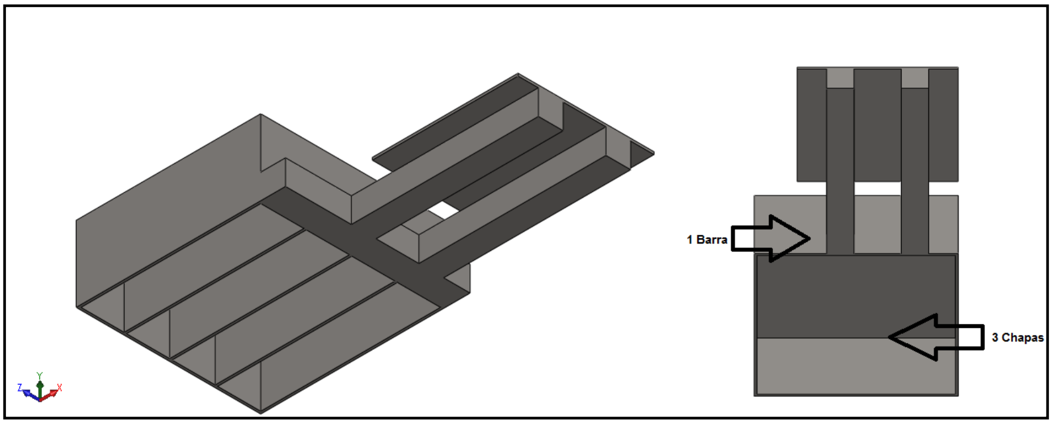


Figure 7. – Compressor base with reinforcements.

Three plates of 4 mm thickness and one bar to increase system stiffness compose these reinforcements. The dimensions and reinforcement positions are shown in Fig. 8.

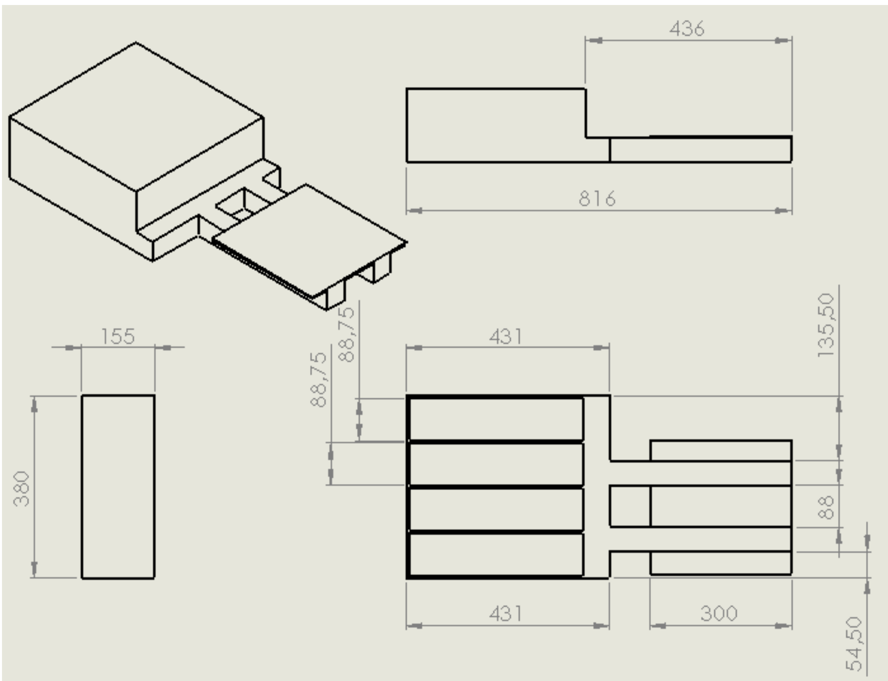


Figure 8. New base dimensions (mm).

5. RESULTS

The natural frequencies obtained in the simulation for each vibration mode are presented on Tab. 2. The results were obtained using an equilateral triangular mesh with 100 mm sides, witch 56905 elements and 103826 nodes.

Table 2. Natural frequency values for the reinforced new base.

Mode	Natural Frequency (Hz)
1	239,07
2	262,53
3	304,67
4	312,44
5	319,18

As expected, it is possible to notice a significant increase in structural stiffness. The first natural frequency was increased from 30.71 Hz to 239.07 Hz according to the obtained data. Therefore, it is possible to assert that this model can be adopted and that theoretically, when built, will not show problems related resonance. Figure 9 shows the structural behavior for the first mode of vibration of the new, reinforced, base.

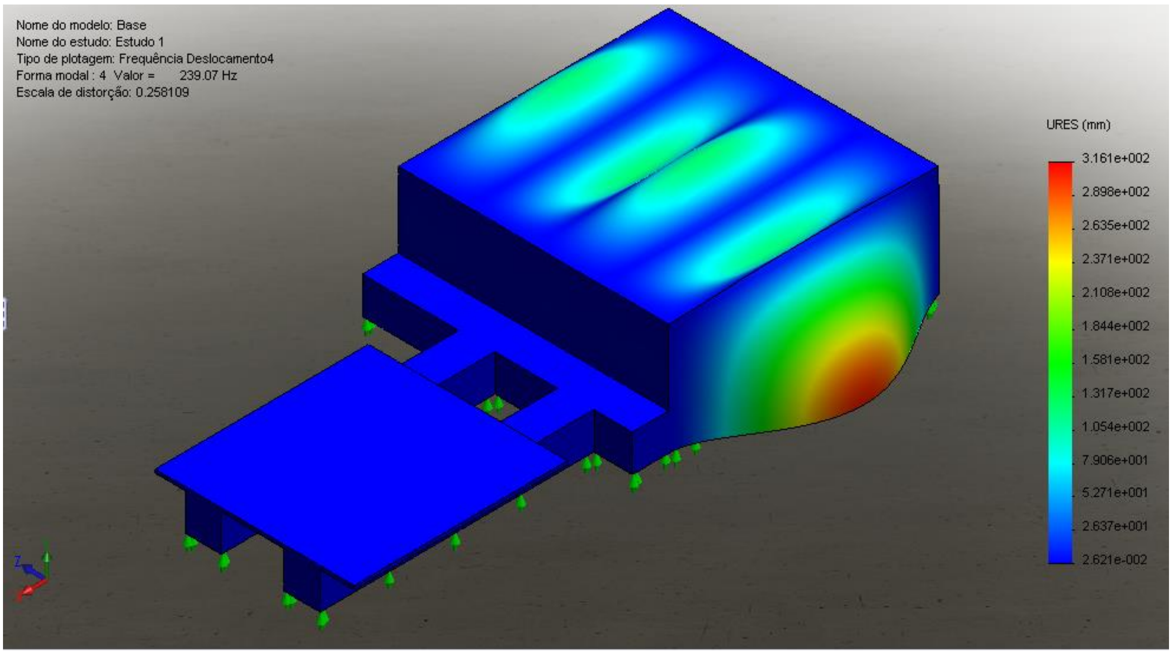


Figure 10. – 1st Mode shape of the new compressor base.

6. DISCUSSION

In this work, we verified through numerical modeling and experiment that the old base was poorly designed. The results show that during the motor's run-up (from start to operation speed) the old base went through three natural frequencies, two of those close to the operation frequency, further justifying the new base project. The difference found between the measurements of the theoretical and practical results were relatively small, with a 2.7% deviation.

It is noteworthy that viable project enhancements were proposed, in a way that allowed the system's first natural frequency to be significantly higher than the frequencies excited by normal conditions of operation of the assembly.

7. ACKNOWLEDGMENTS

We thank the Postgraduate program in Mechanical Engineering (PPGEM) of the Federal University of Technology – Paraná, Coordination for the Improvement of Higher Education Personnel (CAPES) for the financial support, Professor Adailton Silva Borges for the guidance and support that allowed the fulfilment of this work, and finally, the Technological Laboratory of Mechanical Vibrations and Maintenance team involved in this work.

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