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TEST BENCH FOR EXPERIMENTAL STUDY OF THE GEOMETRIC STIFFNESS IN VIBRATION OF BEAMS AS SUPPORT OF ROTATING MACHINES

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Abstract. *In this research paper, we intend to describe a test of bench in order to study the effects of geometric nonlinearities on vibrations of rotating machinery support structures. Dynamic characteristics of structures depend on their stiffness and mass. With these, we determine their natural frequencies and modes of free vibrations. Nevertheless, the initial stiffness of a structure, computed in its unloaded state, are affected by the applied forces, the so-called geometric stiffness. Compressive forces usually reduce the stiffness and the frequencies and may lead to buckling, for zero frequencies. Currently, structural engineering is striving to produce more slender structures, with less material consumption, which has led to the construction economics and with flexible parts.*

Keywords: *Engine Base, Beam Resonance, Geometric Stiffness*

1. INTRODUCTION

It is known that the dynamic characteristics of a structure depend basically on its stiffness and mass. With these two parameters, the natural frequencies and modes of vibration of the system are determined. However, the initial stiffness of a structure, determined in the unloaded situation, are affected by the presence of the loads. This is the case for compression forces, which tend to decrease the stiffness and vibration frequencies, which may, in the limit, lead to loss of system stability, as in the case of columns, or to resonate, as occurring with vibration problems. Works that clarify the concept of stiffness and evaluate the geometric stiffness in the vibration of mechanical systems can be found in Pauletti (2013), Brasil and Wahrhaftig (2014, 2016), Wahrhaftig (2008), and Wahrhaftig, Brasil, and Balthazar (2013).

One class of mechanical systems of importance to the national industry is the machine bases, subject to vibrations induced by the supported equipment. These vibrations can affect the safety of the structure itself, but, in the general case, can have harmful effects on the equipment itself and the quality of the manufactured product. They can also make inappropriate the environment work for the operators. Problems related to vibration of rotating machines in some field of engineering can be found in Adams (2000), Brandão (2013), Assunção (2009), Silva *et al*. (2016), Bachmann and Ammann (1987), Kapsalas *et al.* (2016), Satpal *et al.* (2013) and Sotiropoulos and Tsihrintzis (2016).

In order to evaluate experimentally the effect of geometric stiffness on the dynamic behavior of structures excited by periodic loads, this work describes with design details a bench specifically made for this purpose in laboratory tests.

2. THE CONSIDERED HYPOTHESIS AND MATHEMATICAL BASIS OF THE PROBLEM

The effects of geometric stiffness are studied experimentally, using a physical model in laboratory test with fabricated equipment, sensors and systems acquired; having as reference a theoretical-numerical approach with a mathematical model by the Rayleigh method, designed to represent a simple supported beam AB, intended to function as the base of an engine (Figure 1(a)). In that Figure, P is the prestressing force; Eg and f_E designate the engine and its rotation frequency; EI, L, and $f_n(P)$ are the flexural stiffness, the theoretical length of the beam, and the first natural frequency that depends on P, respectively. The defined cross-section for the beam can be seen in Figure 1(b), where t is the thickness of the wall and h is the external side dimension of the square section adopted. The adopted dimensions of the beam were: h = 102 mm, t = 2.5 mm, and L = 3 m.

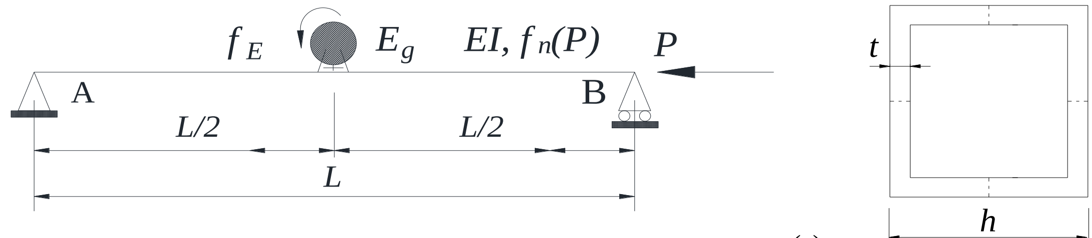


Figure 1. Parameters of the problem. (a) beam model; (b) cross section adopted

The presence of the prestressing force P reduces the beam stiffness and, consequently, its natural frequencies $f_n(P)$, being able to lead to operating regimes not foreseen initially. It is assumed that the original design has taken care to move away the natural frequencies of the system from those of the excitation. However, the presence of the prestressing load, which, by hypothesis, decreases the stiffness of the beam and, consequently, its natural frequencies, can lead to unexpected, potentially dangerous resonance regimes, as seen in Figure 2 by the intersection of presented curves, the black (beam) and the blue (rotating machine), considering the frequency of the engine as 13.33 Hz (800 rpm). The natural frequency of the beam has been calculated by using Eq. (1), where m_v is a distributed mass per length unity (cross section area multiplied by the density of the material, 2700 kg/m^3), and m_c is the lumped mass at the central position of the span (engine mass, 4.7 kg). The mathematical development of that equation can be found in Brasil and Wahrhaftig (2017), and Wahrhaftig, Nascimento and Brasil (2017).

$$f_n(P) = \frac{1}{2}\left[\frac{\pi^2 EI - PL^2}{L^3\left(L m_V + 2m_c\right)}\right]^{\frac{1}{2}}. \tag{1}$$

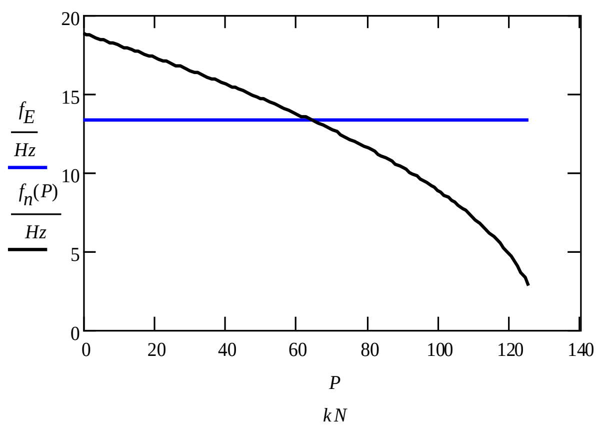


Figure 2. Resonant regimen structure-excitation

3. DESIGN PROCESS AND DETAILS OF THE TEST BENCH

The concept of design stands for a variety of meanings, however, the one that better describes this study is the one of planning, the portrayal of ideas and the process of formulation of something that considers and applies technical knowledge and scientific concepts to define a system or an object to make possible its construction. It's important to highlight that planning begins out of a requirement or a problem, which demands a solution. Therefore, the design process is a series of steps that must be followed to come up with a solution to a problem.

Analyzing the object of study in this case, the bench of tests was initially designed on sketches which made possible the definition of its pieces through laboratory tests. The difficulty inherent in this type of creation is to establish a possible relationship between the theory to be studied and the physics of the problem, making possible the choice or definition of subsystems that mechanically behave as desired and, at the same time, would be commercially usual and relatively easy to be found. Hours of analysis were necessary in order to adjust the dimensions of the pieces, the mechanisms involved with working frequencies, weights, correct data acquisition, forms of application of the horizontal force, availability of electrical energy, connections, connectors, transportation and security, making the project a system of relative complexity that should be harmonized to the physical elements and the mathematics of the study. Thinking about that, the schemes were initially elaborated by means of simple traces, all of them freehand made, where the solution of the problem was graphically elaborated and where it was possible to visualize primarily the object to be constructed. With the problem and the solution in mind, the ideas have been thrown on paper and the first lines gave a shape to the creation, as shown in Figure 3.

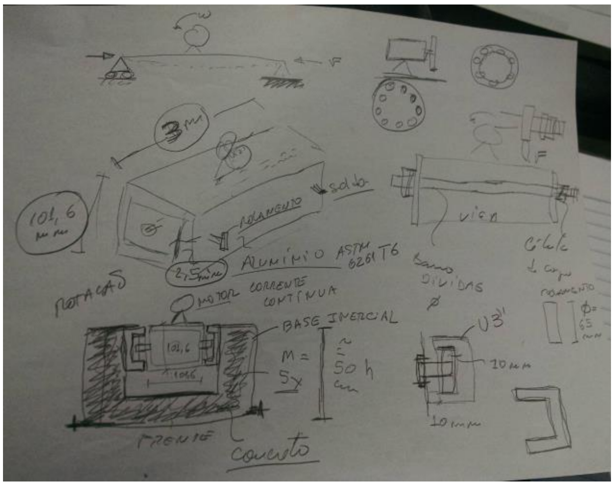
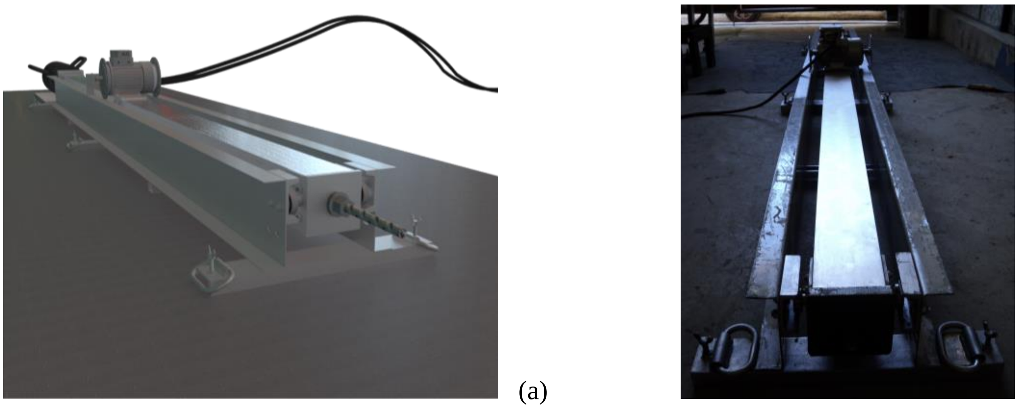


Figure 3. First lines of conception of the test bench

The executive drawings of the bench were consolidated using CAD (Computer-aided design) tools and the 3D modeling tools that enabled the registration of the projects building specification, what facilitates its understanding and replication.

To meet the needs of the present study, a test bench was designed consisting of an aluminum metal beam and a prestressing bar, supporting a rotary machine (Figure 4). The bench design was started with the outline of preliminary tracings that were consolidated in the executive drawings using CAD (Computer-aided design) tools, as seen in Figure 4(a). Aluminum was chosen because of it having a lower yield stress compared to the steel, so, a smaller force is required to produce the phenomenon to be observed. Its lower density also favored the conditions for transportation of the prototype. In addition, aluminum is one of the main structural materials used in aviation and naval engineering, potential areas of application for the present study, together with other sectors of the mobility industry. In designing the system, the frequency ranges of interest and the horizontal forces to be applied were analyzed. These studies defined the specifications of the components that were introduced in the experiment, which was conducted in the fabrication of the test bench presented in Figure 4(b), whose graphical presentation, with a brief description of the principal constituent elements, is the central objective of the present work. The model was then calibrated for the resonance that occurred at the natural frequency of the first mode of vibration of the structural system, for the horizontal force under consideration.



(a) (b)

Figure 4. Test bench: (a) project and (b) prototype

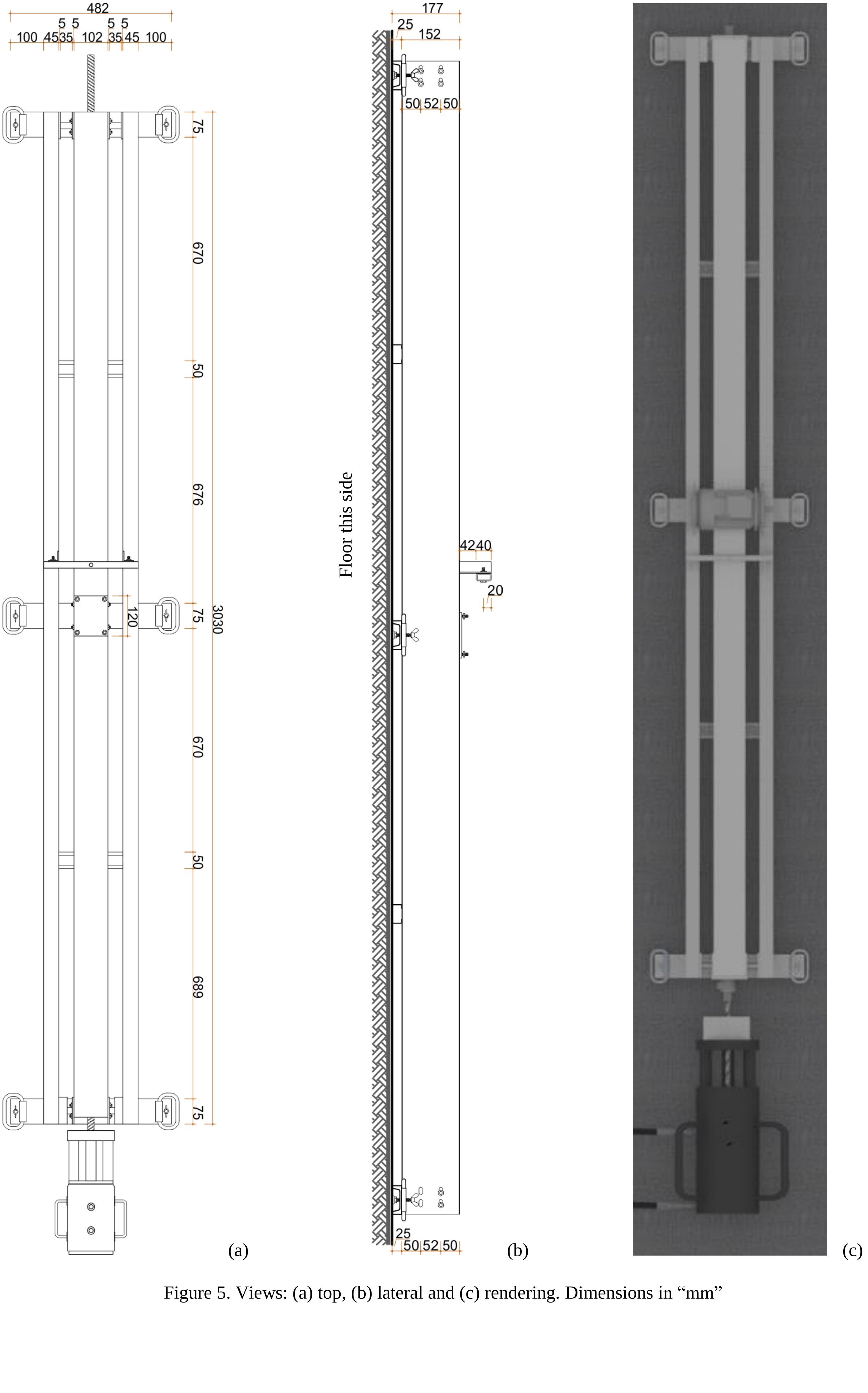
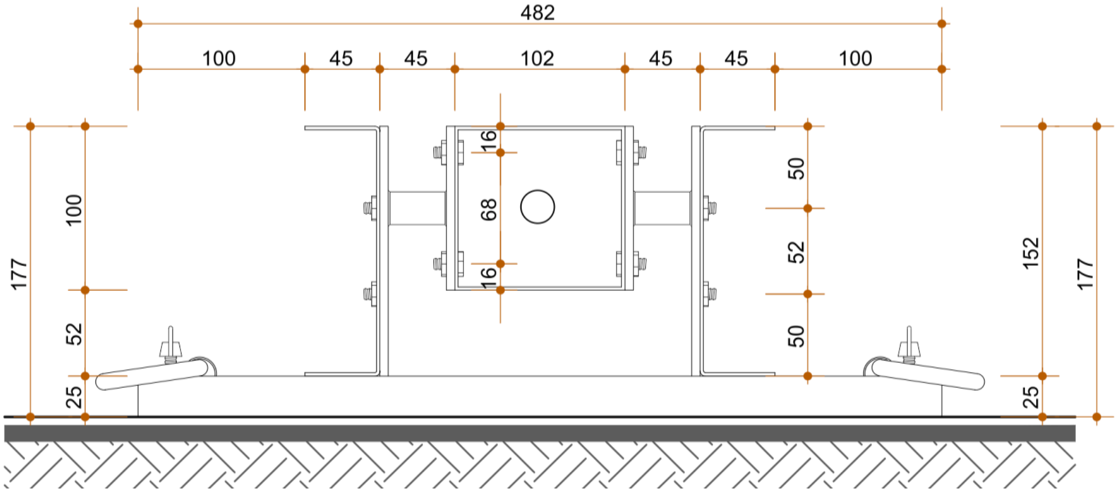
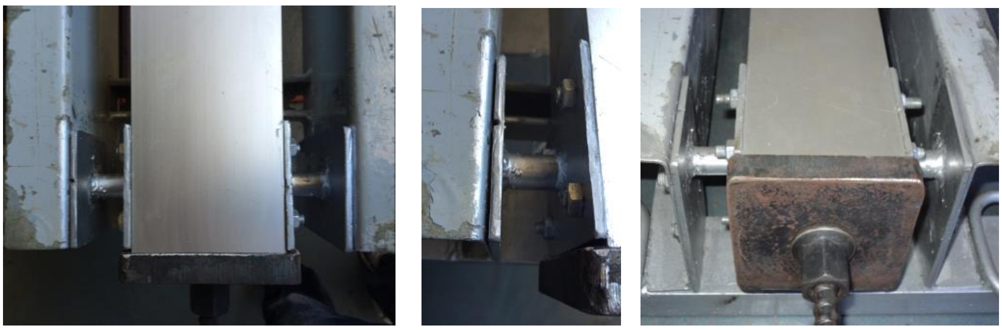


Figure 5. Views: (a) top, (b) lateral and (c) rendering. Dimensions in “mm”

Laterally, on both sides, "U" cold formed metal profiles served as support for the central aluminum beam. Five transversal bracings were arranged along the profiles, stiffening the system. Three of these stiffeners had handles for transporters and height variators at their extremities. The height variators were threadable screws of type "butterfly", with a small circular plate in the other extremity, that enabled manual adjustment of the verticality of the system. The assembly, thus formed was also designed to provide the desired inertial safety to the tests and it has been calculated to be at least five times the mass of the excitation. A plane frame was placed crossing the test bench upper face and welded on both sides over the “U” profiles, seeking to guarantee the safety of the researchers in the case of rupture of the main piece, that ends the design. A top and lateral view graphically presenting the previous description can be seen in Figure 5. Figure 6 and Figure 7 shows the details of the design of the pinned and roller supports. The first consists of a circular metallic axis welded to a square plate fixed by screws to the lateral metal support, on both sides. The second consists of a scheme similar to the first one, but with a bearing added to the end of the axis at the point of contact with the support, being inserted into a U-shaped profile segment facing vertically inwards. In this, a limiter that eliminated the clearance between the bearing and the upper part of the U-shaped bearing had been added.

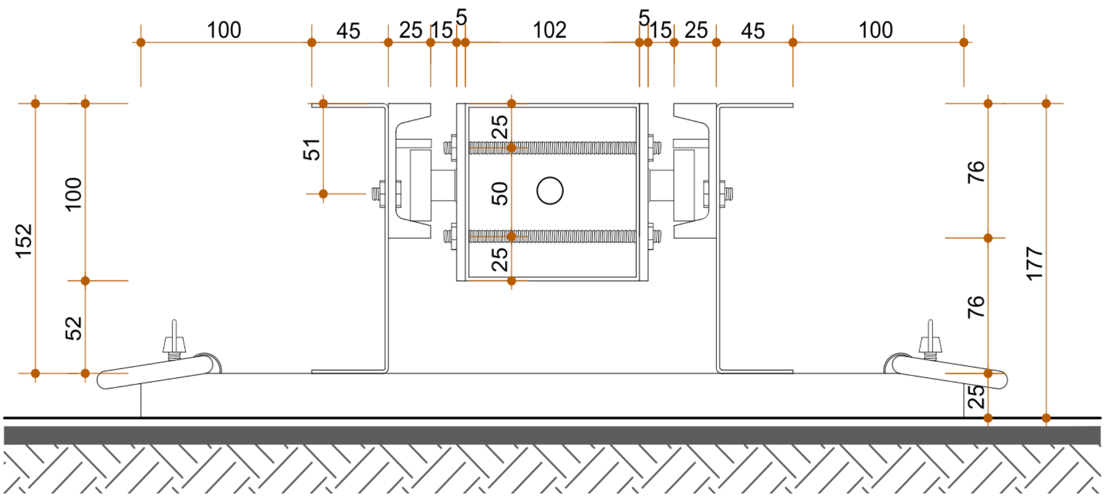


(a)

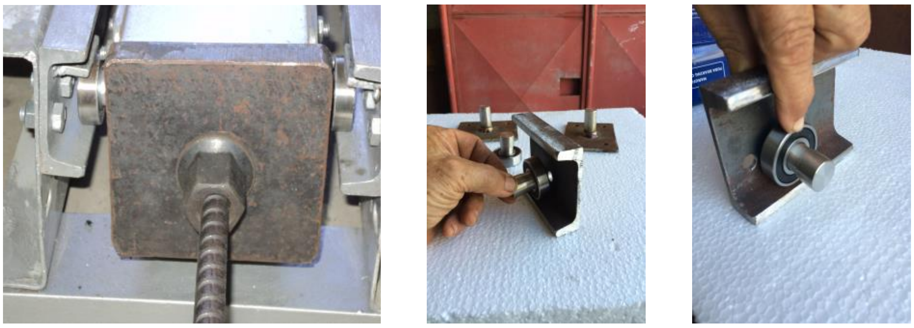


(b)

Figure 6. Pinned support: (a) (Detail 1) - Dimensions in “mm”, (b) Photos of the prototype



(a)



(b)

Figure 7. Roller support: (a) (Detail 2) - Dimensions in “mm”, (b) Photos of the prototype

In the computer-aided design environment, three-dimensional visualizations are valuable features of the designer's service, allowing the visualization of detail and offering a sense of realism that makes it possible to correct even the subtlest elements. To conclude the description of the test bench design, images of Figure 8 present three-dimensional views of the prototype that was manufactured and its service condition. In Figure 9, it is possible to see the mechanism during construction and on the test day.

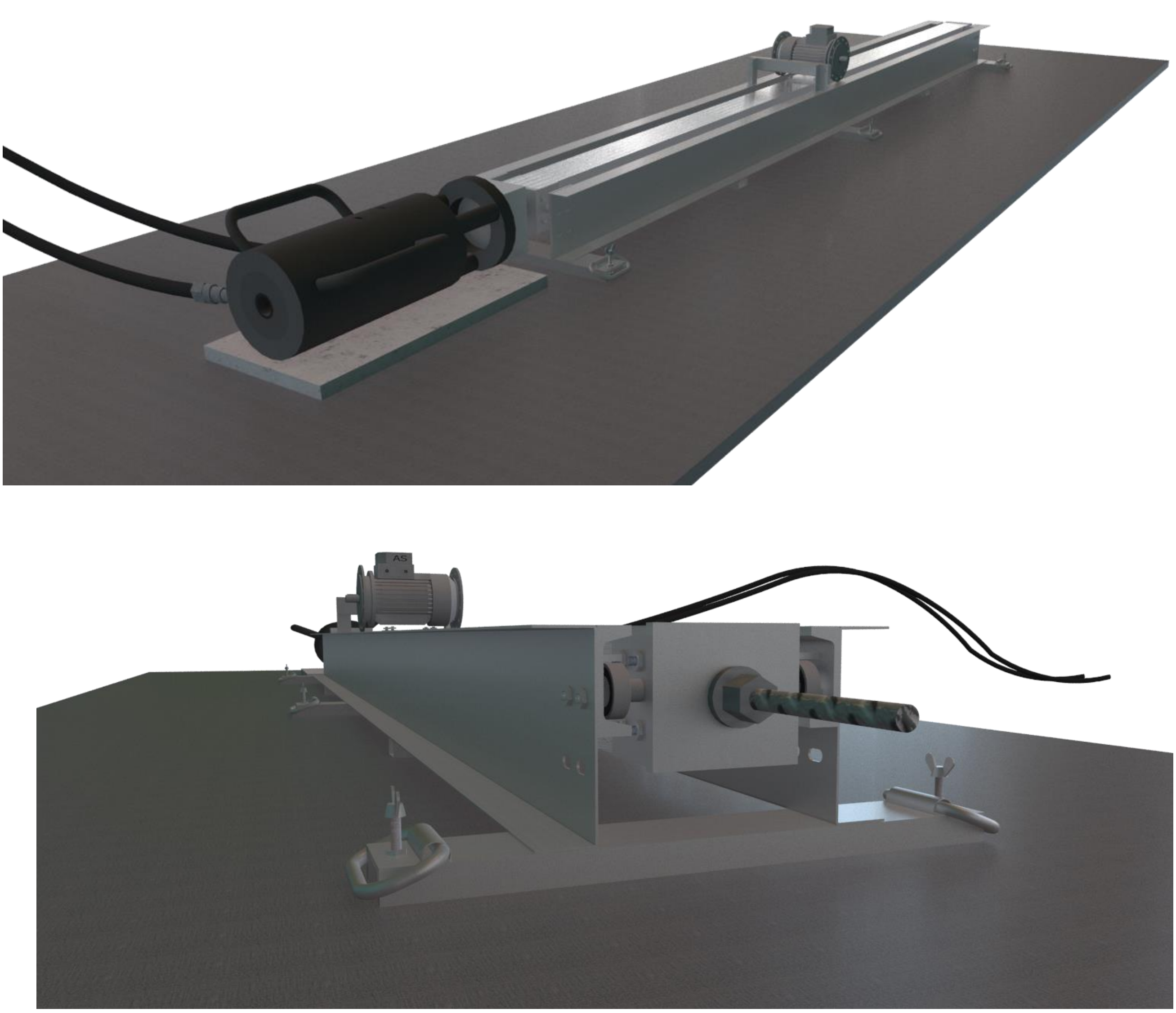


Figure 8. 3D images of test bench designed

(a) (b)

Figure 9. Photos of test bench: (a) in a construction process, (b) on the test day

4. EXPERIMENTAL RESULTS AND A BRIEF DISCUSSION

To attest to the usage of the projected system, the results of the performed tests are presented below. A plan of loading was prepared based on the predicting resonant frequencies for each level of force, as detailed in Figure 10, where P_{lim} represents the load of buckling of the beam. Once the established force for each stage had been applied, a frequency inverter was used to vary the frequency from zero up to overpass the waited frequency for each force, according to Table 1, and returning to zero again. It is important to mention that the slenderness ratio of the beam is 74.08.

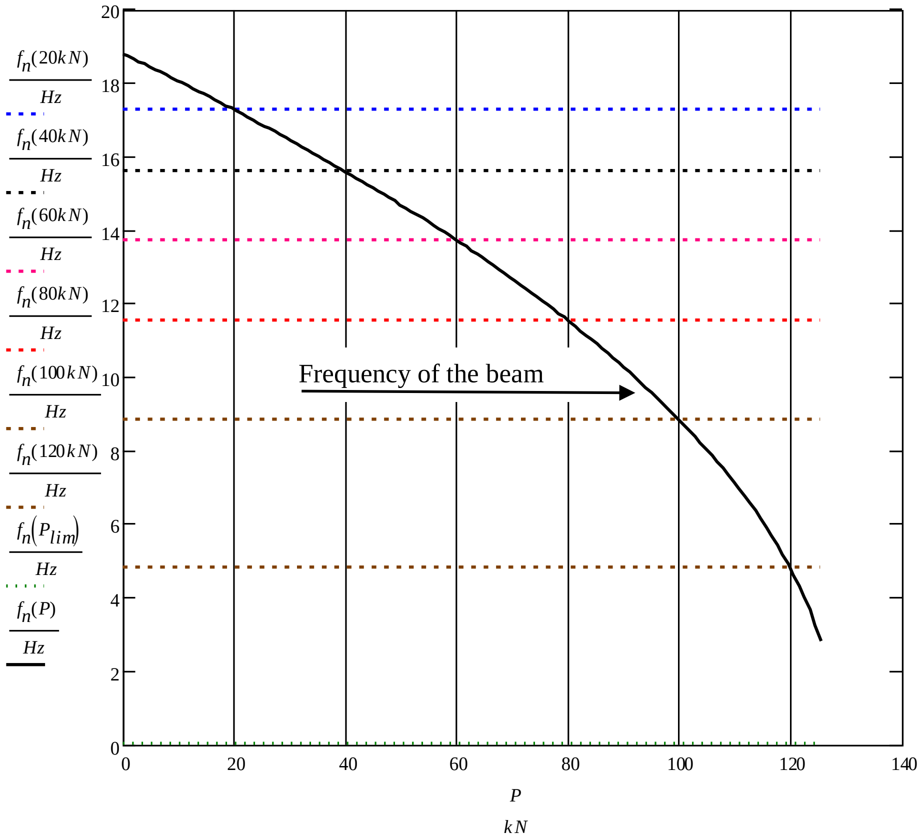


Figure 10. Plan of loading

Table 1. Waited results

Force (kN)	Frequency (Hz)
0	18.7980
20	17.2719
40	15.5972
60	13.7195
80	11.5403
100	8.8393
120	4.805
125.93 (Yielding)	2.599
128.39 (Buckling)	0

The results of tests are summarized in a temporal series, Figure 11, and in Fast Fourier Transform (FFT), Figure 12, in which could be seen the system resonant frequencies for different levels of prestressing force. To compress the beam a 15 mm diameter Dywidag bar of prestressing coupled to a hydraulic jack was used. The applied prestressing force was varied from 0 up to 80 kN (~9 tons) with an interval of 20 kN (~2 tons), with signals present in that order on the graph of Figure 11. The tests were interrupted when the horizontal force reached approximately 84.52 kN (~9.5 tons), which led to the collapse of the central beam by flexion (Buckling), with the test trying to reach 100 kN of the load.

The temporal responses of the structure were obtained by a bi-directional accelerometer with a measuring range of ±50 *g* (Lynx, 2014) (*g* is acceleration of gravity). The force was controlled by a load cell with 200 kN of capacity. Both sensors were connected to a digital data acquisition system and to an electronic portable microcomputer. The hydraulic jack, with 500 kN of capacity, was kindly provided by “*Protende Sistemas e Métodos de Construções Ltda*”.

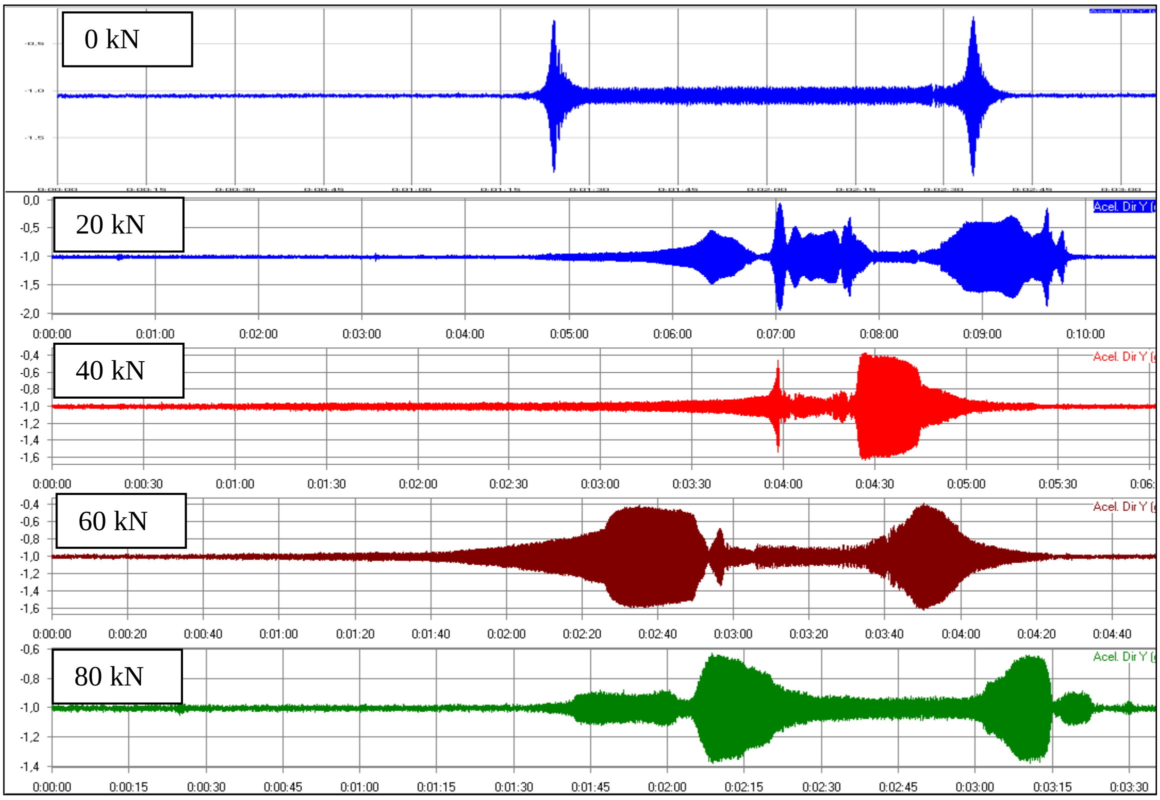


Figure 11. Temporal series

The resonant frequencies were obtained by employing Fast Fourier Transform (FFT), calculated with the AqDAnalysis Program (Lynx, 2014), by the selection of an interval of time considered appropriate and by using a "Rectangular" windows of analysis.

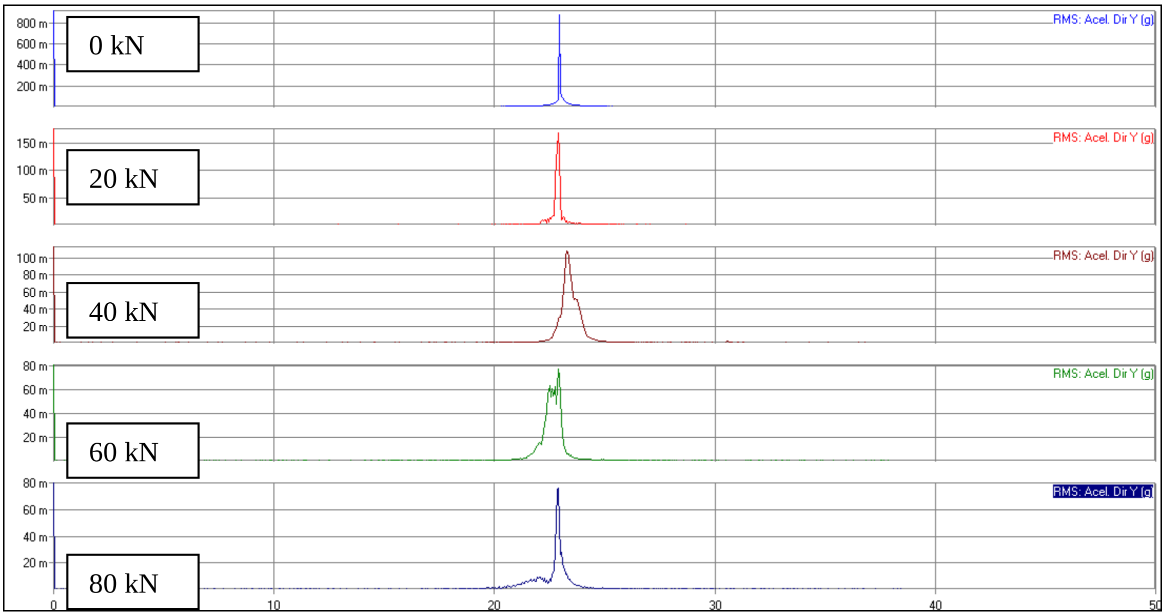


Figure 12. FFT for all temporal series

5. CONCLUSION

It was possible to observe there were no changing on results from *FFT* and the frequency is not altered even with the application of a compressive force, which was capable of destroying the central element of the research. In this sense, it is necessary to investigate the influence of the prestressing bar to the mathematical model or looking for a new arrangement for the test by removing the physical components that could be inferred in the results.

Furthermore, one must be aware of the sensitivity to imperfections of a slender system such as that studied and the difficulty of designing it, considering the physical elements available and the forces involved in a natural scale model.

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