ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

DYNAMIC RESPONSE ANALYSIS OF AN ABSORBER SUBJECTED TO VIBRATION

Francisco Galdino Junior
Filipe Nascimento Silva
Bruno Moura Miranda
Diego David Silva Diniz
Antonio Almeida Silva
Department of Mechanical Engineering (UAEM)
Federal University of Campina Grande (UFCG)
Av. Aprígio Veloso, 882 – Bairro Universitário, CEP: 58429-900, Campina Grande – PB, Brazil
fgaldinojunior@gmail.com, nascimento.filipe@hotmail.com, bmouramiranda@gmail.com, diego_leader@yahoo.com.br, almeida@dem.ufcg.edu.br

Abstract. *Vibrations in machines are undesirable because sometimes it causes unnecessary movements. Such movements affects the products being processed, generating defects in the components, reduction in the machine performance reducing the lifetime of the parts, increase power consumption and produce noises that could affect the health of exposed persons. A machine or system may experience excessive vibration if the excitant force frequency coincide with the natural frequency of the machine or system, causing resonance. In these cases, the vibration of the machine or system can be reduced using a dynamic vibration absorber that is a secondary system associated with a main system, in order to eliminate resonance, to let natural frequencies distant from the excitation frequency. The main focus of this work was the study of dynamic absorbers by an analytical analysis and a posterior comparison with the results obtained in a built experimental prototype and computational simulation held by CAD software. Based on these results, it was concluded that the study was satisfactory, since the error between the used methods was smaller than 3%.*

Keywords: *absorber dynamic vibrations, resonance, analytical theoretical analysis, prototyping, computation simulation*

1. INTRODUCTION

Machinery, industrial equipment, Civil construction structures are often subjected to undesirable mechanical vibrations that could affect the operation and integrity of the components, or decrease operator's comfort. Thus, the vibration is becoming the object of study for research centers all over the world. In order to control the vibration levels using techniques which have the sole purpose of satisfying a comfort condition and operation. Among the techniques employed for this purpose Dynamic Vibration Absorbers (ADV's) are a very good and useful technique, which has become a strategy efficiently and economically.

As RAO (2009), a machine or system can experience excessive vibration if it experiences the action of a force whose excitement almost coincides with a natural frequency of the machine or system. In such cases, a solution found by the engineering was the ADV's utilization which is a simple mass-spring system. Its development is dated from the beginning of the last century and continues having wide applications in mechanical and civil engineering and aerospace. (Borges, 2008).

The operation of the ADV's is based on the principle of anti-resonance, wherein the main vibrating system under a given excitation is connected to the ADV's, a secondary system (dynamic absorber) with the purpose of producing a force of the same magnitude, direction and frequency in opposite phase to the excitement.

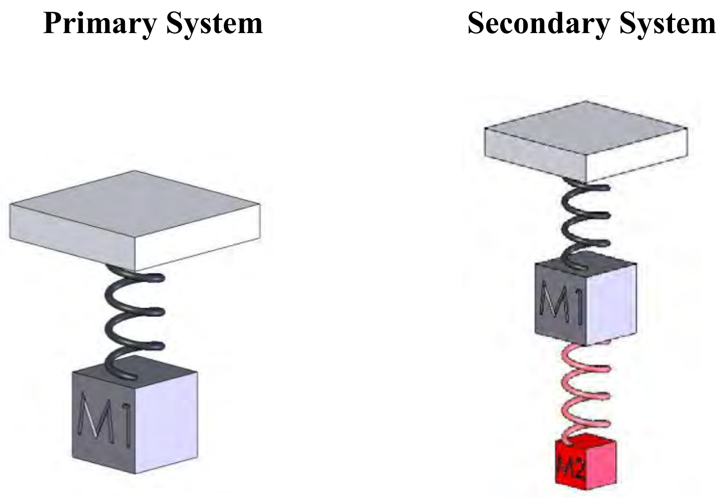
Recently, due to the globalization of information has become feasible to apply control techniques for use in the practice of ADV's. Research in this area has been motivated due to the growing demand for low noise and precision industrial equipment, particularly with regard to machine tools and nanotechnology. By being active in various types of equipment is found in the pipes a key point to reduce overall levels of vibration in these machines object of study of this work. (Fernandes et. Al., 2010).

In this context, the paper presents the development and design of a passive dynamic absorber, connected to the main system that is excited by a motor with unbalanced mass. Thus, to assess whether the vibration levels decreased in the natural frequency of the system, because of the effect of the dynamic absorber were performed experimental and numerical simulation, where it is expected check if the design of the dynamic absorber used is consistent with what the literature reports.

2. THEORICAL FUNDAMENTATION

2.1 Dynamic absorber

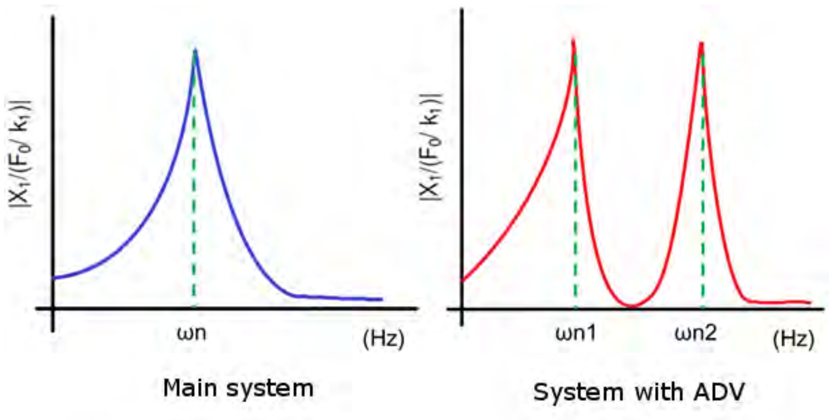
The ADV is a mass, stiffness and damping parameters concentrated device. once coupled to the primary structure whose vibrations desire to mitigate it is able to absorb vibratory energy at the connection point. (Marques 2000). The ADV device is a lumped mass, stiffness and damping, once coupled to the primary structure whose vibrations desire to mitigate is able to absorb vibratory energy at the point of connection. (Marques 2000). Thus, Fig. 1a presents the main system, whose the study is carried around it and Fig. 1b shows one of the ways to connect the main system with the dynamic absorber system.



(a) (b)

Figure 1. Schematic of a dynamic vibration absorber.

The Figure 2 shows graphs function typical frequency response of the two systems. For the system shown in Fig. 1a, which has a degree of freedom, it can be seen in the second graph there is only one natural frequency. In curve (b), which represents the response of the same old system more dynamic vibration absorber (Fig. 1b), we can see that the system has two degrees of freedom, and there are two natural frequencies. The introduction of ADV allows the generation of a anti-resonance which existed previously the resonance.



(a) (b)

Figure 2. Graphs of frequency response of a system with and without ADV.

2.2 Types of absorber

There are three basic types of dynamic vibration absorbers: passive, semi-active and active. They differentiate by form that generate such force. All are based on the same principle: to introduce a secondary structure in a system primary subject to oscillatory action of a force, so that the secondary structure is the same as natural frequency of the exciting frequency, generating the anti-resonance principle. (Fernandes et al, 2010)

The dynamic absorbers present a passive vibration control that deal directly with the physical properties of the machine, such as stiffness, mass and damping. The control passive vibration should act with a basic structural change,

ISSN 2176-5480

the use of other materials, or adding a passive element which is an element whose function depends on an external energy source. (Aquino, 2011)

The Figure 3 shows a typical example of the dynamic vibration absorber passive structure which consists of a cantilever beam with a motor by exciting the unbalanced mass, providing oscillatory movements in the structure. When the beam reaches its natural frequency, the ADV will cause oscillatory movements in opposite phase to the movement of the beam and, consequently, there arises a force that attenuates the oscillation of the system.

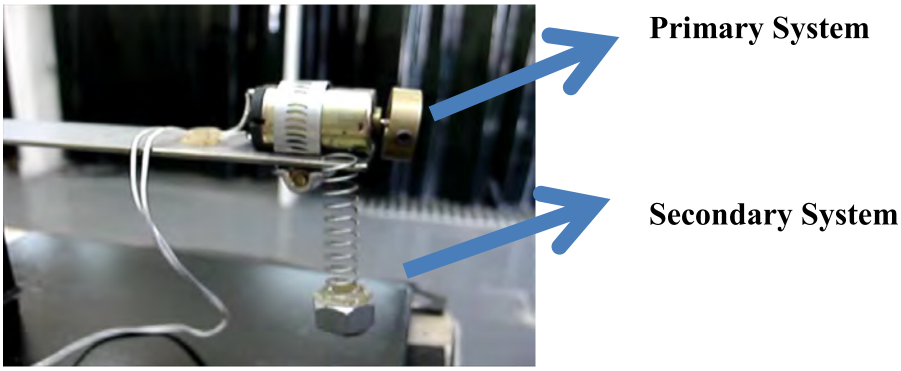


Figure 3. Example dynamic vibration absorber.

The ADVS semi-active or are adaptive system that can adjust physical parameters of mass, stiffness and damping, enabling the capability of tuning devices in a wider range of frequencies. In this context, the recent technological advances in the production of so-called smart materials (piezoelectric materials, shape memory materials, electro-rheological fluids and magneto-rheological) offer ample opportunities for proposing new configurations ADV's adaptive. (Marques 2000).

The active ADV's devices are able to control the vibration of the main system with the use of an external power source. Actuators cause an electromechanical force counteracts the motion of the system and thereby attenuating the vibration for example.

2.3 Mathematical model

The absorbers are capable of promoting the reduction of its vibrations efficiently and consequently reduce noise levels in many cases with the advantage of not requiring high costs for its implementation ("Ohana et al. Al. 2011 "). At Fig. 4 is shown the operating principle of a dynamic absorber for two degrees of freedom composed by springs having linear characteristics whithout damping (Borges, 2008).

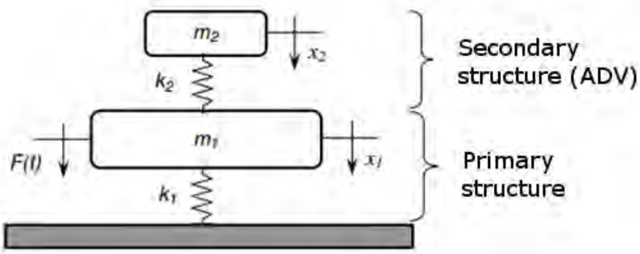


Figure 4. Vibratory system of two degrees of freedom.

The basic equations that describe the motion of the system (main system + ADV) are described in matrix form as follows:

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} F_0 \\ 0 \end{bmatrix} \mathrm{sen}(\omega t) \tag{1}$$

$$x_1(t) = A_1 \mathrm{sen}(\omega t) \tag{2}$$

$$x_2(t) = A_2 \mathrm{sen}(\omega t) \tag{3}$$

Where m_1 and m_2 are the masses of the primary system and ADV, respectively. k_1 and k_2 are the springs, located as shown in Fig. 4. F_0 is the excitation force which is generated by unbalanced mass and finally ω is the frequency of excitation.

Performing some mathematical arrangements it is possible to determine A_1 and A_2 amplitudes, and the two natural frequencies of the system ω_{n1} and ω_{n2}.

$$A_1 = \frac{\left(1-\frac{\omega^2}{\omega_{n2}^2}\right)F_0/k_1}{\left(1-\frac{\omega^2}{\omega_{n2}^2}\right)\left(1+\frac{k_2}{k_1}-\frac{\omega^2}{\omega_{n1}^2}\right)-\frac{k_2}{k_1}} \tag{4}$$

$$A_2 = \frac{F_0/k_1}{\left(1-\frac{\omega^2}{\omega_{n2}^2}\right)\left(1+\frac{k_2}{k_1}-\frac{\omega^2}{\omega_{n1}^2}\right)-\frac{k_2}{k_1}} \tag{5}$$

$$\omega_{n1} = \sqrt{\frac{k_1}{m_1}} \tag{6}$$

$$\omega_{n2} = \sqrt{\frac{k_2}{m_2}} \tag{7}$$

For A_1 is equal to zero, in other words amplitude of the main system equal to zero the above equation must conform to the following hypothesis:

$$A_1 = 0 \rightarrow \omega^2 = \frac{k_2}{m_2} \tag{8}$$

The Figure 5 shows the curves of frequency response due to the variation of the amplitude of vibration of the machine (X_1/δ_{st}) with the speed of the machine (ω/ω_1). There are two curves where the dashed is the response of the primary system and the full-line curve is the total system response (primary + absorber). It can be observed in the curves below what was previously evaluated where $X_1 = 0$ at $\omega = \omega_1$ ($\omega / \omega_1 = 1$) when using the dynamic absorber it is clear that there is a drastic reduction of the vibratory motion at that time.

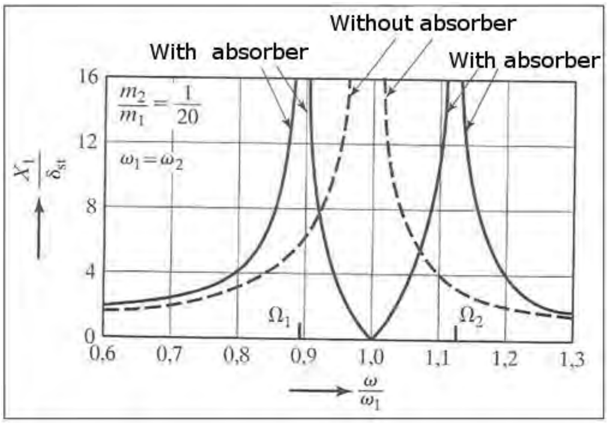


Figure 5. Effect of an undamped vibration absorber on the response of the machine (Rao, 2009).

Thus, in order to have an effect efficient dynamic absorber, its natural frequency (ω_2) must be equal to the frequency of the main system. This condition must be reached from the choice of the mass and spring that compose the ADV.

3. METHODOLOGY

To validate the theoretical framework it was designed a passive type of dynamic absorber. The configuration is shown below. The ADV was modeled and simulated by the dynamic module of Autodesk Inventor 2012 student version. To validate the theoretical framework it was designed a dynamic absorber passive type the configuration shown below. The ADV was modeled and simulated by the dynamic module of Autodesk Inventor 2012 student version. Furthermore, the system (primary + ADV) was evaluated experimentally, their frequency responses through a piezoelectric accelerometer and Labview software. These tests and simulations were performed in the laboratory of mechanical simulation (LMS)

ISSN 2176-5480

and the experimental tests at the laboratory of vibrations (LVI) and Multidisciplinary Laboratory of Materials and Structures Active (LAMMEA), both from the Federal University of Campina Grande.

The Figure 6 shows the main system excited by a motor fixed to the bottom of the structure in form of a unbalanced system.

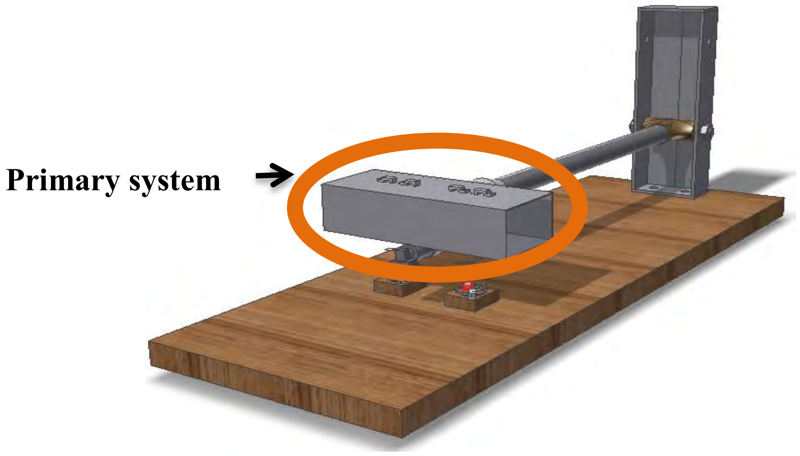


Figure 6. Main system in the environment Inventor Professional 2012.

The Figure 7 shows the secondary system added to the main system, in order to cancel the resonance frequency.

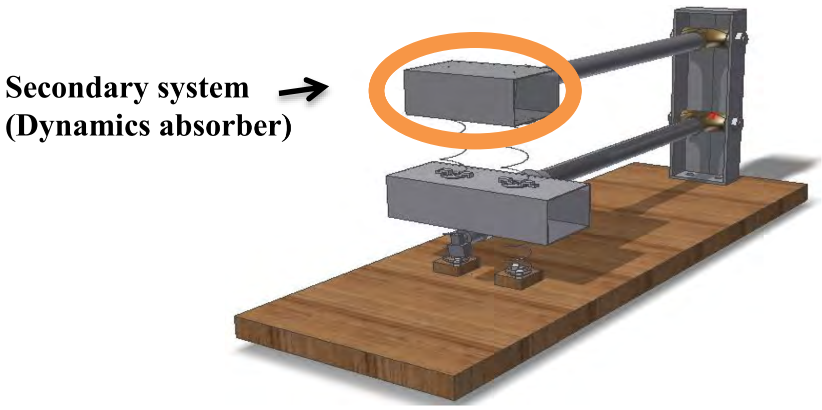


Figure 7. Secondary system in the environment Inventor Professional 2012.

The Figure 8 shows the complete system assembled with two components: the primary system and the dynamic absorber.

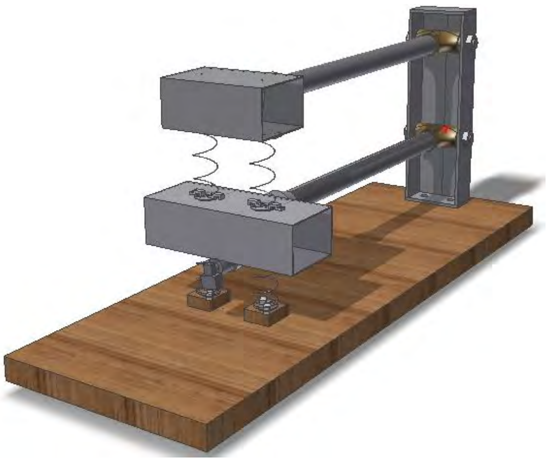


Figure 8. System mounted in the environment Inventor Professional 2012.

ISSN 2176-5480

For this experiment, it was used the following materials for the construction of the system: electric motor, carbon steel coil springs, unbalanced mass, shafts, bearings and aluminum rectangular profile.

The main system, as illustrated in Fig. 9, consists of an engine which has a mass unbalance responsible for the excitation of the system. The total mass of this system is 700 grams, the springs fixed in this component have a stiffness constant 1500 N/m and 1640 N/m, these values were determined by Instron E10000 at the LAMMEA at UFCG.

Figure 9. Main System with engine.

The secondary system illustrated below in Fig. 10, comprises a mass of 338 grams, it is comprised by two springs whose stiffness constant is 772 N/m and 740N/m, respectively. This small difference is probably because of the manufacturing process.

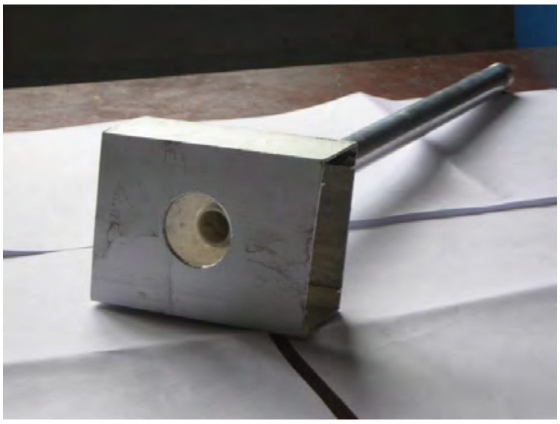


Figure 10. Secondary system.

The Figure 11 shows the assembled system and ready for the test phase.

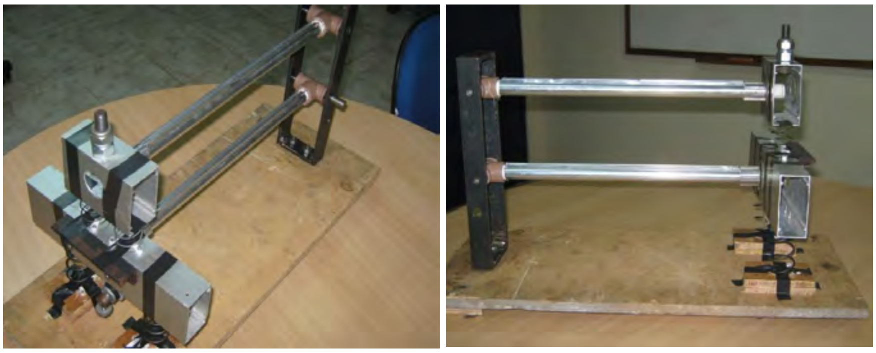


Figure 11. Mounted system.

In the experiment it was decided to evaluate in two distinct stages. In the first stage was generated a frequency response curve with the use of a piezoelectric accelerometer and the LabView software, only for the main system. Then the vibration neutralizer was added and again it was generated a curve of frequency response.

In both experiments, the test conditions were the same, where the tensions generated by a power supply in the engine that was ranged from 0 V to 8 V.

To determine the mass of the dynamic vibration absorber, it was determined the natural frequency of the main system, which must be equal to the natural frequency of the secondary system. So, the total mass of the main system is 700 grams and the stiffness constants of the springs are 1500 N/m to 1640 N/m.

To m_1 mass:

Data: m_1=0,7 Kg ; k'_1= 1500 N/m $k"_1$= 1640N/m

Logo, k_{eq1} = K'_1+ $K"_1$ = 1500 + 1640 = **3140 N/m**

Thus, $\omega_{n1} = \sqrt{\frac{k_{eq1}}{m_1}}$ = **66,98 rad/s ou 10,65 Hz**

To m_2 mass:

Data: k'_2= 772 N/m ; $k"_2$= 740 N/m

Logo, k_{eq2} = k'_2+ $k"_2$ = 772 + 740 = **1512 N/m**

Condition: $\omega = \omega_1 = \omega_2 = 66{,}98$ rad/s

Thus, $\omega_{n2} = \sqrt{\frac{k_{eq2}}{m_2}} \Rightarrow m_2 = \frac{k_{eq2}}{(\omega_{n2})^2} \Rightarrow m_2 = 0{,}337$ Kg

4. RESULTS AND DISCUSSION

The Figure 12 shows the simulation done in the Autodesk Inventor, for the excitation of the primary system. It is noted that the graph of the frequency response shows only one peak corresponding to the resonance region, because the system has 1 degree of freedom.

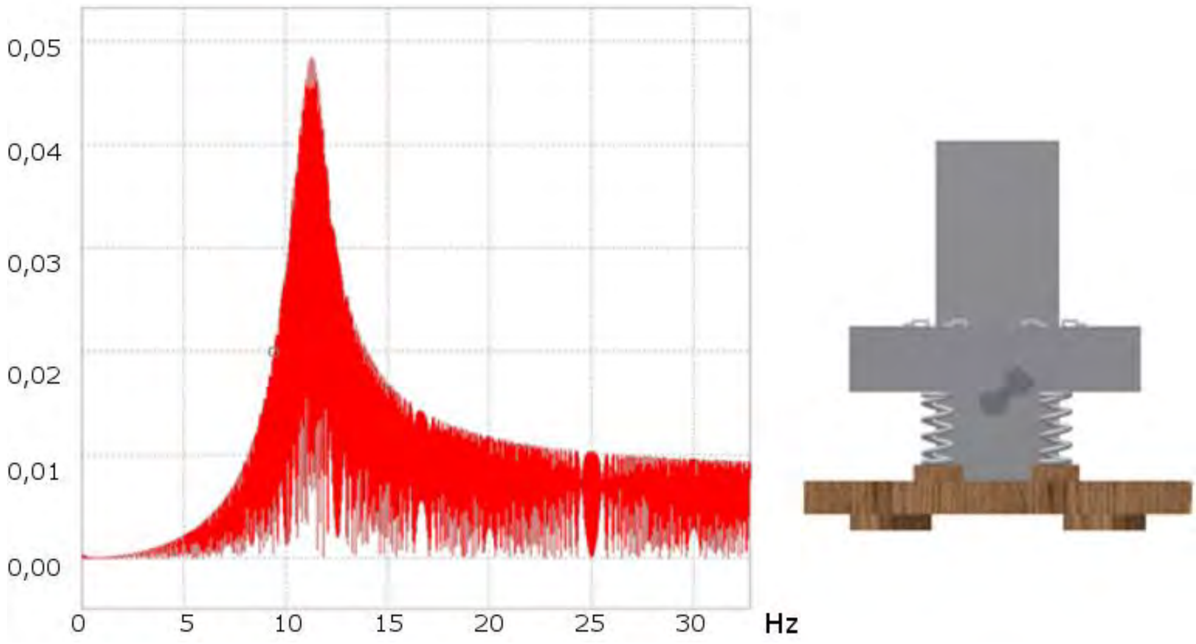


Figure 12. Frequency response curve for the main system.

ISSN 2176-5480

The Figure 13 shows the simulation with the addition of the secondary system. In this graph, with the addition of the dynamic absorber, the system has two degrees of freedom and two peaks are expected as could be seen in the graph of the frequency response below.

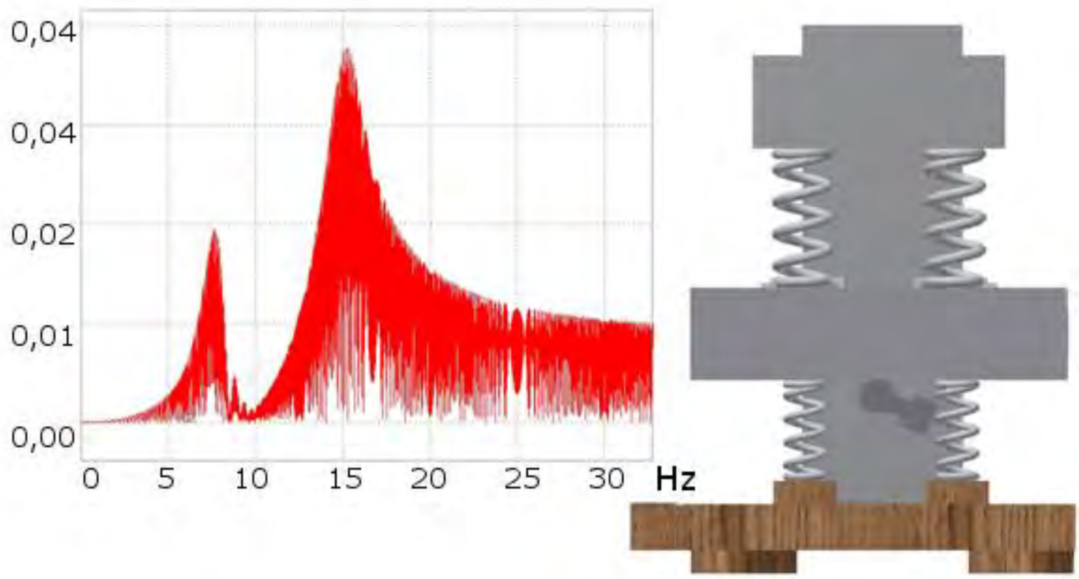


Figure 13. Frequency response curve for the system with ADV.

Evaluating the Fig. 12 and Fig. 13 it is clear that near to the maximum amplitude achieved for the primary system was close to 10 Hz and can be seen in the frequency response curve of the primary system + ADV, in that frequency the oscillation decreases sharply, showing the effect of ADV.

In Figure 14 the graph shows the amplitude in the frequency domain found experimentally for the system with one degree of freedom (primary system) which shows only a peak area, corresponding to resonance at a frequency 10 Hz. Furthermore, as there is a structural damping this peak does not go to infinity, with a limit as shown in the figure below.

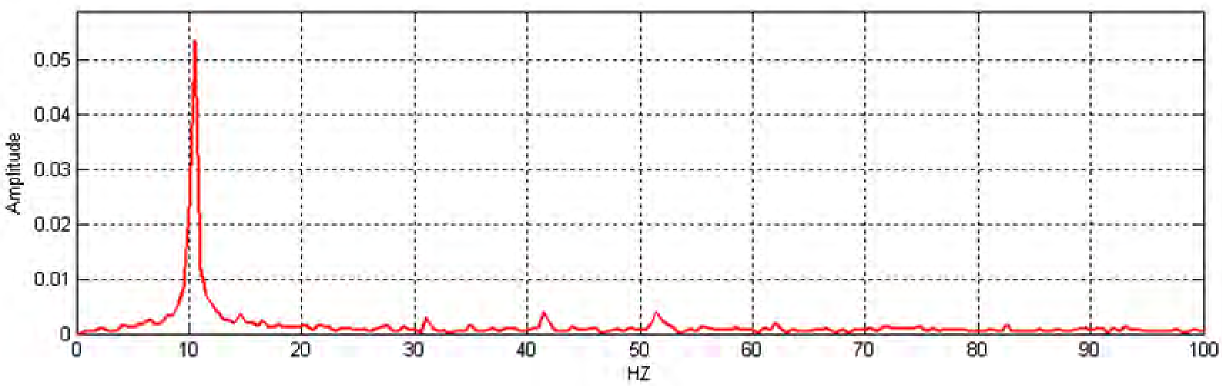


Figure 14. Graph of amplitude versus excitation frequency vibrations without the neutralizer.

The following graphin the Fig. 14 is related to the system composed of a primary system and the dynamic absorber, in other words the system has two degrees of freedom and have two peaks. This system was designed with the intention that the absorber would allow the system to work in the primary resonance region with the least possible extent. This is possible because the dynamic absorber system works against the force of the secondary system, overlapping the chart 14 15 overlaps the chart we can see that the resonance shown in Fig. 14 will be the minimum amplitude in chart 15, making this system work as a true ADV.

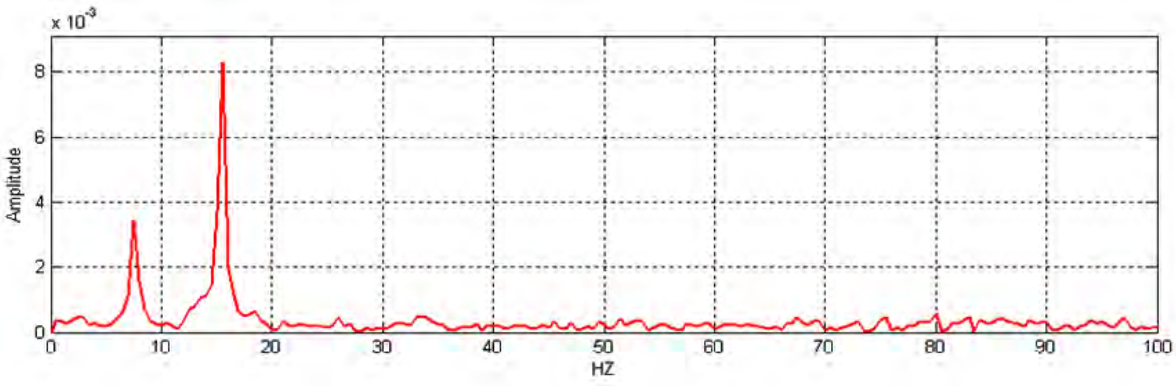


Figure 15. Graph of amplitude versus frequency of the excitation vibration neutraliser.

The table below represents the calculated natural frequencies by three methods: analytical, computational simulation made by the inventor software and experimental study in the laboratory. In addition, it was calculated the error between the analytical method with the simulation and experimental results.

$$\text{Error}=\left|\frac{Fexp-Fana}{Fana}\right| x100\% = \left|\frac{10{,}50-10{,}66}{10{,}66}\right| = 1{,}5\%$$

$$\text{Error}=\left|\frac{Fsim-Fana}{Fana}\right| \text{x}100\% = \left|\frac{10{,}95-10{,}66}{10{,}66}\right| = 2{,}7\%$$

Table 1. Comparison of results obtained by the three methods.

	Analytical	Simulation	Experiment
Frequency (Hz)	10,66	10,95	10,50
Error (%)		2,7	1,5

Where, F_{exp}, F_{ana}, F_{sim} are the natural frequencies found experimentally analytically and by simulation respectively.

5. FINAL CONSIDERATIONS

After the analytical, experimental and simulation results, it is concluded that: there was the verification of the effect of the dynamic vibration absorber of the primary system adopted. This was demonstrated at the computer simulation and the experiment.

The simulation results proved satisfactory when compared with the experimental results and obey what has been said in the literature.

The prototype built showed that the secondary system, the neutralizer vibration, decreased maximum amplitude in the resonance region generating two new amplitudes.

The errors between the experimental theoretical results and the simulation were admissible because they were below to 5%.

6. ACKNOWLEDGEMENTS

The authors of this paper thank the Unit of Mechanical Engineering, Federal University of Campina Grande (UAEM) and the laboratories involved, LAMMEA, LVI and LMS, as well as funding agencies that keep these laboratories.

7. REFERENCES

Aquino, Alberdan Santiago, 2009. *Controle de vibração de um sistema sob desbalanceamento rotativo utilizando atuador de liga com memória de forma*. Tese de Doutorado, Universidade Federal De Campina Grande, Campina Grande, Paraíba.

Borges, R. A., 2008. *Contribuição ao estudo dos absorvedores dinâmicos de vibrações não-lineares*. Tese de Doutorado, Universidade Federal de Uberlândia, Brasil.

ISSN 2176-5480

Fernandes, v. ; Castilho Filho, e. L. V. ; Melo, G. P. 2010. Optimum positioning of dynamics vibration absorbers applied to pipes using finite elements numerical analysis. In: *2º International Conference On Engineering Optimization-EngOpt2010*. Lisboa, Portugal.

Marques, R.F.A., 2000, *''Estudo Teórico e Numérico de Absorvedores Dinâmicos de Vibrações Ativos e Adaptativos''*, Dissertação de Mestrado, Universidade Federal de Uberlândia, Uberlândia, MG.

Mercado, Elvys. C.; Lima, Raul G.; Tamai, Edilson H.; Bistafa, Sylvio R.; Fujimoto, Fábio; Belardo, Carlos. 2007. "Metodologia para Projetar Absorvedores Dinâmicos usando Métodos de Elemento Finito", 9 f., In: *8º Congresso Iberoamericano De Engenharia Mecânica-CIBIM2007*. Cusco,Peru.

Ohana, G.J.;Vale, A.M.; Soeiro, N.S.; Melo, G.V. 2011. Projeto e dimensionamento de absorvedor dinâmico viscoelástico para controle de vibração da tampa de inspeção de um reator elétrico. In: *I Workshop de vibração e acustica da região norte-2011*. Tucuruí, Brasil.

Rao, S.S., 2009. *Vibrações Mecânicas*. Pearson Prentice Hall, 4nd edition.

8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.