DIAGNOSIS OF FAULT USING STATE OBSERVERS IN SYTEMS WITH DYNAMIC VIBRATIONS ABSORBERS(DVAs)

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Abstract. *In the last years the Dynamic Vibrations Absorbers (DVAs) have been widely used in machines, mechanical equipments, and civl construction aiming to assure satisfactory operation conditions, safety and confort. In this context there has been a lot of effort in order to design control systems abble to attend new developed structures. The main goal of this work is to use DVAs with vibrating blade and discribe the technique of State Observers to detect fault in systems varyng in frequency(to be apply in the next work). The purpose is to syncronize a DVA for a specific primary structure and apply the global and robust State Observers to detect and locate fault in especified parameters of the system. With the preliminary results it was possible to design a DVA using the finite element method(plate elements) in the vibrating blade. The primary structure was was excited with 60 and 450 rad/s harmonic forces.*

Keywords*: dynamic vibrations absorber, fault detection, state Observers.*

1. INTRODUCTION

With the increase in production process, are more and more demands of the industries for machines and equipment capable of executing a greater number of functions in less time and in many cases to be capable to act continuously. This leads to the fact of that these systems are submitted the high dynamic forces. Normally these mechanisms are expensive and therefore one of the major concerns of industry is to keep its equipment functioning without necessary breakdowns. With this constant concern, in the last times, we verify much development of new techniques of detection and localization of faults in mechanical systems submitted dynamics loads. In order to guarantee continuo operation of the mechanical systems, they must be supervised and monitored so that the faults are diagnosed and repaired as fast as possible, if not so the disturbance in normal operation can to take the one's a deterioration of the performance of the system or the dangerous situations. Robust observers can reconstruct the states not measured or estimate the motion of the system that can't be measured directly. Thus, faults can to be detected without knowledge of the motion at many points in the system by being able to monitor them through the reconstruction of the states. The existing methodologies using state observers are usually used in control problems and detection of possible faults in sensors and in instruments.

This work searches to develop one technique of identification, localization and minimization of imperfections of a primary system (main structure) connected to a DVA (secondary structure). First, in an interactive process, the structural properties of the DVA (secondary structure) are calculated and this, in turn, generates a force of equal intensity to the excitement force, however in opposition of phase to this, minimizing the vibrations in a band of desirable frequency, in the majority of the times searches the frequencies where the oscillations are the greatest. This phenomenon is known as anti-resonance. After that, a set of state observers is used, which can reconstruct the states not measured or the values proceeding from points of difficult access in the system complete (primary structure connected to the dynamic vibration absorber type vibrating blade) with the capacity to attenuate vibrations of this structure monitored in a frequency band. Techniques of identification of parameters using orthogonal functions are necessary so that the parameters of the system are calculated and these serve as given of entrance in the use of the technique of state observers. Moreover, when if it does not know the forces of excitement in the primary structure, the use of identification techniques becomes, which also are entered for the state observers, being able finally to realize the diagnosis of the damage of the structure to be followed (Morais, 2006).

2. BIBLIOGRAPHICAL REVISION

2.1. Identification of Parameters

Identification Methods of forces or parameters, with the objective of diagnosis of faults in mechanical systems, using orthogonals functions, have been developed since the end of 80's until the current days. These methods have begun with Chun (1987), Melo and Steffen (1993) used the series of Fourier for the identification of the structural parameters, and developed the inverse methodology for the identification of the forces. Pacheco (2001), in his doctor thesis, used some orthogonals functions for parameters identification through of the comparisons between the functions. Pacheco and Steffen (2003) published a work where the orthogonals functions were used for identification of parameters in non linear systems. In the work Melo (2004) studied the behavior of the error found in the identification of the parameters varying the number of terms of expansion of the orthogonal functions for some functions (Pezerat and Guyader, 2000). In the recent work Melo and Morais (2005a) of different way, had as objective to identify the forces and the parameters of the mechanical systems together, in the described works previous, the identification of the parameters alone was possible with the previous knowledge of the inputs.

2.2. State Observers

Luenberger (1964) states that the major part of the theory of modern control is based on the assumption that the state vector of the system to be controlled is available fro direct measurement. However, in many practical situations, just few of output database available. The author shows how the inputs and outputs that are available can be used to build an estimate observer, or just observer. This work states the state observer theory. Ge, W. and Fang, C. Z. (1988) have described a novel conception for the detection of components under failure by robust observation. Considering a mathematical model corresponding to "m" components coupled by non-estimated states. They have determined the design of devices to monitor the operation of those "n" components and faults detection. In the case an observable system, some first or superior order components can be monitored for the purpose of diagnosis without information of possible faults modes. Due to the observer robustness, the authors have analyzed some reactions such as linearization and measurement errors, noise presence, numerical errors, and so on. Melo, G. P. (1998) has developed a methodology for the detection and localization of faults in mechanical system employing reduced order state observers. He has shown the way non measured states can be reconstructed. By "means" of robust state observers, he could provide localization of faults, with the help of several robust state observer data bank, for each system parameter, he has proved that it is possible to quantify the system faults. He has preserved computing simulations and laboratory experiments "validating" the theory. Lemos (2004) has presented the state observer methodology for detection and location of faults in rotation systems, taking into account their foundations. According to him, the state observer methodology is able to reconstruct non measured states or estimate values coming from difficult access locations in the system. On fact, those faults can be detected without the need for a direct measurement. Morais (2006) in its M. Sc. Dissertation developed a methodology of State Observers using unknown inputs.

2.3. Dynamic Vibrations Absorbers (DVAs)

Breaking itself of the principle of functioning of the DVAs, which is based on the generation of a force of equal intensity to the excitement force, however in opposition of phase to this, it must be chosen the structural parameters of the DVA so that this excites the primary structure with a force of same intensity and in opposite direction to the specific external excitation proceeding from another way acting on the structure, providing the appearance of a known phenomenon of anti-resonance. Occurring such fact we can said that the DVA is synchronized.

Kotinda (2005), in its M. Sc. Dissertation, even though approached the possible loss of efficiency of the absorber in case that it has a variation in the excitation frequency, or in the parameters of primary and/or secondary structure through its degradation with the use. Three strategies had been analyzed.

Den Hartog (1956), and Ren (2001) had presented a study which use the optimization of the parameters of the DVA in order to get minimum amplitude of vibration in a band of the possible widest frequency. Both used a system of two degrees of freedom, or either, a primary structure excited by external forces subject the high levels of vibration, connected to a DVA. The difference between the realized studies is the point of anchorage of the damper of the DVA, while Den Hartog anchored the damper of its DVA in the primary structure; Ren anchored it in the soil. Cunha (1999), in its M. Sc. Dissertation, also presented some methods for the optimization of the parameters of the DVAs. Rade and Steffen (2000), had published a work where to get the excellent parameters of some DVAs connected to a structure to several g.d.l through techniques of modal coupling, they had used classic methods of optimization. Such techniques make the use of the passive DVAs calls.

Kotinda (2005) studied semi-active or adaptatives DVAs (DVAA), which can be capable to modify its parameters, especially of stiffness and damping.

The third technique studied for Kotinda (2005) was the use of the active dynamic vibration absorbers, which apply a force in the structure through of actuators.

3. METHODOLOGY

3.1. State Observers

The state observer's technique consists on a method capable of reconstructing the states in cases for which their measurement becomes difficult or even impossible. This way, flaws can be detected in those points even without the knowledge of their measurements, could also monitor them through those reconstructions of your states. On the other hand, this technique consists on developing a model for the system under analysis and comparing the output for the observer with the one of the system.

3.1.1. Concept of State Observer

Many control systems are based on the supposition that the full state vector is available for direct measurement, but in practice, not always all the variables are available, and those unavailable must be estimated. In such a way, an observer can be built to estimate them (Melo and Lemos, 2004). The schematic is as follows:

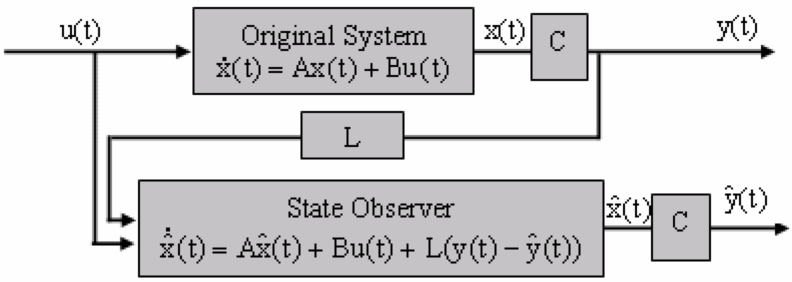

Figure 1 - Scheme of a State Observer

The observer is basically a copy of the original system; having the same input and almost the same differential equation. An extra term compares the actual measured output y(t) to the estimated output $\hat{y}(t)$.

Control systems using state observers can reconstruct the non-measured states or to estimate the values of points of difficult access in the system. However, the necessary condition for this reconstruction is that all the states should be observable (Luenberger, 1964; D'Azzo and Houpis, 1988).

3.1.2. Observers of Global and Robust State

The observer of global state is responsible for the detection of the fault, the robust state observer is responsible for the location of the same one. The global observer is a copy of the original system, and analyzes all the system detecting possible faults. The robust state observer can detect the fault if this occurs in the parameter for which it was projected. We have to project a bank of robust observer, each one in relation to a parameter to be monitored, to become possible a good location of fault.

When the system is functioning adequately, without indications of faults, the observer of global state answers equal the real system. When one component of the system in question starts to fail, the state observer feels the influence of this process quickly. The objective is using this effect of the state observer to locate and to quantify the fault in the mechanical system. They are put in a bank of observers and the RMS values of the differences between the real signs in displacement (measured) and the generated for the observers are analyzed in a unit of logical decision that it analyzes the trend of the progression of the fault and sets in motion, when will be necessary, an alarm system. The alarm system can be initialized when we have a parameter variation.

3.2. Dynamic Vibrations Absorbers (DVAs).

The methodology presented to follow corresponds to the principle of functioning of the type simplest of Dynamic Vibrations Absorber. In this case we work with Dynamic Absorber type vibrant blade.

Either a vibratory system of two degrees of freedom without damping composed for a primary structure (m_1, k_1) and for a secondary structure, the proper DVA (m_2, k_2).

It's admitted that the primary structure is excited by a harmonic force of F_0 amplitude and frequency of excitation, of fixed value, that does not coincide necessarily with the natural frequency of this structure, express second:

$$F(t) = F_0 . e^{i.\Omega.t} \tag{2}$$

To syntonize the DVA the parameters must be chosen (m_2, k_2) in order to annul the force of excitation *F(t)*. It will be chosen the natural frequency of the DVA equal to the frequency Ω of the force, the harmonic reply of the primary mass m_1 will have null amplitude for this frequency of excitation. To demonstrate this affirmation, the motion equations of connected system can be written:

$$m_1.\ddot{x}_1(t) + (k_1 + k_2).x_1(t) - k_2.x_2(t) = F(t)$$
$$m_2.\ddot{x}_2(t) + k_2.[x_2(t) - x_1(t)] = 0 \qquad (3)$$

The forced vibration of this system will be of the form:

$$x_1(t) = X_1.e^{i.\Omega.t}$$
$$x_2(t) = X_2.e^{i.\Omega.t} \qquad (4)$$

Thus, substituting the previous equations, the differentials equations are changed into the following algebraic equations:

$$X_1.(-m_1.\Omega^2 + k_1 + k_2) - k_2.X_2 = F_0$$
$$-k_2.X_1 + X_2.(-m_2.\Omega^2 + k_2) = 0 \qquad (5)$$

Introducing the notation,

$\omega_1 = \sqrt{\frac{k_1}{m_1}} \mapsto$ Natural frequency of the primary structure, considered separately;

$\omega_2 = \sqrt{\frac{k_2}{m_2}} \mapsto$ Natural frequency of the secondary structure, considered separately;

The following expression for the X_1 amplitude of the primary structure can be gotten from the Eq. (5):

$$\frac{X_1}{F_0.k_1^{-1}} = \frac{\left[1 - \left(\frac{\Omega}{\omega_2}\right)^2\right]}{\left[1 + \frac{k_2}{k_1} - \left(\frac{\Omega}{\omega_1}\right)^2\right].\left[1 - \left(\frac{\Omega}{\omega_2}\right)^2\right] - \frac{k_2}{k_1}} \qquad (6)$$

It can be observed in the Eq. (6), the amplitude of the primary structure is null when the numerator is equal the zero, occurring when the frequency of the excitement force it is same that the natural frequency of the absorber (ω_2).

If $X_1 = 0$ *e* $\Omega = \omega_2$, is had that the amplitude of vibration of the mass m_2 is given by:

$$X_2 = -\frac{F_0}{k_2} \qquad (7)$$

In this situation, the force exerted for the secondary structure on the primary structure is given by:

$$F_0 = -k_2.X_2 \qquad (8)$$

Of this form, an equal and opposing force to the excitement force is exerted on the primary structure for the secondary structure, having as consequence the balance of the system.

4. SIMULATED RESULTS

We worked with computational simulation of a DVA connected to a system shaped by plate elements. First we separated the blade for the method of finite elements, using plate elements, whose matrices mass and stiffness used referring to an element is given for Waideman, 2004. We can see in Fig. 2, that the plate element used has translation in *z*, rotation in *x* and *y*.

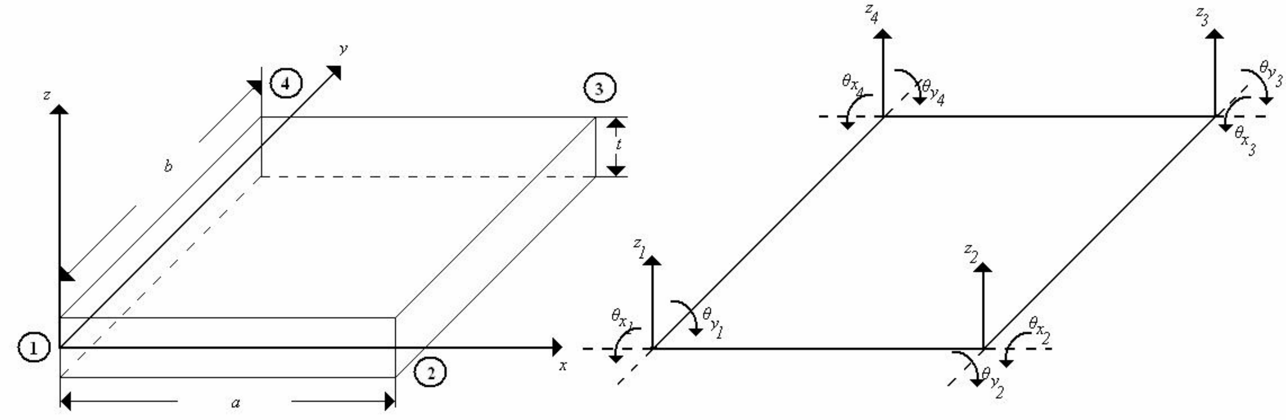

Figure 2. Degrees of freedom of a plate element.

The ADVBV contain 9 elements, composition for 20 knots with 3 degrees of freedom (dof) each one. Therefore the used DVA contains 60 dof. It can be observed, through of Fig. 4, that the shaped blade possesses the central element with all the dof restricted or more necessarily with its solidary displacements to the point of coupling of the primary structure.

A plate was used as primary structure. This was divided in 144 elements, contend therefore 432 dof, with dynamic characteristics identical to the presented in Fig. 2. The used parameters were:

- $Lx = 50cm$;
- $Ly = 40cm$;
- $Lz = 5cm.$

Were considered in such a way for the primary structure how much for secondary a proportional damping supplied for the equation, where used they are 1 and 1E-7, respectively. The Fig. 3 presents the built in surface and the place of application of the force of excitation in the primary structure.

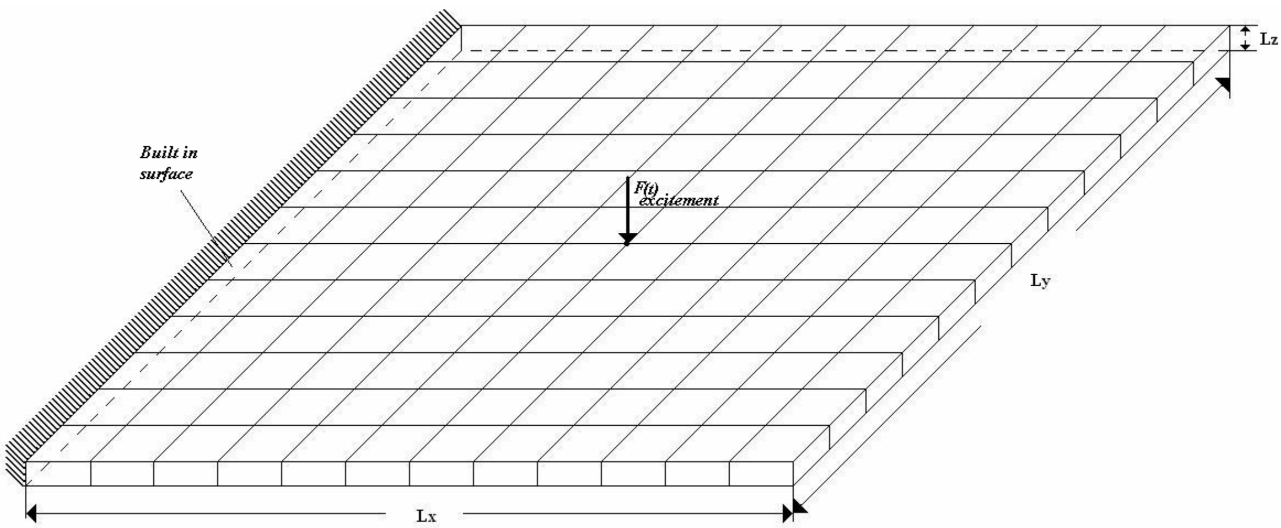

Figure 3. Localization of the excitation force in the primary structure.

The method of connection of the two structures used and presented to follow can be seem in Maia et all, 1997.

Considering two structures without damping P and S whose matrices of mass and stiffness have order n_P and n_S, respectively. The equations of equilibrium for each subsystem, acted on only by interconnecting forces, are:

$$\begin{bmatrix} [{}_PM_{ii}] & . & [{}_PM_{ic}] \\ . & . & . \\ [{}_PM_{ci}] & . & [{}_PM_{cc}] \end{bmatrix}_{n_P\,x\,n_P} \begin{Bmatrix} \{{}_P\ddot{u}_i\} \\ . \\ \{{}_P\ddot{u}_c\} \end{Bmatrix}_{n_P x1} + \begin{bmatrix} [{}_PK_{ii}] & . & [{}_PK_{ic}] \\ . & . & . \\ [{}_PK_{ci}] & . & [{}_PK_{cc}] \end{bmatrix}_{n_P\,x\,n_P} \begin{Bmatrix} \{{}_Pu_i\} \\ . \\ \{{}_Pu_c\} \end{Bmatrix}_{n_P x1} = \begin{Bmatrix} \{{}_P0_i\} \\ . \\ \{{}_Pf_c\} \end{Bmatrix}_{n_P x1}$$

$$\begin{bmatrix} [{}_S M_{ii}] & . & [{}_S M_{ic}] \\ . & . & . \\ [{}_S M_{ci}] & . & [{}_S M_{cc}] \end{bmatrix}_{n_S\,x\,n_S} \begin{Bmatrix} \{{}_S \ddot{u}_i\} \\ . \\ \{{}_S \ddot{u}_c\} \end{Bmatrix}_{n_S\,x1} + \begin{bmatrix} [{}_S K_{ii}] & . & [{}_S K_{ic}] \\ . & . & . \\ [{}_S K_{ci}] & . & [{}_S K_{cc}] \end{bmatrix}_{n_S\,x\,n_S} \begin{Bmatrix} \{{}_S u_i\} \\ . \\ \{{}_S u_c\} \end{Bmatrix}_{n_S\,x1} = \begin{Bmatrix} \{{}_S 0_i\} \\ . \\ \{{}_S f_c\} \end{Bmatrix}_{n_S\,x1} \tag{9}$$

Where subscript "c" mentions n_c to its involved coordinates in the physical connection and "i" to co-ordinate remains of the primary structures and secondary. The equations below characterize physically the compatibility of the displacements and the balance of forces between the subsystems suffering free vibrations.

$$\begin{aligned} \{{}_P f_c\} &= -\{{}_S f_c\} = \{f_c\} \\ \{{}_P u_c\} &= -\{{}_S u_c\} = \{0\} \end{aligned} \tag{10}$$

It can be observed that the complete system (primary structure + secondary structure) have $N = n_A + n_B - n_c$ degrees of freedom. The matrices of mass and stiffness of the two connected substructures are presented below.

$$[M]_{N\,x\,N} = \begin{bmatrix} [{}_P M_{ii}] & . & [{}_P M_{ic}] & . & [0] \\ . & . & . & . & . \\ [{}_P M_{ci}] & . & [{}_P M_{cc}]+[{}_S M_{cc}] & . & [{}_S M_{ci}] \\ . & . & . & . & . \\ [0] & . & [{}_S M_{ic}] & . & [{}_S M_{ii}] \end{bmatrix} \tag{11}$$

$$[K]_{N\,x\,N} = \begin{bmatrix} [{}_P K_{ii}] & . & [{}_P K_{ic}] & . & [0] \\ . & . & . & . & . \\ [{}_P K_{ci}] & . & [{}_P K_{cc}]+[{}_S K_{cc}] & . & [{}_S M_{ci}] \\ . & . & . & . & . \\ [0] & . & [{}_S K_{ic}] & . & [{}_S K_{ii}] \end{bmatrix} \tag{12}$$

$$[C] = \alpha.[M] + \beta.[K] \tag{13}$$

After esteem the matrices of mass, damping and stiffness of the system it was used Runge-Kutta Method of 4º order, in this case for a forced system. The modal and structural parameters had been recalculated. It can be noticed that the signals of reply gotten through of simulation are used as entered for the technique of parameters identification.

It was broken of the concept given for the methodology of Dynamic Vibrations Absorbers, where in a DVA, one of its multiples dof must vibrate with a natural frequency of same value that the frequency of excitement of the force acting on the primary structure. Therefore, a system was implemented where the matrices of mass and stiffness of each one of the elements of the DVA can be modified through 3 ways, until the condition so that the amplitude of displacement it primary system is next to zero, being able to affirm therefore that the DVA is synchronized. The value of the RMS (Root Mean Square) of the reply signal was used as criterion of stopped implemented in the program.

The three ways of alteration of the secondary structure are:

- Modification of the localization of the intent mass on the blade;
- Thickness of the intent mass;
- Length of the blade.

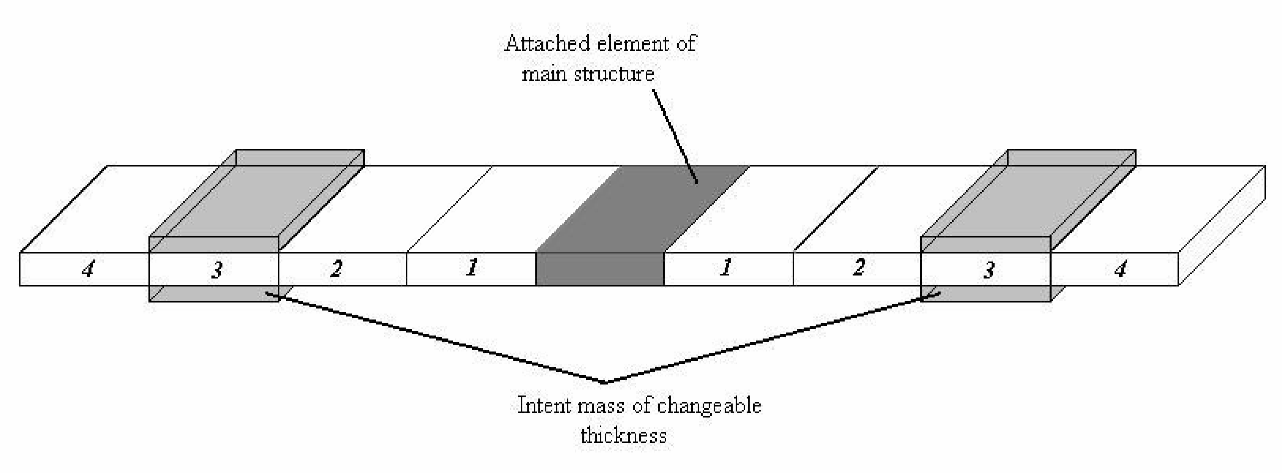

Figure 4. Division of the DVA. In the case the intent mass is on 3º element.

4.1. 1º. Case - Angular excitation frequency of 60 rad/s.

Applying a sine harmonic force given by Eq. (14), the DVA was synchronized in according with the design of Fig. 4, with longitudinal length of 5cm, transversal length of 2cm and thickness of 21 mm. We can verify that the DVA vibrated in accordance with its first way.

$$f(t)_{excit} = 5 * sen(60 * t) \quad (14)$$

The Fig. 5 presents it the signal of the respective force of excitation and the displacement of the primary structure.

4.2. 2º. Case - Angular excitation frequency of 450 rad/s.

Applying a sinusoidal harmonic force given by Eq. (15), the DVA was synchronized with the mass of thickness of 15mm on 1º element, whose blade has a longitudinal length of 3cm and a transversal length of 2cm. We can verify that the DVA vibrated in accordance with its first way again.

$$f(t)_{excitação} = 5 * sen(450 * t) \quad (15)$$

The Fig. 5 presents the signal of the respective excitation force and the displacement of the primary structure.

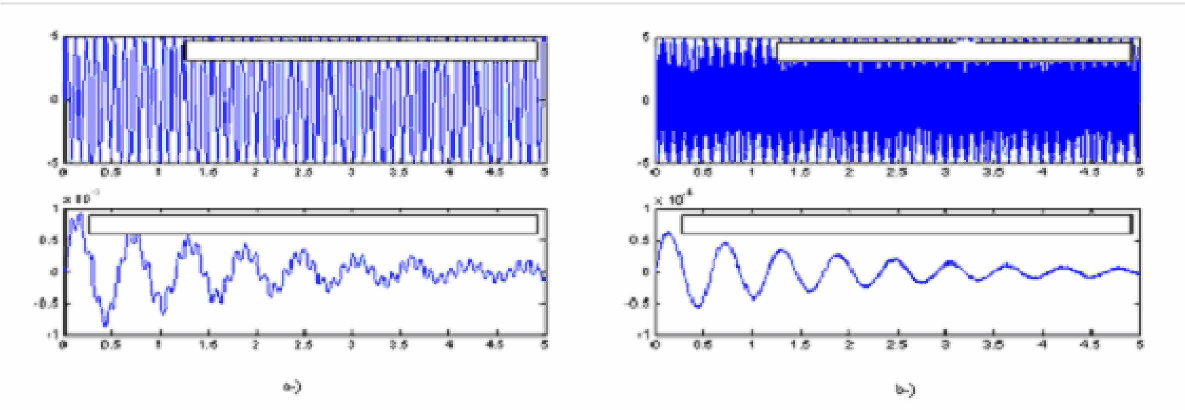

Figure 5. Signal of the force and displacement acting on the primary structure with synchronized ADV
a-)Excitation Force(N) x time(s) and Displacement(m) x time(s)- $\omega_{excit} = 60\, rad/s$;
b-)Excitation Force(N) x time(s) and Displacement(m) x time(s)- $\omega_{excit} = 450\, rad/s$.

5. DISCUSSIONS AND CONCLUSIONS

We had satisfactory results with the attenuation of the vibration of the primary structure in numerical simulation applications. The technique to be employed using State Observers to detect faults can reconstruct non measured states or to estimate the values of points of difficult access in a system, detecting faults in those points without the knowledge of measured data, monitoring them through the reconstructions of its states. It should be stressed that is satisfied if only if the analyzed system is observable using the number of measurements is satisfied carried out. In case this does not happen, new measurement should be carried out until the system becomes observable. Besides, the observers' eigenvalues should be selected to be a little at least to the left in the complex plan of the eigenvalues of the observed system in order to guarantee the stability and the fast convergence of the process. The numerical simulation and experimental works has shown the efficiency of this technique. This way, this methodology can be implemented for several types of mechanical systems. Breaking itself of the first gotten results, presented for Fig. 5, it could be observed resulted satisfactory like the attenuation of the vibration of the primary structure. It is important observe that we used a blade divided by finite elements and got goods and coherent results in DVAs.

In the sequence of this research we will have the implementation of State Observer to detect and locate possible faults in the general system (primary system connected to DVA). All the computational routines are being developed using software MATLAB 6.5.

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