SYSTEM OF VIBRATION CONTROL OF ROTATING MACHINERY BY APPLYING CONCENTRATED MASSES

Raúl Oscar CURADELLI
Ms. Eng., Graduate Student, CPGEC, Federal University of Rio Grande do Sul, Brasil
Ernesto Alves 369/402, Floresta, RS. POA. Brasil. Tel. 51-32247565.
e-mail: ocuradelli@yahoo.com

Ricardo Daniel AMBROSINI
Dr. Eng., Associate Professor, Structures Institute, Nat. Univ. of Tucumán, CONICET, Argentina
e-mail: dambrosini@herrera.unt.edu.ar

Marta Graciela Amani
Ms. Eng., Graduate Student, Structures Institute, Nat. Univ. of Tucumán, CONICET, Argentina
e-mail: amanimg@yahoo.com

Abstract. The main objective of this paper is the comparison between the efficiency of systems of vibration control of rotating machinery, in order to present alternatives for the cases in which it is not possible or convenient to use traditional systems. Moreover, the formulation of the theoretical dynamic load that should be used in the dynamic analysis is discussed. An experimental and numerical study is presented considering as an alternative system of mitigation of vibration, the incorporation of masses on the support structure. Masses of 5%, 10% and 15% of the total mass of the vibrating machinery are included in the analysis. A set of tests about a case study on the spring (traditional) and built-in masses (alternative) systems is presented. Moreover, a linear dynamic analysis of the models with the finite element code SAP 2000 was carried out. The behaviour of the built-in masses system is very good, reducing the acceleration response on the support structure, in the two first flexural vibration modes, to levels comparable to traditional systems.
Keywords. ***Vibration control, rotating machinery.***

1. Introduction

Many works have been published since around 1930, mainly in the analysis of base isolation systems of vibration control, which are summarized in many books (Richart, 1970; Macinante, 1984; Den Hartog, 1985). In the last years, advances have been achieved, especially in theoretical procedures to consider the problem of vibration control of machines (Feng et all, 1995; Zapfe et all, 1996; Natsiavas et all, 1996; McMillan et all, 1996,1997; Yi et all, 1998; Li et all, 2000; Cui et all, 2000). However, according to Arpaci, (1996), because most of the investigations on the absorbers are theoretical, special attention must be paid to the need to conduct experiments. For this reason, good quality experimental data can be useful for theoretical and numerical researches. In general, the vibrations produced by rotating machinery are reduced or mitigated by means of traditional systems of base isolation such as spring or viscoelastic materials. However, in many cases of rotating machines supported on plate structures it would not be possible to use this alternative of isolation. For example, machines not initially isolated that for any change in the support structure or the operating conditions, produce unexpected high levels of vibrations and it is inconvenient to stop the machine to install a base isolation system. In these cases, an attractive alternative is to use masses incorporated to the support structure.

The objectives of this paper are, first, the comparison between the efficiency of control vibration systems of rotating machinery, in order to present alternatives for traditional systems and, in the second place, to obtain guides towards the theoretical dynamic load that should be used in this type of problem. According to these objectives, an experimental and numerical study is presented considering as alternative system of mitigation of vibration the incorporation of masses on the support structure. Masses of 5%, 10% and 15% of the total mass of the vibrating machinery are included in the analysis. A set of tests about a case study on the spring (traditional) and masses (alternative) vibration control systems is presented. The time histories of the acceleration in different points are recorded. On the other hand, a numerical analysis using the finite element program SAP 2000 N (Wilson, 1997) was carried out.

2. Experimental model

2.1. Case study

In order to study the efficiency of the systems of vibration control, an air compressor was used as an excitation-generating machine. A wood table was selected as support structure in order to focus the attention on the important case in which the support structure is flexible. Moreover, important levels of vibration appear and can be analysed the efficiency of the systems more reliably.

Initially, in order to evaluate the severity of the excitation, the machine was mounted rigidly on the support structure as it shows in Fig. 1a. Then, two systems of vibration control were mounted: System 1, that uses helical springs, and System 2 that adds masses in the antinodes of the first two vibration modes of the support structure (Fig.

1b). Varying the mass of System 2 by steps of 5%, 10% and 15% of the total mass of the machine, the sensitivity in the efficiency of the vibration control was studied. In all the cases, the system responses were recorded.

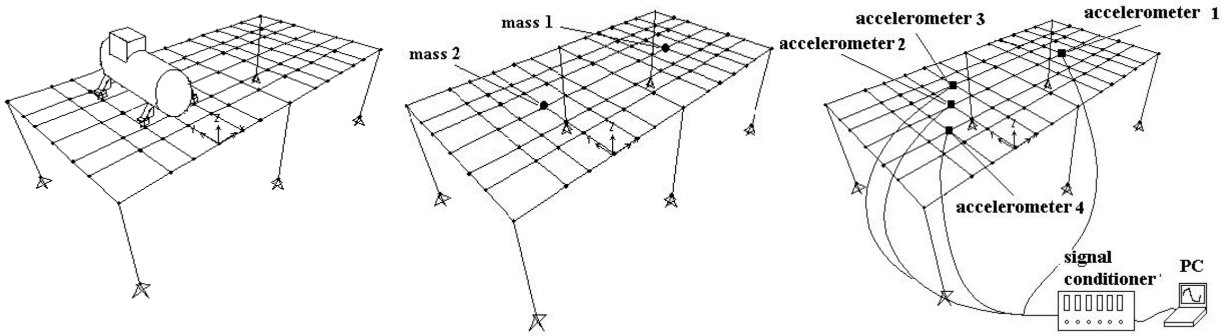


Figure 1. Case study. a) Assembly of the exciting machine, b) Masses location, c) Accelerometers disposition.

2.2. Experimental set-up and instrumentation

For the three mentioned configurations, with the machine acting in regime, the vertical acceleration in different points from the structure (Fig. 1c) was measured.

Four accelerometers KYOWA AS-GB were used to measure the dynamic response of the support structure. A dynamic strain amplifier KYOWA DPM-612B amplified the signal generated by the accelerometers. A data acquisition board Computerboars PCM-DAS16D/16 of 100 KHz was mounted on a notebook computer in order to record and process the signals by means of the program HP VEE 5.0 (Helsel, 1998). A scheme of the equipment is presented in Fig. 1c. Four channels were used in all tests and the signals were sampled at 500 Hz each channel.

2.3. Dynamic response

In this section, the system responses obtained in the forced vibration tests are shown for the three configurations above mentioned (with and without vibration control), measured in points 1 and 2 (see Fig. 1c). Initially, free vibration tests were performed obtaining 19 and 44 Hz as the frequencies of the two first flexural modes of the structure (see section 3.3). A resonant response could be expected between the second natural frequency (44 Hz) and the fourth component of the dynamic load

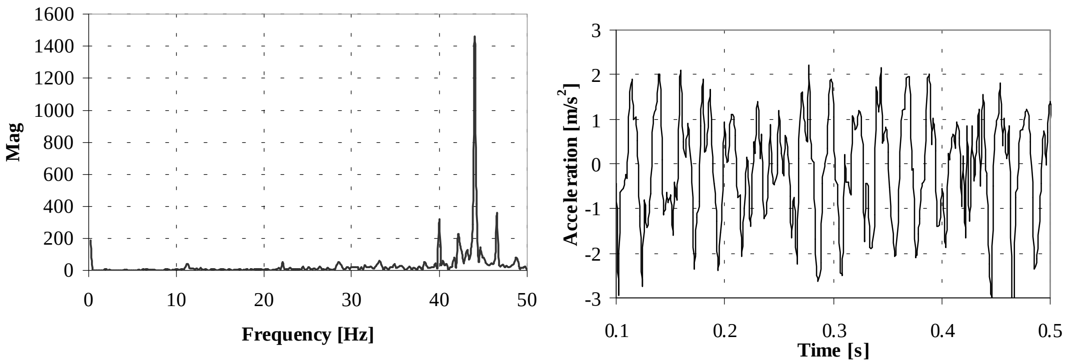
2.3.1. Point 1

In the frequency domain, (Fig. 2a) a main component in approximately 44 Hz, is observed. This frequency is next to the natural one of the second mode of vibration of the support structure, which is presented in section 3.3.

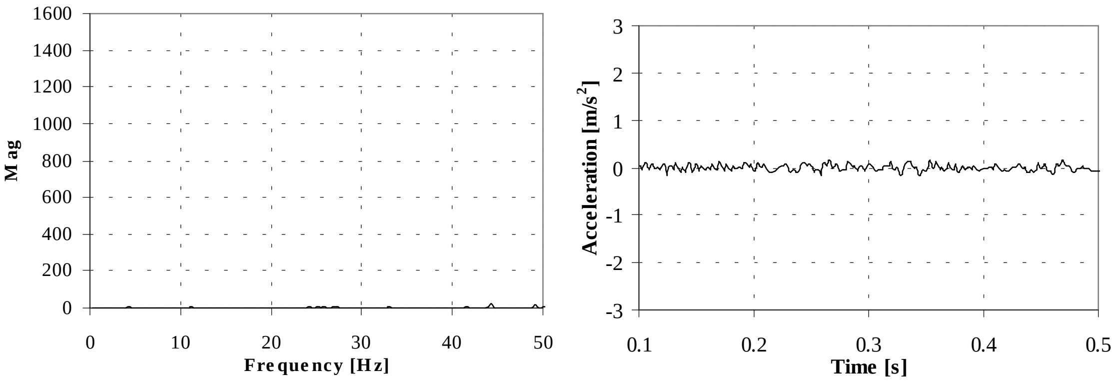
In Fig. 2b the time history and the spectrum, of the system mounted on helical springs are shown. It is to emphasize the remarkable reduction in the level of vibrations measured in this point.

For space reasons, only the system response with built-in masses of 15% of the total mass of the machine, is presented in Fig 2c.

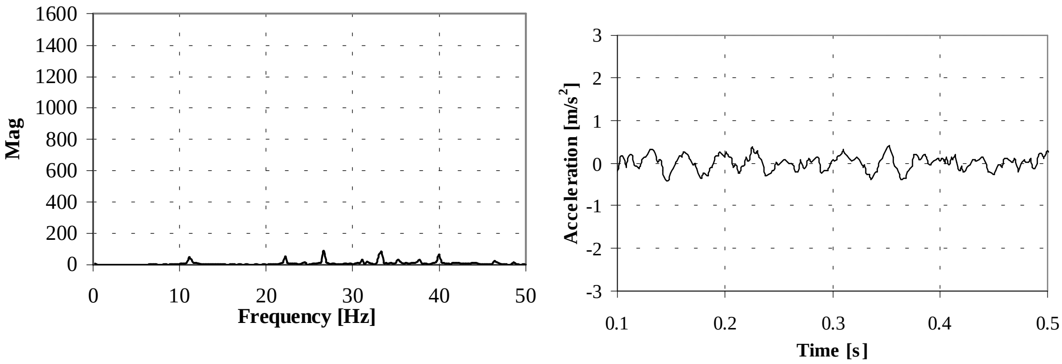
It is important to note that with the proposed system, the response obtained in point 1 of the structure shows levels of reduction of vibrations similar to those obtained by the traditional vibration control system.



a)



b)



c)

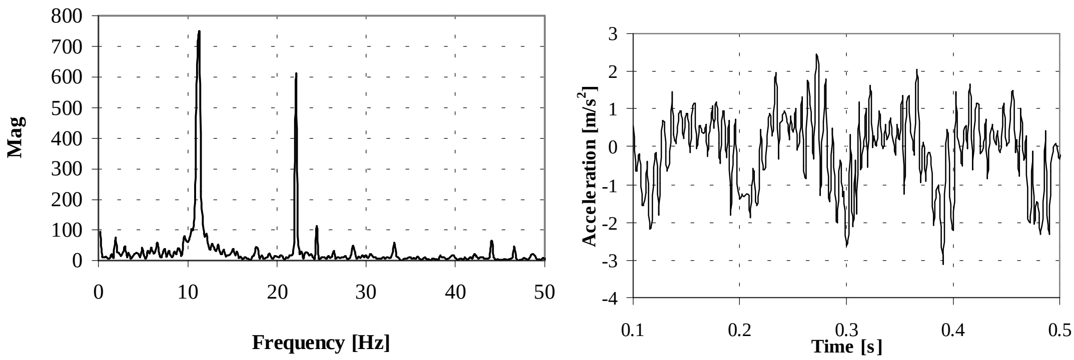
Figure 2. Experimental results. Point 1. Spectrum and time history. a) Assembly without vibration, b) Assembly with springs, c) Assembly with incorporated masses of 15%.

2.3.2. Point 2

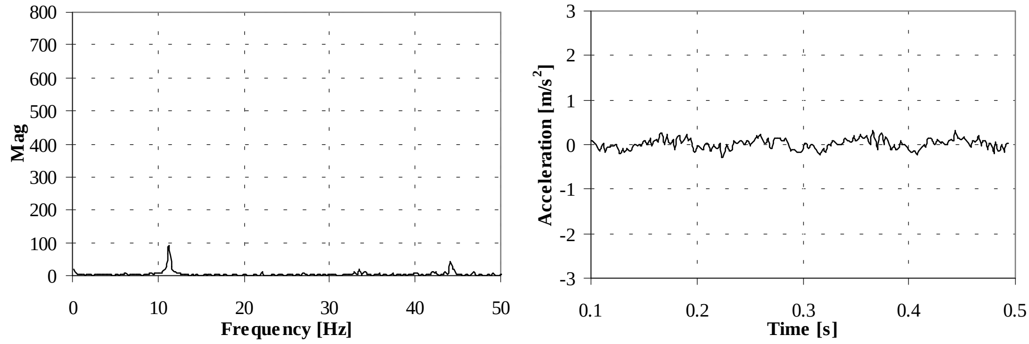
For the system without vibration control, in the frequency domain (Fig. 3a) two peaks in 11 Hz and 22 Hz appear. These frequencies agree with the main components of the excitation load.

In Fig. 3b the large efficiency obtained with the traditional system of vibration control in a measurement point that is underneath the machine, is observed. The vibration levels were reduced to less of 10% of the original level. In the spectrum, only a small peak in 11 Hz appears, that corresponds with the fundamental harmonica of the exciting load.

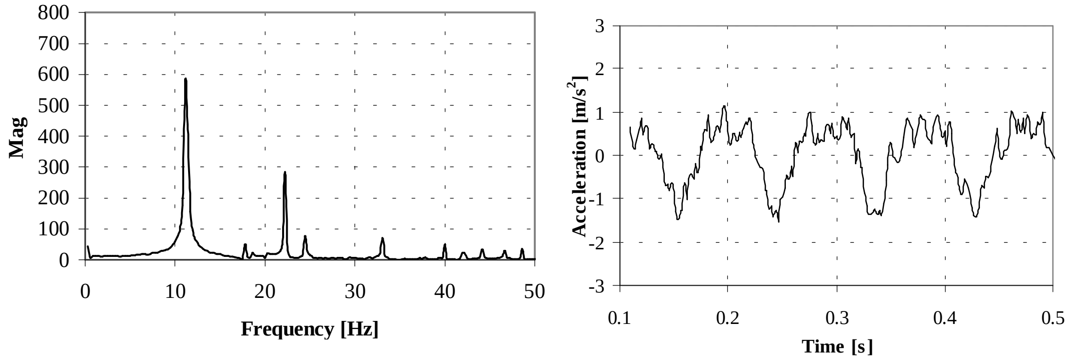
Figures 3c shows the response measured in the system with built-in masses for the case mentioned.



a)



b)



c)

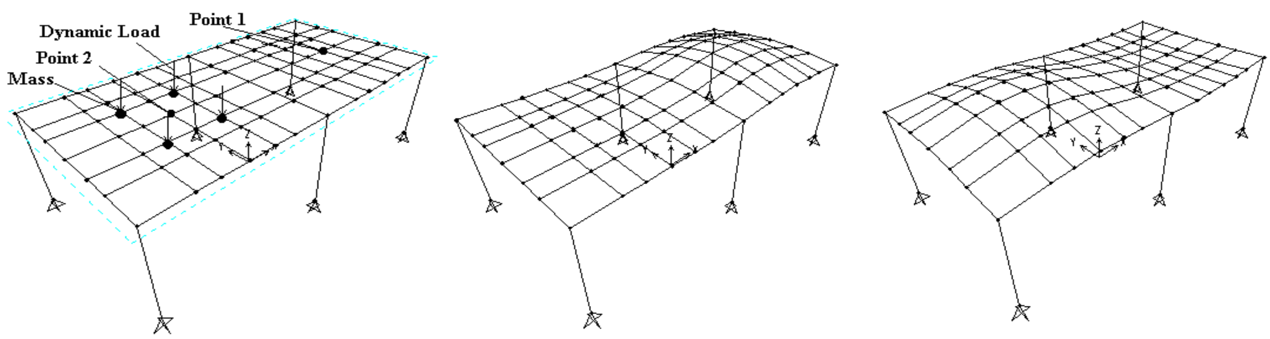
Figure 3. Experimental results. Point 2. Spectrum and time history. a) Assembly without vibration, b) Assembly with springs, c) Assembly with incorporated masses of 15%.

3. Numerical analysis

3.1. Description of the model

The behaviour of the complete prototype (structure-machine) was simulated in the software of Finite Element Sap2000 Nonlinear. The legs as well as the beams were modelled with type Frame elements and the plate with type Shell elements. In addition, concentrated masses were added (to represent the mass of the machine), located in nodes corresponding to the position of each one of the four legs of the machine (Fig. 4a). On these same nodes, the dynamic loads produced by the connecting rod-crank mechanism and the air pressure within the cylinder, were located. The method to obtain the dynamic load, in order to use it in the numerical model, is exposed in Curadelli et all (1999).

Figure 4a shows the mesh used in the board model with the representative masses of the compressor and the location of the dynamic load and the points 1 and 2 on which the study was concentrated.



a) b)

Figure 4. FEM Model. a) Mesh of the structure, c) First (19.2Hz) and second (44.2 Hz) mode of vibration.

3.2. Dynamic response

Initially, in the numerical analysis, the modal shapes and natural frequencies of vibration were determined. Figure 4b shows the first two bending modes of the machine-structure set without control of vibration. The values obtained for these modes were 19.2 and 44.2 Hz respectively, that agree with those obtained experimentally.

This section shows the response of the system, obtained through numerical analysis, on points 1 and 2 (Fig. 4a) for the cases: without and with vibration control using springs and incorporated masses. For space reasons, only the case that represents 15% of the total mass of the machine is presented.

3.2.1. Point 1

In the spectrum presented in Fig. 5a two components in 11 Hz and 22 Hz are observed. They correspond to the frequencies of the components of the load. The calculated response does not show the fundamental peak measured (Fig. 2a) in approximately 44 Hz.

For the case with helical springs a peak in 44 Hz is found as shown in Fig. 5b. For the case of built-in masses with 15% of the total mass of the machine (Fig. 5c) it does not reproduce all the peaks obtained in the measurement (Fig. 2c).

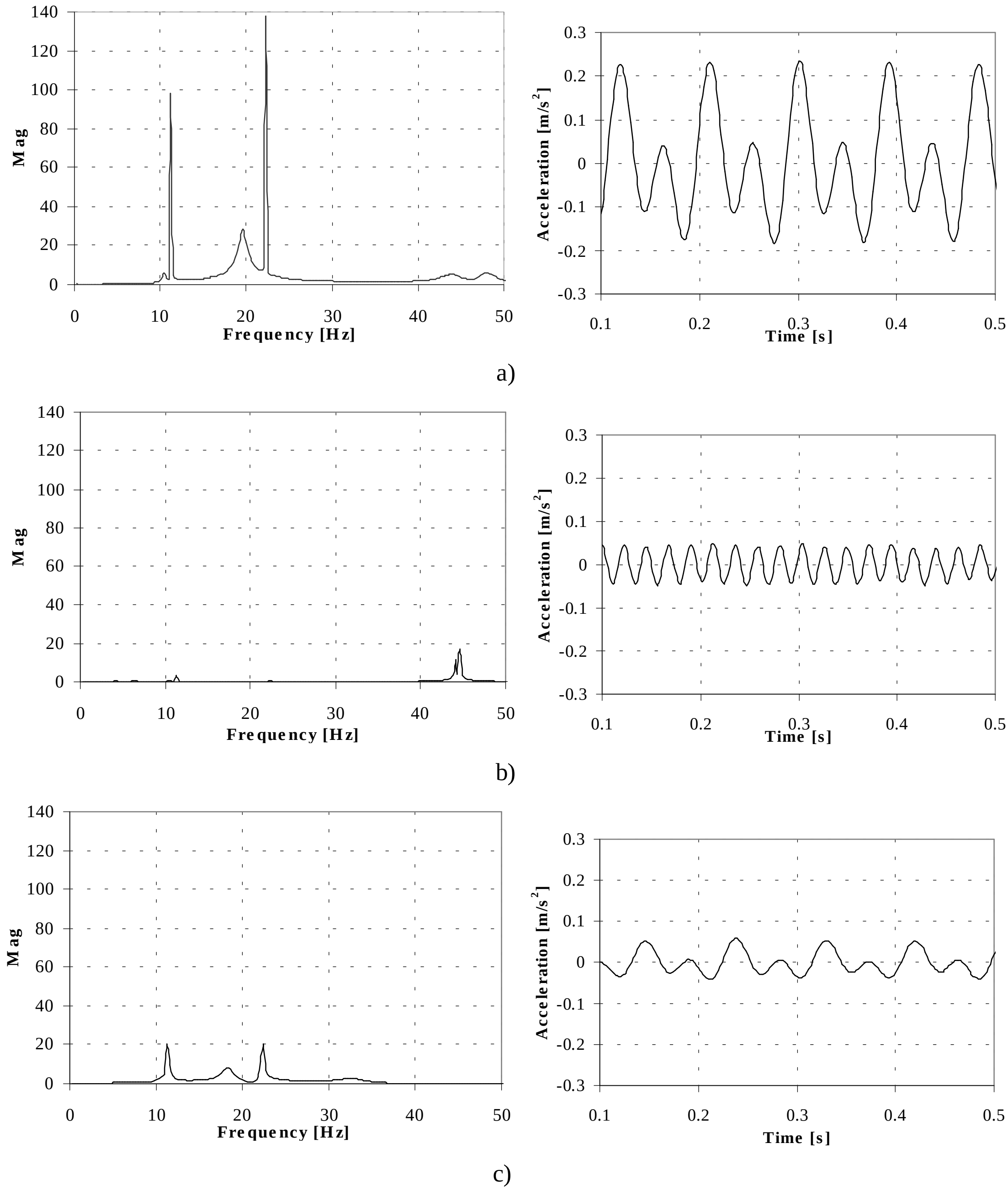


Figure 5. Numerical results. Point 1. Spectrum and time history. a) Assembly without vibration control. b) Assembly with springs. c) Assembly with incorporated masses of 15%.

3.2.2. Point 2

Figure 6a shows the response obtained in the numerical analysis, for the case of the system without vibration control. It is noted that the spectrum measured (Fig. 3a) is reproduced accurately.

For the case of the system with helical spring, the numerical analysis (Fig. 6b) catches the components measured in 11 Hz and 44 Hz (Fig. 3b) accurately. In the same way, in Fig. 6c good numerical approach of the experimental response is observed (see Fig. 3c).

4. Comparison and discussion

In this section a comparative study of the results obtained in the experimental and numerical analysis from the system with and without vibration control will be presented.

4.1. Efficiency. Comparison between experimental results

In order to assess the efficiency of the vibration control systems, only measured experimental results are compared. This is carried out by means of the frequency Spectrum analysis, Peak and R.M.S. values of accelerations.

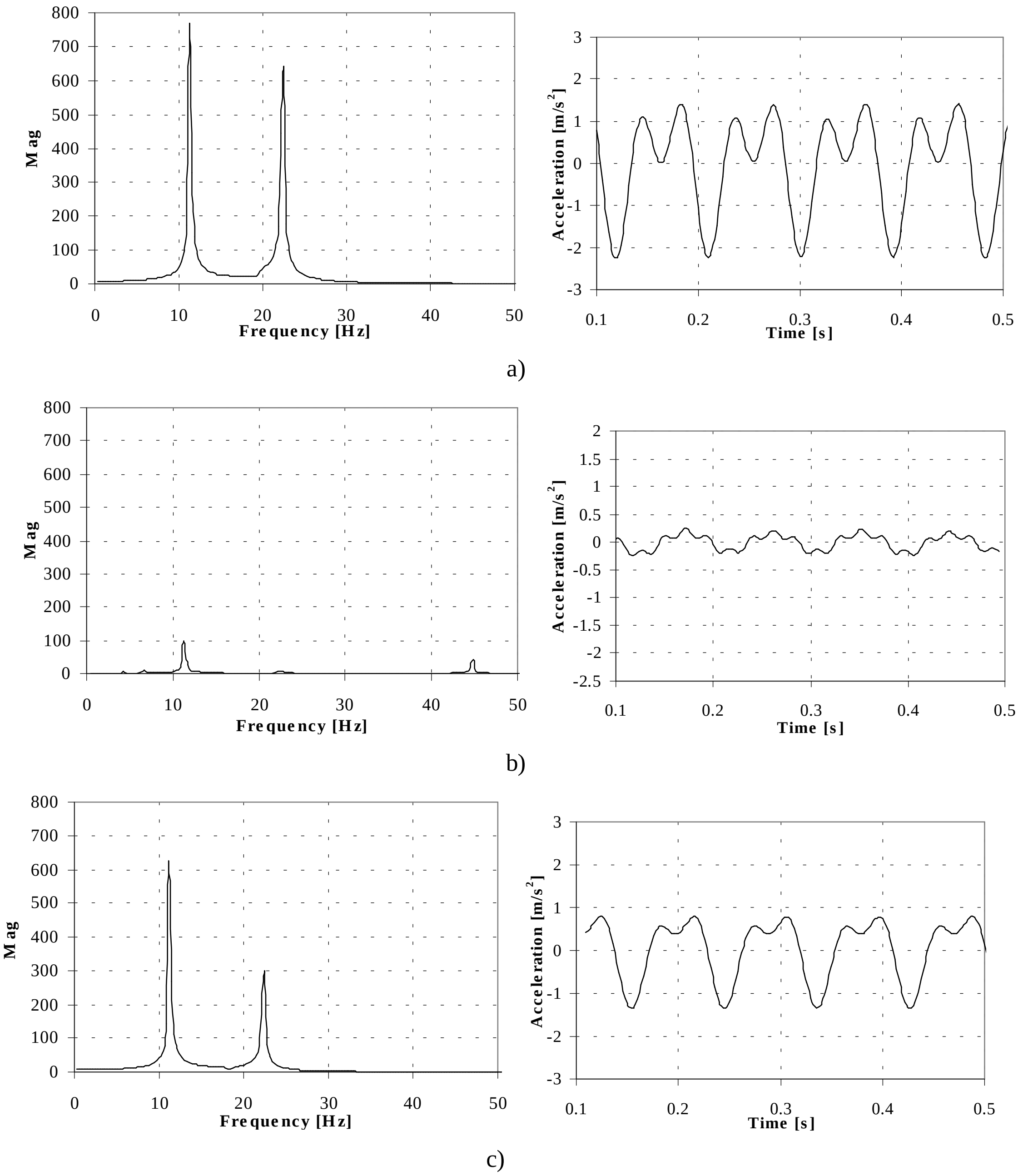


Figure 6. Numerical results. Point 2. Spectrum and time history. a) Assembly without vibration control. b) Assembly with springs. c) Assembly with incorporated masses of 15%.

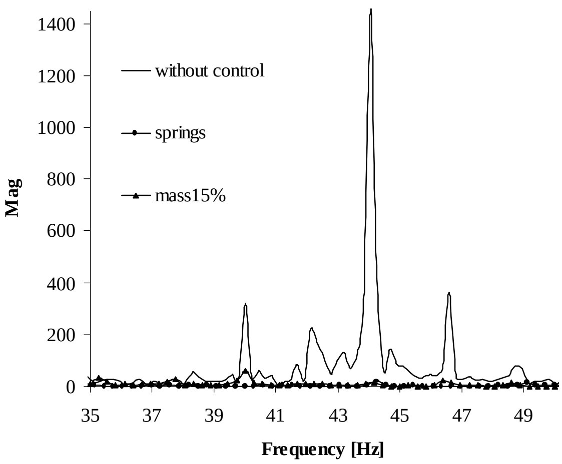
4.1.1. Spectrum

Because of the dynamic characteristics of the system and the excitation, the energy mainly is transmitted in the band of 44 Hz for point 1 and, in the band of 11 Hz and 22 Hz for point 2, only those frequency bands are shown.

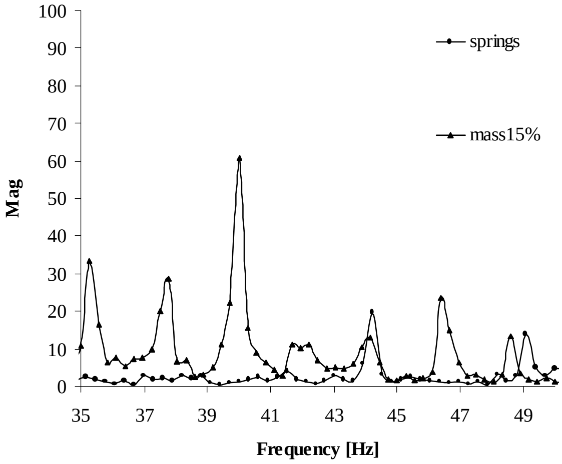
a) Point 1 (possible receiver of vibration)

Figure 7a shows that in a possible receiver point (like point 1) it is possible to reduce the level of vibration in 98% for the main component (44 Hz) with both control systems, springs and built-in masses with 15% of the total mass of the machine. For the sake of clarity, Fig. 7b shows the comparison between the control vibration systems only.

It should be noted that, in point 1, the levels of vibration of the structure are completely similar with both control systems.



a)

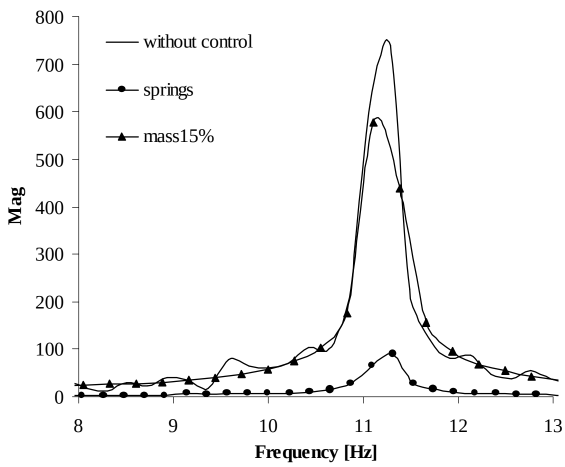


b)

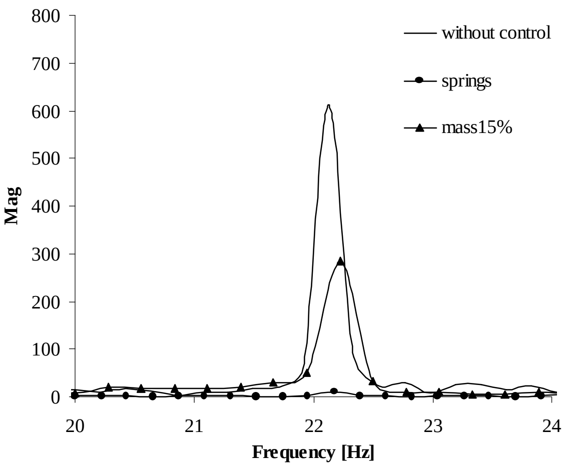
Figure 7. Efficiency of the vibration control systems. Point 1. a) Spectrum of the system with and without vibration control, b) Spectrum of system with vibration control, in each cases (springs and built-in masses).

b) Point 2 (underneath the machine).

Figure 8a shows that on the first component 11 Hz (mass of 15 %) an attenuation of 20% approximately is obtained. In Fig. 8b, a reduction of 60% on the second component 22 Hz is observed. Consequently, for both components with the built-in masses system, the levels of reduction reached by the system with springs cannot be obtained.



a)



b)

Figure 8. Efficiency of the vibration control systems. Point 2. a) First component, b) Second component.

4.1.2. Peak and R.M.S. values of accelerations

Table 1 shows the average of the peak acceleration values measured in the positive and negative phase and the reduction factor that is obtained in points 1 and 2 with each vibration control system. Table 2 shows R.M.S. acceleration values and reduction factors in points 1 and 2.

From the comparative analysis, the following observations can be made:

In the system that uses helical springs as well as the built-in masses of 15%, reductions in peak values of around 90% in a receiver possible point (point 1) are obtained.

In a point located underneath the machine (point 2), the reduction reaches 88% for the system with springs and 40% for which it incorporates masses. Something similar happens to R.M.S. values.

These results show that, in a possible receiver point, the proposed system gets efficiency of the same order that those achieved with conventional systems. Therefore, it is a right choice in those cases where the traditional systems have not application because of the conditions of the problem. However, in problems with high levels of vibration in locations closer to the exciting machine, the traditional system is recommended.

Table 1. Peak acceleration values (absolute) and Reduction Factors.

Mounting	Point 1		Point 2	
	peak value [m/s^2]	reduc. factor [%]	peak value [m/s^2]	reduc. factor [%]
Without vibration control	3.080	ref.	2.490	ref.
Helical springs	0.196	94	0.300	88
Incorporated masses (15%)	0.420	86	1.500	40

Table 2. RMS acceleration values and Reduction Factors.

Mounting	Point 1		Point 2	
	rms value [m/s^2]	reduc. factor [%]	rms value [m/s^2]	reduc. factor [%]
Without vibration control	1.210	ref.	1.040	ref.
Helical springs	0.300	75	0.120	88
Incorporated masses (15%)	0.180	85	0.710	32

4.3. Numerical-experimental comparison.

With the objective of verifying the methodology implemented in the design (to be used as reference in future projects) a numerical modelling was carried out. A comparison between theoretical and experimental results is made. The response is shown in terms of time history of accelerations and frequency Spectrum for the system without vibration control and those with vibration control using springs and built-in masses with 15% of the total mass of the machine.

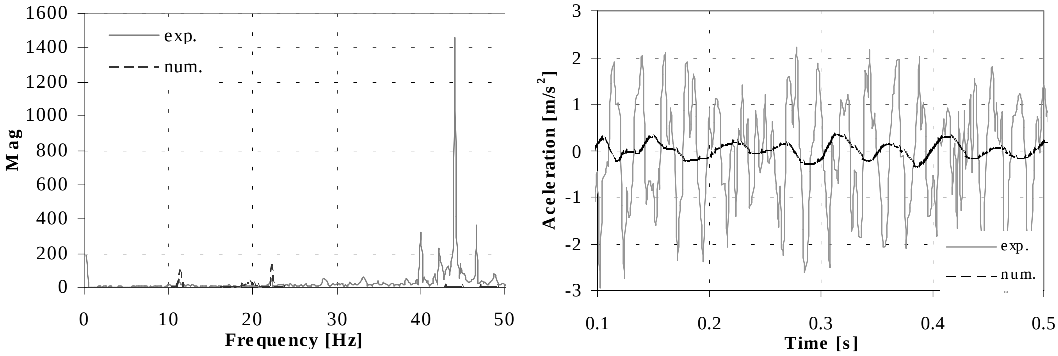
4.3.1. System without vibration control

a) Point 1

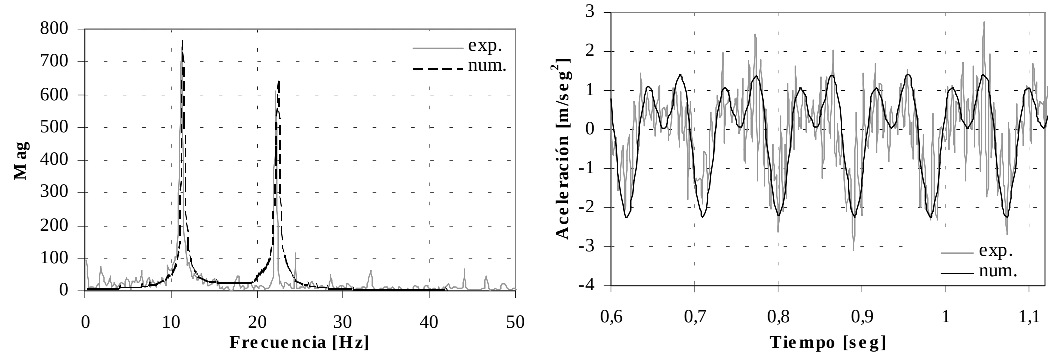
The impossibility to reproduce, in the numerical analysis, the main component in approximately 44 Hz is observed in Fig. 9a, consequently in the time history, the measured amplitude is not reached.

b) Point 2

From Fig. 9b it is observed that the numerical model reproduces accurately the response of the measured system, as much in amplitude of time history as in the main frequency bands of the spectrum. Obviously, insignificant amplitude of the harmonicas of superior order does not appear in the numerical analysis. This is because in the numerical model the external load with just two components (11 Hz and 22 Hz), was considered.



a)



b)

Figure 9. Numerical-Experimental comparison. Without vibration control. Spectrum and time history. a) Point 1. b) Point 2.

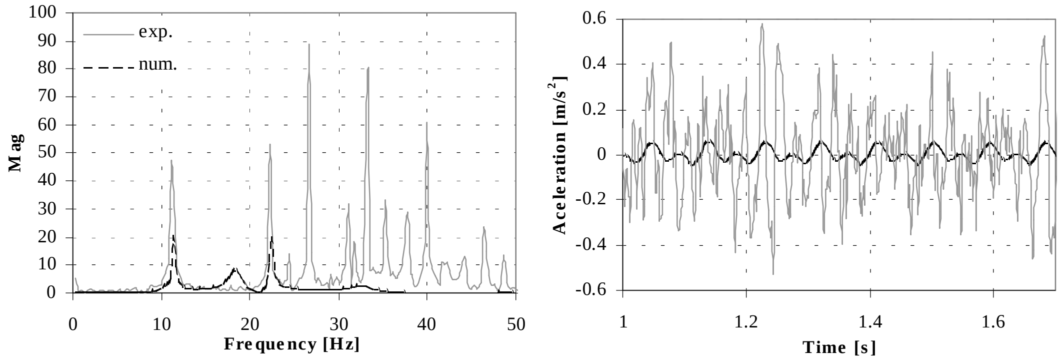
4.3.2. System with vibration control

a) Point 1

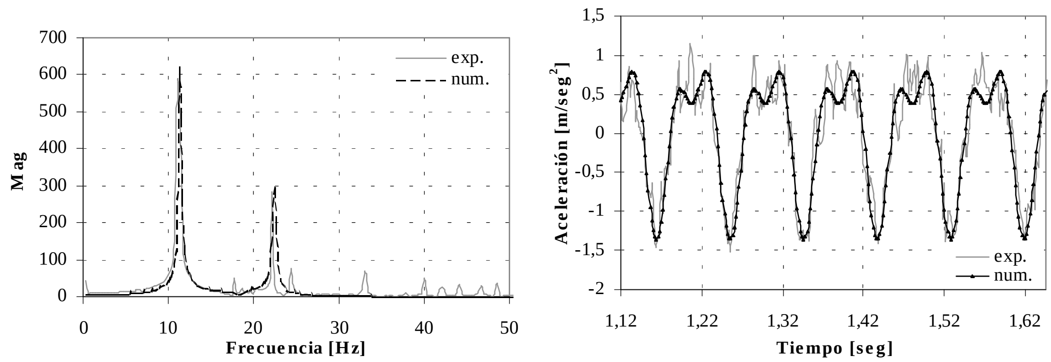
Figure 10a shows that in the case of built-in mass, the numerical model is unsatisfactory. It is evident that it is not possible to reproduce numerically the resonant response in this point, due to the proximity existing between the frequency of the second mode of vibration of the system (approximately 44 Hz) and the corresponding component of the dynamic load, neglected in the model. Consequently, in the spectrum obtained from numerical analysis for this point, small amplitude in the frequency interval of 44 Hz, is observed. This fact is because in the load model only the fundamental component of 11 Hz and the second component of 22 Hz were considered. In absence of superior order components, from numerical simulation the recorded phenomenon becomes difficult to reproduce.

b) Point 2

Figure 10b shows that with the numerical model it is possible to reproduce accurately the measured system response with vibration control in places next to the exciting machine. Moreover, the small amplitudes of high frequency, not important to the control vibration system design, are difficult to reproduce.



a)



b)

Figure 10. Numerical-experimental comparison. Assembly with incorporated masses of 15%. Spectrum and time history. a) Point 1. b) Point 2

5. Conclusions

A numerical-experimental study with the objective of determining the efficiency of the proposed vibration control system has been addressed. On the basis of the results obtained, the following conclusions may be drawn:

1. The proposed vibration control system presents a level of efficiency similar to a traditional one when a possible receiver is far from the exciting machine. Consequently, this alternative is able to replace the traditional system in those cases where its application becomes difficult.
2. When high levels of vibration in places near the vibration source exist, traditional systems of vibration control like those using helical springs (isolation of the vibration source) are recommended.
3. For the case of rotating machines, it is important the accurate incorporation of the dynamic load, mainly in the number of harmonic components that must be considered in analytical models. In many manuals and books of design it is suggested to use only the fundamental harmonica or the two first components. Nevertheless, this criterion would have to be reviewed. Several times, because of the proximity between frequency of some components of the dynamic load and the natural frequency of the support structure or critic parts of it, the resonance phenomenon is developed, defining the magnitude of the vibrations in the point. Therefore, a detailed analysis evaluating the natural frequencies of the structure as a whole and the control of vibration system is necessary to determine the number of harmonicas that in the excitation must be included.
4. With a relatively simple numerical model where, all the constructive details are considered, the behaviour of the dynamic system with sufficient approach could be predicted. Thus, with a series of simulations it is possible to determinate the optimal location and size of mass for the proposed vibration control system.

Acknowledgements

The authors wish to thank the help received from technical staff, Mr. Eduardo Batalla and Mr. Daniel Torielli, during the development and preparation of the tests. Moreover, the financial support of CONICET and the National University of Tucumán is gratefully acknowledged.

References

Arpaci, A., 1996, "Annular Plate Dampers Attached to Continuous Systems", Journal of Sound and Vibration, 191, 781-793.

Cui, D. and Craighead, I.A., 2000, "Reduction in Machine Noise and Vibration Levels Based on the Statistical Energy Analysis Method", Proceedings of the Institution of Mechanical Engineers, 214, 147-155.

Curadelli, R.O., 1999, "Estudio Numérico-Experimental de la Eficiencia de Sistemas de Control de Vibraciones en Máquinas", Master thesis, National University of Tucumán, Argentina. (in spanish).

Den Hartog, J.P., 1985, "Mechanical Vibrations", McGraw-hill, Inc, New York.

Feng, N.S. and Hahn, E.J., 1995, " Including Foundation Effects on the Vibration behaviour of Rotating Machinery", Mechanical Systems and Signal Processing, 9, 243-256.

Helsel, R., 1998, HP VEE . Visual Programing. Hewlett Packard Company.

Li, D.B. and Yam, L.H., 2000. "Modal Synthesis Method for Vibration Isolation Design of Massive Rotating Machines Resiliently Supported by an Elastic Structure", Journal of Sound and Vibration, Letters to the Editor, 231, 233-245.

Macinante, J., 1984, "Seismic Mountings for Vibration Isolation", John Wiley and Sons, N.Y. Chichester Bribane Toronto Singapore.

Mc Millan, A.J. and Keane, A.J., 1996, "Shifting Resonances from a Frequency Band by Applying Concentrated Masses to a Thin Rectangular Plate", Journal of Sound and Vibration, 192, 549-562.

Mc Millan, A.J. and Keane, A.J., 1997, "Vibration Isolation in a Thin Rectangular Plate Using a Large Number of Optimally Positioned Point Masses", Journal of Sound and Vibration, 202, 219-234.

Natsiavas, S. and Tratskas, P., 1996, "On Vibration Isolation of Mechanical Systems with Non-Linear Foundations", Journal of Sound and Vibration, 194, 173-185.

Richart, F.E., Hall, J.R., and Woods, R.D., 1970, "Vibration of Soils and Foundations", Prentice-Hall, Englewood Cliffs, N. J.

Wilson, E.L., 1997, Sap 2000-Integrated Finite Element Analysis and Desing of Structures. Computer and Structures, Inc. Berkeley, Ca.

Yi, D.,Dietz, P. and Hu. X., 1998, "Vibrational Power Flow in Beam-Plate Structures with Isolation Components", Proceedings of the Institution of Mechanical Engineers, 212, 89-101.

Zapfe, J.A. and Lesieutre, G.A., 1996, "Broadband Vibration Damping Using Highly Distributed Tuned Mass Absorbers", AIAA Journal, Technical Notes, 35, 753-755.