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Letter to the Editor

Vibration control by attaching masses to a plate excited by rotating machinery

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1. Introduction

Many researchers have considered the topic of vibration control in the last century but Ormondroyd and Den Hartog [1] published the first mathematical treatment of passive systems of vibration control in 1928. The above-mentioned paper can be considered as the beginning of the systematic treatment of the problem and many works have appeared since then, mainly in the analysis of base isolation systems, which are summarized in many books [2–4].

In the last years, advances have been made, especially in theoretical procedures to consider the problem of vibration behaviour and machines control [5–11]. However, according to Arpaci [12], since most of the investigations on the absorbers are theoretical, special attention must be paid to the need of conducting 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 springs or viscoelastic materials. However, in many cases of rotating machines supported on plate structures it would not be possible or convenient to use this alternative of isolation. For example, consider machines not initially isolated that for any change in the support structure or the operating conditions produce unexpected high levels of vibrations. Moreover, 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 main objective of this paper is to compare the efficiency of vibration control systems of rotating machinery, in order to present alternatives for traditional systems, mainly based on base isolation by using helical spring or viscoelastic materials. According to this objective, 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%

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of the total mass of the vibrating machinery are included in the analysis. A set of tests concerning a case study on the spring (traditional) and masses (alternative) vibration control systems is presented. The acceleration time histories in different points are recorded. On the other hand, a numerical analysis using a finite element program [13] was carried out.

Related to this research, Zapfe et al. [14] show that using a large number of inertial absorbers distributed in space and in frequency, a broadband damper can be designed to provide energy dissipation over a specific frequency band that is immune to the adverse effects of in-plane tensile load. McMillan and Keane [15,16] presented an approach to make minor modifications to the structure which are sufficient to change the frequency response of a narrow plate so that the transmission of vibrations in a given frequency band is suppressed. In this paper, an attractive alternative using masses incorporated into the support structure to mitigate vibration is proposed. However it is important to note that because of the different applications analyzed, McMillan and Keane [15,16] loaded the plate with a number of small point masses in variable positions, in contrast to the method described here which uses only one or two masses located at the antinodes of the first flexural modes.

Moreover, other alternatives can be found in the literature as the incorporation of viscoelastic materials in suitable arrangements [17], an annular plate attached to a main system [12] and a semi-active dry friction device [18]. On the other hand, a physically similar problem is the slab vibration of buildings due to the movement of people on them, causing human discomfort [19]. The solutions proposed in this paper are also applicable to this type of problem.

2. Experimental study

2.1. System description

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

Initially, in order to evaluate the severity of the excitation, the machine was mounted rigidly on the support structure as shown in Fig. 1(a). Then, two vibration control systems were employed:

System 1: traditional, base isolation with helical springs.

An adequate tuning ratio of the helical springs was selected so that the fundamental frequency of the auxiliary system was sufficiently far from the excitation band of frequencies. Consequently, the stiffness of the springs is low.

System 2: alternative, built-in masses.

The masses are attached to the antinodes of the first two flexural vibration modes of the support structure (Points 1 and 2 in Fig. 1(b)). Varying the masses by steps of 5%, 10% and 15% of the total mass of the machine, the sensitivity in the efficiency of the vibration control system was studied.

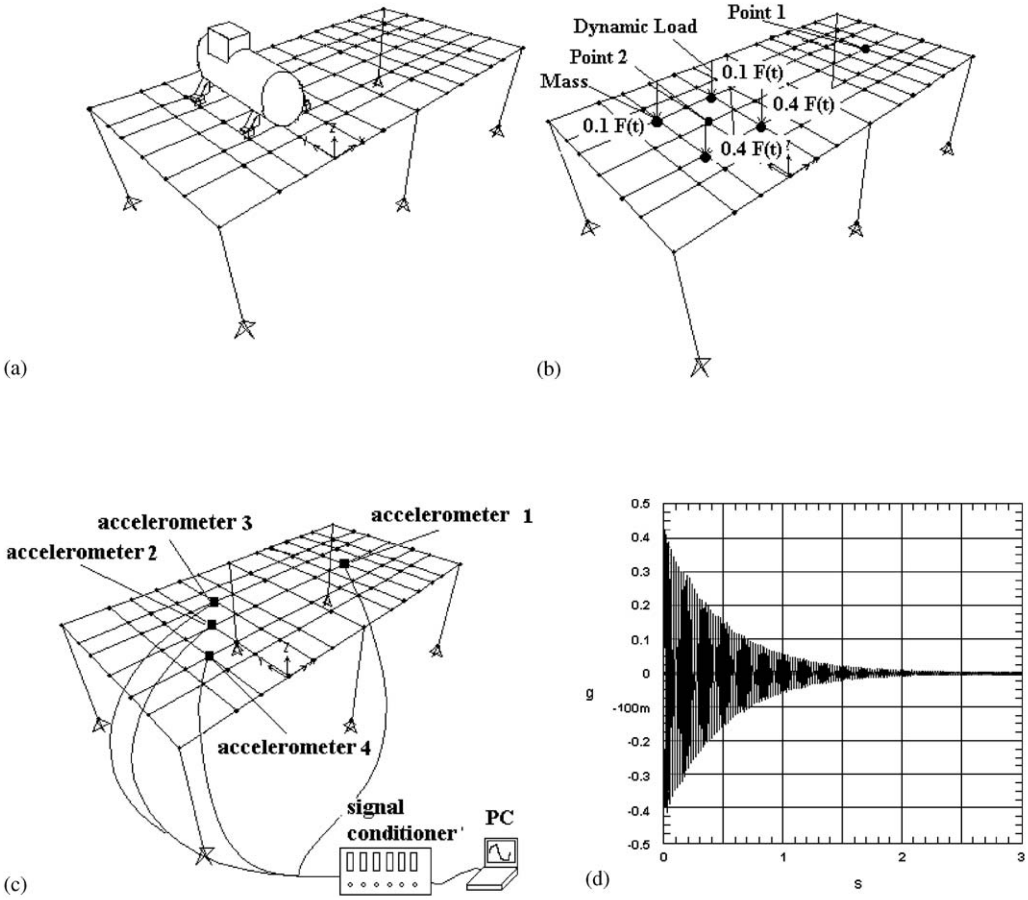

Fig. 1. Case study: (a) assembly of the exciting machine, (b) masses location and dynamic load distribution, (c) accelerometers disposition, (d) acceleration record.

2.2. Experimental set-up and free vibration response

For the three mentioned configurations, with the machine running, the vertical acceleration at different points on the structure (Fig. 1(c)) 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 Computerboards 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 [20]. A scheme of the equipment is presented in Fig. 1(c). Four channels were used in all the tests and the signals were sampled at 500 Hz each channel.

Free vibration tests were carried out and natural frequencies of vibration were determined for the first flexural modes, obtaining 19 and 44 Hz. Moreover, the equivalent viscous-damping ratio was estimated using the exponential decaying method [21]. Many determinations were performed on various tests in order to obtain more reliable results. The mean value obtained was 1.3% and the coefficient of variation was 23%. A typical free vibration test is shown in Fig. 1(d).

2.3. Dynamic response

In this section, the responses obtained in the forced vibration tests are shown for the three configurations mentioned above (rigid support, base isolation and built-in masses), measured at

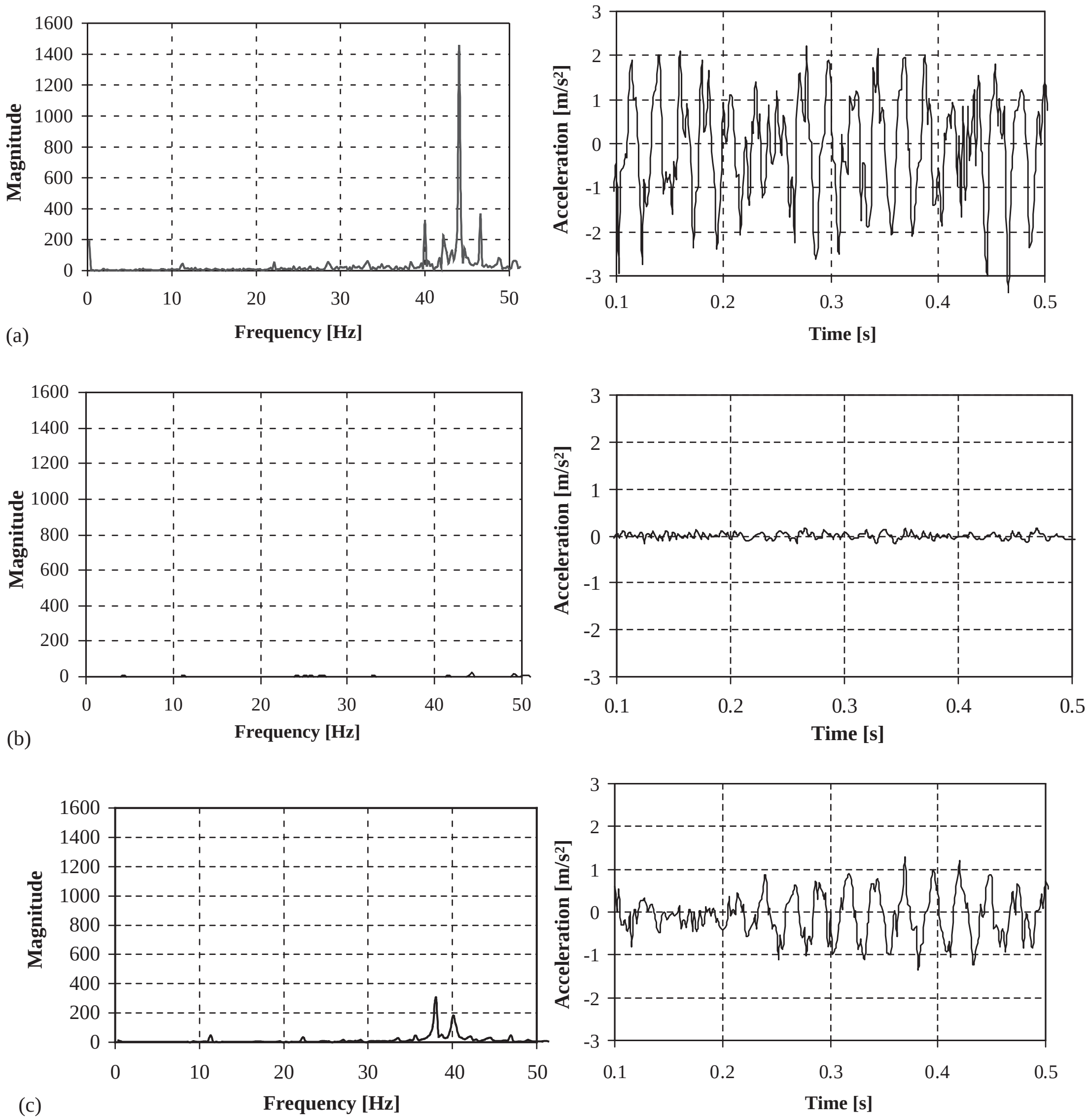

Fig. 2. Experimental results at Point 1. Frequency spectrum and time history: (a) assembly without vibration control, (b) assembly with springs, (c) assembly with incorporated masses of 5%, (d) assembly with incorporated masses of 10%, (e) assembly with incorporated masses of 15%.

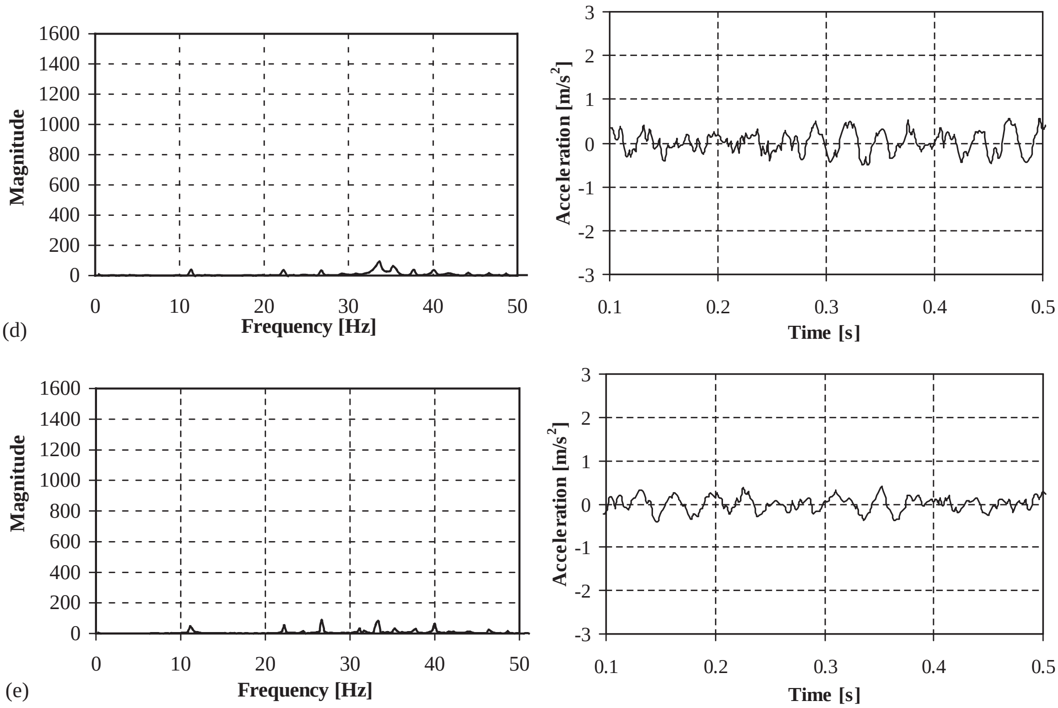

Fig. 2 (*continued*).

points 1 and 2 (see Fig. 1(b)). A resonant response could be expected at the second natural frequency (44 Hz) due to the fourth component of the dynamic load.

2.3.1. *Point 1*

A main component in approximately 44 Hz is observed in Fig. 2(a). This frequency is close to the second natural frequency of the support structure. In Fig. 2(b) the time history and the spectrum, of the system mounted on helical springs are shown. The remarkable reduction in the level of vibrations measured at this point is worth noticing. In Figs. 2(c–e) the responses to the 5%, 10% and 15% built-in masses systems are presented.

It is important to note that, with the proposed system, the responses obtained at point 1 of the structure show levels of vibrations reduction of the same order to those obtained by the traditional vibration control system.

2.3.2. *Point 2*

For the system without vibration control two peaks at 11 and 22 Hz appear in Fig. 3(a). These frequencies agree with the main components of the excitation load. The harmonics of 33 and 44 Hz are of minor importance. In Fig. 3(b) the large efficiency obtained with the traditional vibration control system at a measurement point underneath the machine is observed. The vibration levels were reduced to less than 10% of the original level. In the spectrum, only a small peak at 11 Hz appears that corresponds to the fundamental harmonic of the exciting load. Figs. 3(c–e) show the responses measured in the system with built-in masses for the three cases. A

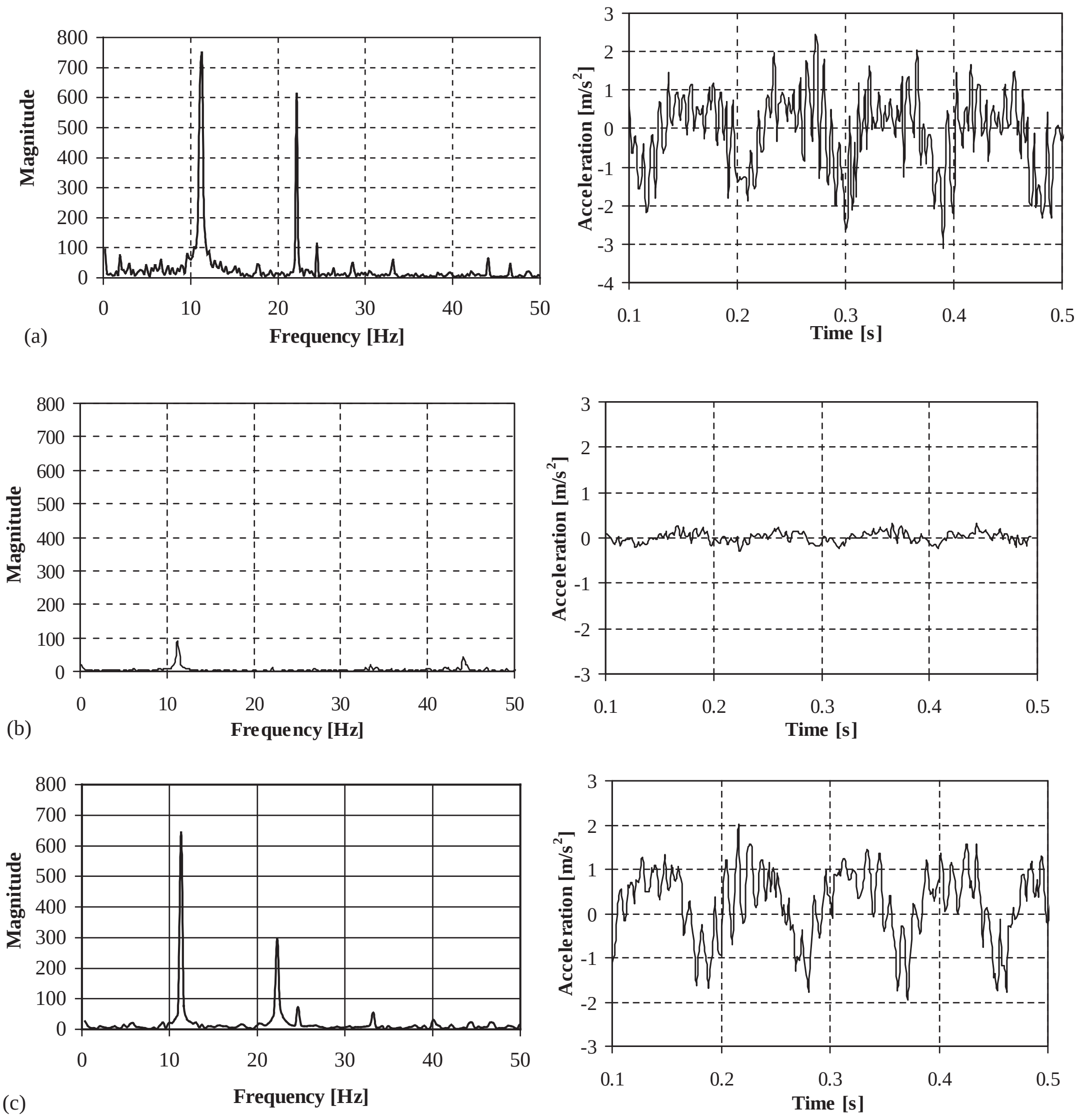

Fig. 3. Experimental results at Point 2. Frequency spectrum and time history: (a) assembly without vibration control, (b) assembly with springs, (c) assembly with incorporated masses of 5%, (d) assembly with incorporated masses of 10%, (e) assembly with incorporated masses of 15%.

progressive reduction in the vibration level is observed but of smaller order than the one obtained with traditional control system.

The progressive reduction in amplitude of the peaks should be noted as it increases the mass considered in the vibration control system, at both points of measurement, as Tables 1 and 2 show. The reduction factor is defined by the relation between the maximum peak acceleration of the records with and without vibration control system.

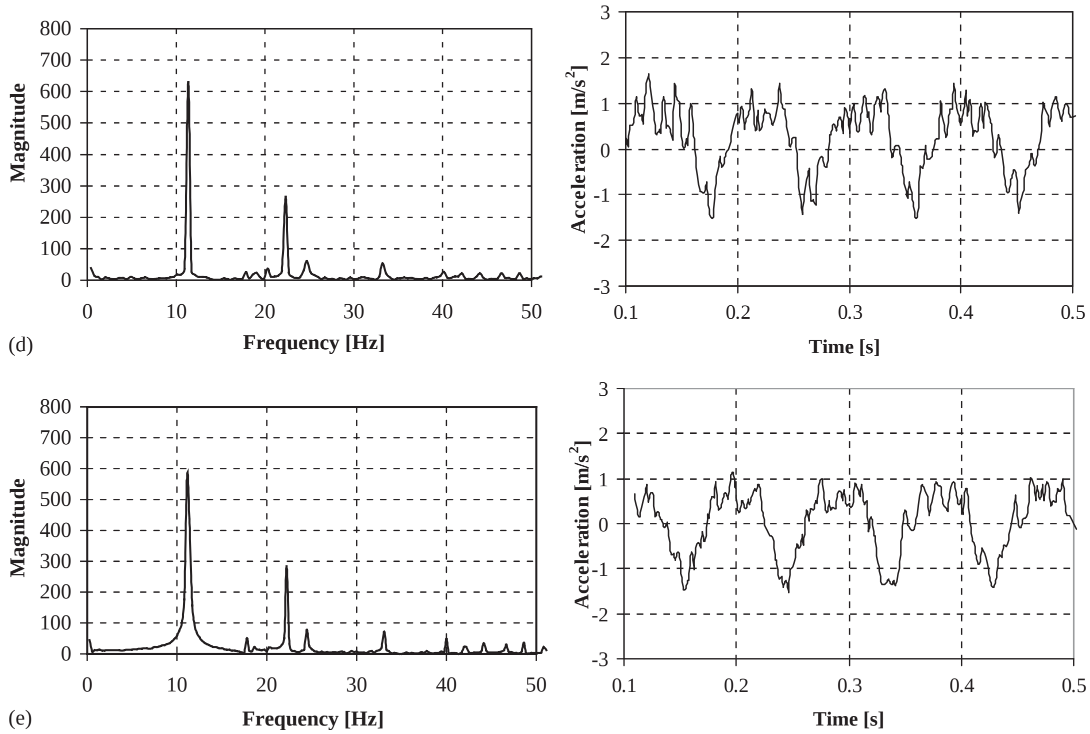

Fig. 3 (*continued*).

Table 1
Peak values of acceleration (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 (5%)	1.360	56	2.00	20
Incorporated masses (10%)	0.500	84	1.630	35
Incorporated masses (15%)	0.420	86	1.500	40

3. Numerical analysis

3.1. Model description

The behaviour of the complete system (structure-machine) was simulated in a finite element software [13]. The structure columns as well as the beams were modelled using frame elements and the plate using shell elements. In addition, concentrated masses were added to represent the mass

Table 2
r.m.s. values and reduction factors

Mounting	Point 1		Point 2	
	R.m.s. value (m/s^2)	Reduc. factor (%)	R.m.s. 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 (5%)	0.467	61	0.840	19
Incorporated masses (10%)	0.230	81	0.710	32
Incorporated masses (15%)	0.180	85	0.710	32

of the machine, located in nodes corresponding to the position of each one of the four legs of the machine (Fig. 1(b)).

Fig. 1(b) shows the mesh used in the 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. The dynamic loads produced by the compressor were determined and added in the computational model applying them on the nodes in correspondence with the legs of the compressor. These loads were distributed according to the machine asymmetry as Fig. 1(b) shows.

The helical springs were modelled using springs elements located under the masses of the compressor and the stiffness was determined experimentally. For the case of the built-in masses control system, two numerical models were analyzed. Initially, the masses were concentrated at the nodes that correspond to points 1 and 2 and later the masses were distributed in a 0.1×0.1 m area such as the real area of the masses. In the second model, a response approximately 11% greater than the first model was obtained for the 15% built-in masses system. Consequently, the distributed mass model was adopted because it represents the physical problem analyzed in a better way.

3.2. Dynamic response

Initially, the modal shapes and natural frequencies of vibration were determined. The values obtained for the first flexural modes were 19.2 and 44.2 Hz, which agree with those obtained experimentally. The equivalent viscous-damping ratio obtained in free vibration tests was used in the numerical analysis.

3.2.1. Point 1

For the case of the system without vibration control the response does not show the fundamental peak measured (Fig. 4(a)) at approximately 44 Hz. For the case of 15% built-in masses it does not reproduce all the peaks obtained in the measurement (Fig. 5(a)).

3.2.2. Point 2

For the case of the system without vibration control the time history and spectrum measured (Fig. 4(b)) are reproduced accurately. In the same way, a good numerical approximation of the experimental response is observed for the case of the built-in masses system (Fig. 5(b)).

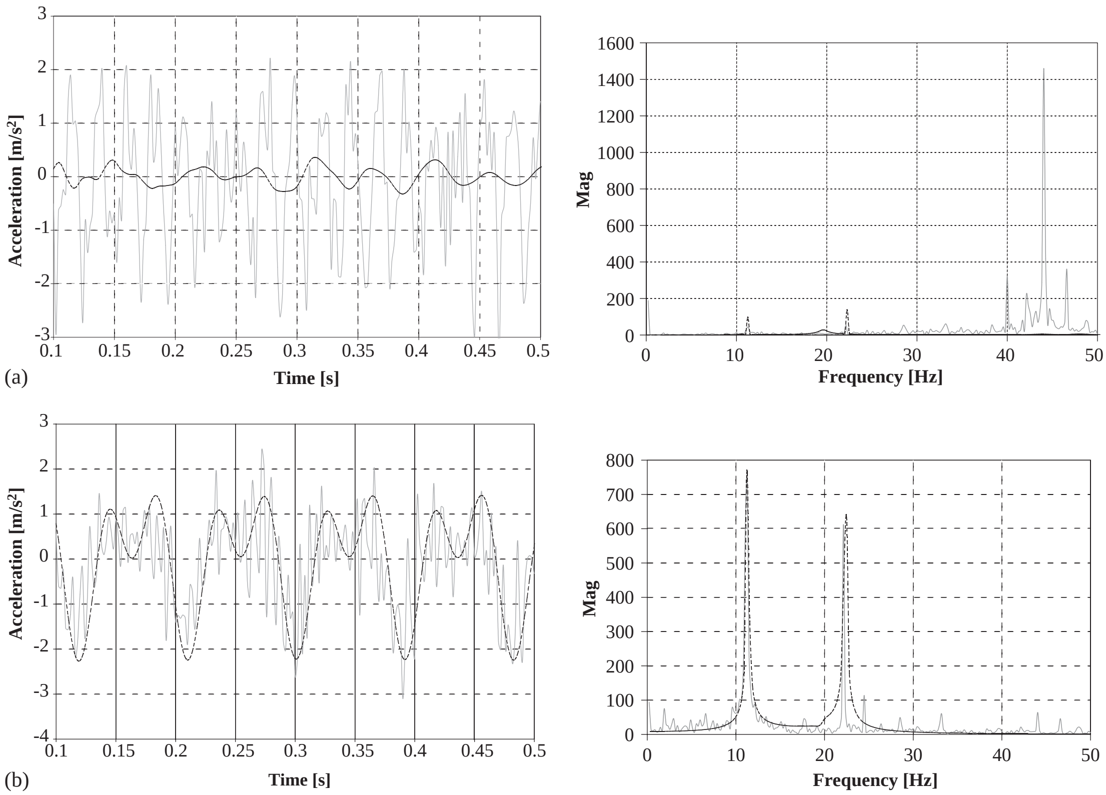

Fig. 4. Numerical–experimental comparison, without vibration control. Time history and frequency spectrum: (a) Point 1, (b) Point 2; —— , experimental; - - -, numerical.

4. Comparison and discussion

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

4.1. Numerical–experimental comparison

With the objective of verifying the methodology implemented in the design as well as the numerical model, a comparison between theoretical and experimental results was made. For space reasons, only the response for the system without vibration control and those with vibration control using springs and 15% built-in masses is shown.

4.1.1. System without vibration control

At point 1, according to the frequency components used in the dynamic load model, it is impossible to reproduce the main component at 44 Hz in the numerical analysis. Consequently, in the time history, the measured amplitude is not reached (Fig. 4(a)). From Fig. 4(b) it is observed

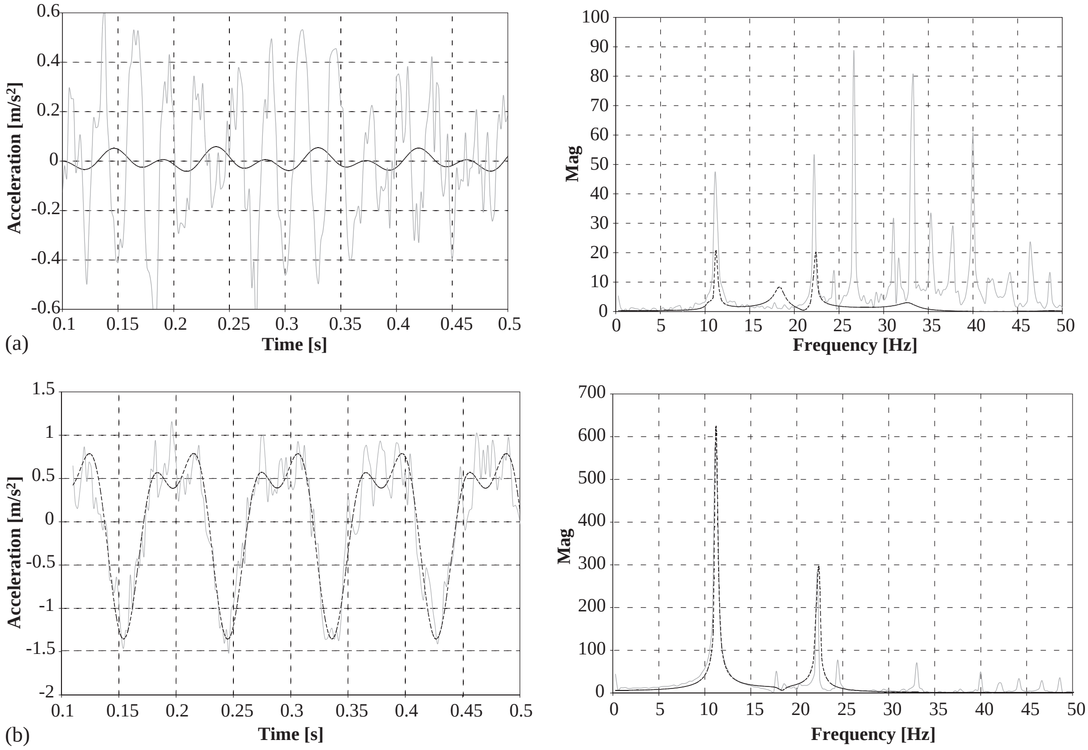

Fig. 5. Numerical–experimental comparison, assembly with incorporated masses of 15%. Time history and frequency spectrum: (a) Point 1, (b) Point 2; ——, experimental; - - -, numerical.

that the numerical model reproduces accurately the response of the measured system at point 2, as much in amplitude of time history as in the main frequency bands of the spectrum. Obviously, insignificant amplitudes of the harmonics of higher order do not appear in the numerical analysis. This is because, in the numerical model, only two components at 11 and 22 Hz represented the external load.

4.1.2. System with vibration control

For point 1, Fig. 5(a) shows that in the case of built-in masses, the numerical model is unsatisfactory. It is evident that it is not possible to reproduce numerically the resonant response at this point, due to the proximity existing between the frequency of the second vibration mode of the system (approximately 44 Hz) and the fourth component of the dynamic load, neglected in the model. Consequently, in the spectrum obtained numerically, smaller amplitudes around 44 Hz are observed. In the same way as in the system without vibration control, only the load components of 11 and 22 Hz were considered in the numerical model. In absence of higher order terms, the recorded phenomenon becomes impossible to reproduce.

This problem of resonant response can also be observed between the natural frequencies of slabs of buildings and the frequencies of people's movement [19].

For point 2, Fig. 5(b) shows that with the numerical model it is possible to reproduce the measured response of the system with vibration control in locations close to the exciting machine accurately. Moreover, small amplitude and high frequency peaks, not important to the control vibration system design, are difficult to reproduce.

4.2. 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. acceleration values.

4.2.1. Spectrum

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

(1) *Point 1* (*possible vibration receiver*)

In point 1 it is possible to reduce the vibration level by 98% for the main component (44 Hz) with both control systems, springs and 15% built-in masses. Fig. 6(a) shows the comparison between the vibration control systems.

It should be noted that the vibration levels of the structure are completely similar to point 1 with the control system that uses springs and with 10% and 15% built-in masses.

(2) *Point 2* (*underneath the machine*)

Fig. 6(b) shows that for the first component of 11 Hz (mass of 15%) around 20% attenuation is obtained. In Fig. 6(c), a 60% reduction on the second component of 22 Hz is observed. Consequently, for both components with the built-in masses system, the reduction levels reached by the system with springs cannot be obtained.

4.2.2. Peak and r.m.s. acceleration values

Table 1 shows the mean of the peak acceleration values measured in the positive and negative phase and the reduction factor that is obtained at points 1 and 2 for both vibration control systems. Table 2 shows r.m.s. acceleration values and reduction factors in the same points.

From the comparative analysis, the following observations can be made. In the system that uses helical springs as well as 10% and 15% built-in masses, reductions around 84% at a possible receiver point (point 1) are obtained. At a point located underneath the machine (point 2), the reduction reaches 88% for the system with springs and 36% for 15% built-in masses.

These results show that, at a possible receiver point, the proposed system has efficiencies of the same order as those achieved with conventional systems. Therefore, it is a good choice in those cases where the traditional systems cannot be applied because of the conditions of the problem. However, in problems with high vibration levels in locations close to the exciting machine as well as when the masses cannot be attached, the traditional system is recommended.

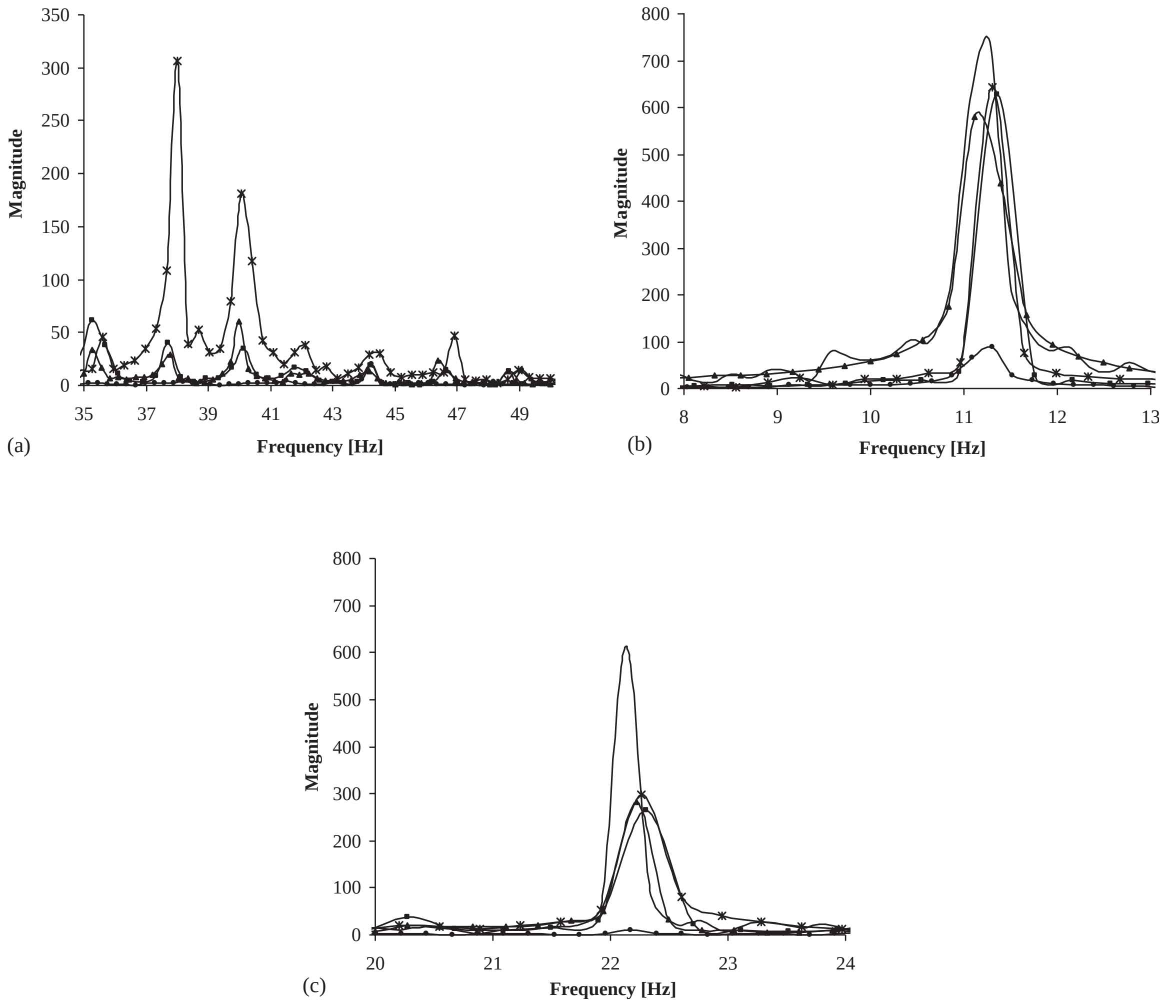

Fig. 6. Efficiency of the vibration control systems: (a) Point 1, vibration control systems only. (b) Point 2, first component. (c) Point 2, second component. Key: —— , without control; –●–, springs; –▲–, mass 15%; –■–, mass 10%; –∗–, mass 5%.

5. Conclusions

A numerical–experimental study with the objective of determining the efficiency of the attached masses vibration control system has been addressed. Moreover, experimental data that could be used for checking the accuracy of a variety of calculation methods are provided.

The built-in masses vibration control system presents an efficiency level similar to base isolation systems at points far from the exciting machine. In this case the alternative method is able to replace the traditional system in those cases where its application becomes difficult and where it is possible to attach the masses to the plate. According to Tables 1 and 2, it is enough to use 10% of the total mass of the machine as additional mass in the supporting structure to obtain a significant level of vibrations reduction at distant points. When high levels of vibration exist in places near the vibration source, base isolation systems of vibration control are recommended.

In the numerical analysis of rotating machines, it is important to incorporate the dynamic load accurately, mainly in the number of harmonic components that must be considered. Sometimes, because of the proximity between the frequency of some load components and the natural frequency of the support structure or critical parts of it, the resonance phenomenon results in an increase in the dynamic amplification factor. Therefore, a detailed analysis evaluating the natural frequencies of the structure and the machine as a whole is necessary in order to determine the number of harmonics that the excitation must include. Moreover, in the numerical model it is important to distribute the concentrated masses over the real area of the masses incorporated.

The alternative system of built-in masses can be used in the control of slab vibration due to the movement of persons on the slabs when other alternatives such as tuned mass dampers [19] are aesthetically or architecturally non-viable.

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