Volume 11, Number 7 (February 2018) p. 95 – 114 • ISSN 1983-4195
http://dx.doi.org/10.1590/S1983-41952018000100006

IBRACON

REVISTA IBRACON DE ESTRUTURAS E MATERIAIS
IBRACON STRUCTURES AND MATERIALS JOURNAL

Attenuation of sound radiation in concrete structure through the reduction of mechanical vibration

Atenuação de radiação sonora em estrutura de concreto através da redução de vibração mecânica

R. V. DE HOLANDA [a]
ricardo.holanda@ifg.edu.br

M. A. V. DUARTE [b]
mvduarte@mecanica.ufu.br

M. A. B. C. BADAN [a]
mabcbadan@gmail.com

J. L. DE O. PENA [a]
pena.jlo@gmail.com

R. C. ROSA [a]
engcanedorosa@gmail.com

Abstract

The efficiency of sound irradiance in structure has direct relation with its vibratory movement. Dynamic vibration absorbers (DVAs) are a low cost viable option for reducing vibrations in passive structures. Secondary systems attached to the primary system (structure) in order to reduce vibration. In this work it was used an experimental modal analysis procedure (EMA) for vibratory responses through impulsive excitations to determine the natural frequencies and the location of points suitable for attachment of DVAs in a concrete beam. Later it was designed and built DVAs to reduce vibration in a frequency band where the response of the human auditory system is more sensitive. The best project configuration for DVAs was evaluated for sensitivity thereof with respect to the change of the loss factor of the viscoelastic material used. Obtained reduction of more than 36% over the considered frequency band and over 70% in the region of the resonance frequency in which the DVAs were tuned.

Keywords: dynamic vibration absorber, experimental modal analysis.

Resumo

A eficiência da irradiação sonora em estrutura possui relação direta com seu movimento vibratório. Absorvedores dinâmicos de vibrações (ADVs) são uma opção viável de baixo custo para redução de vibrações em estruturas de forma passiva. São sistemas secundários fixados ao sistema primário (estrutura) com propósito de reduzir as vibrações do sistema primário. Neste trabalho foi utilizado um procedimento de Análise Modal Experimental (AME) para obter experimentalmente as frequências naturais e a localização dos pontos adequados para fixação dos ADVs em uma viga de concreto. Posteriormente, foram projetados e construídos ADVs para redução de vibração em uma banda de frequência onde a resposta do sistema auditivo humano é mais sensível. A melhor configuração de projeto para os ADVs foi avaliada quanto à sensibilidade dos mesmos com relação à alteração do fator de perda do material viscoelástico utilizado. Obteve-se redução superior a 36% ao longo da banda de frequência considerada, e superior a 70 % na região da frequência de ressonância na qual os ADVs estavam sintonizados.

Palavras-chave: absorvedor dinâmico de vibração, análise modal experimental.

[a] *Instituto Federal de Goiás, Campus Goiânia, Departamento de Engenharia Mecânica, Goiânia, GO, Brasil;*
[b] *Universidade Federal de Uberlândia, Faculdade de Engenharia Mecânica, Uberlândia, MG, Brasil.*

Received: 06 Jul 2016 • Accepted: 22 May 2017 • Available Online: 15 Feb 2018

1. Introduction

The construction industry has reduced its costs through the adoption of some principles of management and standardization of components and processes, allied to the use of calculation procedures, where structural parameters that reproduce more accurately the loads, the characteristics of the materials and processes employed, resulting in lighter and leaner structures were used.
Concrete structures made according to these principles are an important factor in the technological evolution, as it entails reducing costs, due to the shorter execution time, lower volume of materials used and improvement of environmental indicators due to less generation of debris. However, the lower volume of material, leads to lower rates of vibration absorption which has caused acoustic discomfort to the residents of the buildings. Batista; Varela [1] observed the existence of high levels of vibration in slabs of residential buildings, causing acoustic discomfort to the residents, in spite of the rules and design criteria being obeyed.
All types of buildings, regardless of their intended use, whether public, such as hospitals, schools, hotels, etc., or private such as multifamily residential buildings expose their occupants to the harmful effects caused by noise. These sounds have diverse origins such as: footsteps, voices, hydro sanitary facilities, impact of objects on the slab, televisions, sound equipment, among others.
Researches carried out in 110 apartments in the Goiânia region obtained the following results in relation to the acoustic comfort of the dwellings: 73.40% fall into the classification as very bad, bad or regular and 26.60% were considered as good (MARTINS; SAHB; NETO , [11]).
Impacts on floor slabs of buildings are transmitted via structure in function of the inherent vibratory processes. Studies showed that in traditional construction systems, the indirect transmission of sidewall noise through vibration is responsible for approximately 50% of the sound transmission between any two environments (RINDEL, [15], NUNES; DUARTE, [13]). The effects of dynamic floor excitations are not limited to the receiver environments located immediately below the source. For example, in gymnasiums, with an aerobic character, the induced vibrations can be perceived laterally to a distance of 30m in the same slab, or to 10 pavements below the source (LONG, [9]). This occurs because the slab becomes a sound energy irradiator over a wide range of frequencies, due to the vibratory movement induced by the localized excitation of the impact type (BISTAFA, [2]).
For source-ground contact noise, any solution that reduces structural vibration levels will result in attenuation of the noise transmitted to other environments.
To reduce impact noise, there are many different techniques, all varying in price and material. The most traditional solution is the use of viscoelastic materials between the floor and the subfloor and taking care of the noise transmission by the flanks (NUNES; DUARTE, [14]). An interesting option for new constructions and buildings in use could be the use of ADVs.
ADV is a secondary system added to a primary structural system and can act on any frequency whose vibration or radiated noise is to be controlled. ADV control has high robustness (its action is a consequence of the natural mechanical interaction of physical components), low cost and in its construction materials which allow the design of models with different geometries that can be incorporated into residential buildings environments ([5]) are used.
In his PhD thesis, Holanda [6] presents a pioneering methodology of ADV projects aiming at the reduction of impact noise in slabs. Simulations with a room modeled via finite elements resulted in attenuations of the order of 16 dB in impact noise.
For the experimental validation of the developed methodology, several absorbers were designed and applied in a bi-supported concrete beam. All tuned to frequencies where the level of audibility is most significant for the human auditory system, even when the sound pressure level is not high.
First, two absorbers with different loss factors were designed and constructed to experimentally evaluate the influence of the loss factor on the performance of the absorbers in impact noise reduction.
After the loss factor sensitivity assessment, three dynamic vibration absorbers were constructed, having as their design parameters the loss factor and three natural frequencies associated to the vibration modes of the reinforced concrete beam, estimated through experimental modal analysis (EMA). Then tests were carried out to validate the methodology proposed by Holanda [6].

2. Theoretical aspects

2.1 Vibration control

There are three ways to reduce vibration levels:
I. Acting on the excitation force by eliminating it, reducing its amplitude and / or altering its frequency;
II. Changing the structure by varying its dynamic characteristics (mass, rigidity and damping);
III. Adding an auxiliary system to the structure in order to eliminate or reduce vibration and its effects.

For example, from a suspension bridge of a few miles in length (Millau, France) to cooling ducts where high speed gases pass, there is a need to identify and interfere with the dynamic properties of either components or of assembled products, so as to ensure the structural integrity, durability and proper functioning of the equipment. The comfort of individuals who are directly involved in the operation and use of equipment in terms of exposure to acceptable noise levels, according to the legislation, can not be ignored.
It is hardly possible to act on the force of excitation, especially when one imagines that it can be the wind, or due to the impact of any object with random force and place of fall. Acting on the structure also becomes a complex problem in situations where it is a building, bridge or other large construction ready to use, or when it is symbolic and the appearance can not be changed for aesthetic reasons. Consequently, only the third hypothesis is feasible in a large number of applications where vibration reduction and even elimination, is desired.
The auxiliary system or secondary system coupled to the main system can be of two types according to the desired objective:
I. Type MK - spring mass: Den Hartog [4] has shown that the vibration amplitude of the primary system for a given natural frequency of interest tends to zero when the frequency of secondary system, known as Dynamic Vibration Neutralizer NDV (Dayou [3]), coincide with the frequency of vibrational excita-

tion source.The secondary system is attached at a determined point in the main structure where it is desired to eliminate the vibratory amplitude. Reaction forces are generated redistributing the vibratory energy, changing the structure response such that the original natural frequency of the primary system is eliminated. However it gives rise to two new natural frequencies around the extinct frequency.

II. Type MCK - with spring-damping mass: in addition to generating reaction forces at a certain point of the structure the viscous or hysteretic damping element promotes the dissipation of the vibratory energy, attenuating the amplitude of the new resonant frequencies. For this reason it is called Dynamic Vibration Absorber (ADV). The use of the damping element has the ability to extend the energy dissipating effect of the ADV over a frequency band around the tuning frequency.

MK-type NDVs have only elements that store kinetic energy and elastic potential, there being no type of vibratory energy dissipation, the control occurs through the equilibrium between the excitation and the reaction forces of the absorber. Due to this configuration two problems can occur: the vibration amplitude of the NDV becomes high, which can cause its fatigue rupture; And variations in the excitation frequency may cause the frequency de-tuning of the NDV to occur, which may increase the amplitude of vibration of the composite system due to a coincidence with the resonant frequency, leading to collapse of the structure. The solution for this type of occurrence is to introduce damping in the NDV, transforming it into an ADV.

2.2 Dynamic Vibration Absorber (ADV)

The theoretical spatial model of the simplest structural system has only one degree of freedom, consisting of a mass (m2) fixed/attached to a stiffening spring (k2) and a damper (c2), which may be viscous (figure [1a]) or hysteretic (figure [1b]). The model of two degrees of freedom (coordinates x1 and x2) moves as a function of the force F variants in time.

The physical damping models usually used in the dynamic modeling of ADVs are:

I. Viscous - which occurs between a solid part and a viscous fluid (lubricating oil, for example) interposed between moving parts of the mechanical system.

II. Hysteretic - which occurs by the internal friction between molecules when the solid is deformed, causing the energy to be dissipated by materials with viscoelastic characteristics.

Viscoelasticity is a type of mechanical behavior of certain materials possessing the property on being deformed, exhibit elastic behavior (tension proportional to the deformation), storing mechanical energy, and also viscous behavior (tension proportional to the velocity of deformation), dissipating energy in the form of heat. These materials have temperature-dependent mechanical properties (generally considered constant for simplification purposes) and vibration frequency and can be characterized by two transient properties: creep and relaxation. They are produced in the most diverse forms such as rubbers, resins, foams, enamels, acrylics and films. They have complex stiffness represented by Equation (1) NASHIF et al., 1985 [12]:

$$K_c(\Omega) = LG_c(\Omega) = LG(\Omega)[1 + i\eta(\Omega)] \tag{1}$$

Where:
- Is the complex stiffness;
- Is a form factor;
- Is the complex shear modulus of viscoelastic material;
- Is called the loss factor (PF).

The dimensionless parameter that characterizes the energy absorption in viscoelastic materials is denominated loss factor, being defined as the ratio between the viscous and elastic response of the materials. Materials with high loss factor (high mismatch between excitation and response) have a high viscous effect, dissipating more energy than materials having lower values for loss factors.

The reasons why viscoelastic material has been widely used in the construction of ADVs is that it has good resilience, great dissipation of energy and ease to be modelled. The energy-dissipating effect reduces the level of vibration in the ADV, as well as producing the effect of "spreading" the vibration absorption in frequencies close to the tuning frequency, making it more effective in a larger frequency band.

The efficiency of the vibration control on a structure depends on an adequate design of the ADV, with the specification of the param-

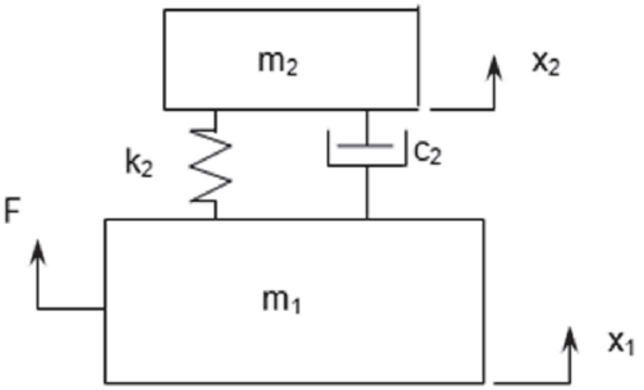

Figure 1a
Structural System (m1) fixed to an ADV with viscous damping

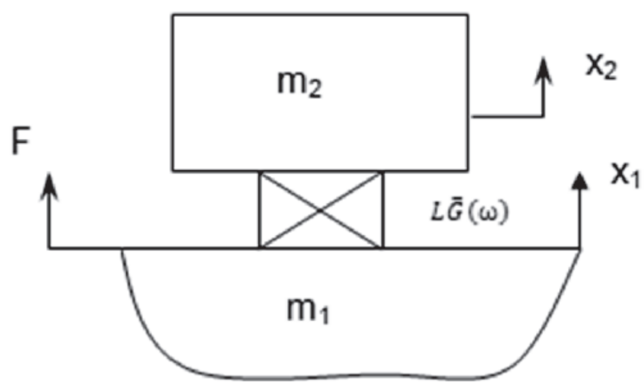

Figure 1b
Structural System (m1) fixed to an ADV with hysteretic or viscoelastic damping

eters of mass, damping and stiffness, as well as the determination of the location of its fixation/attachment in the main structure. Generally the optimum attachment point should be at the maximum modal amplitude and the opposite situation would be anti-node where if fixed would have no effective effect whatsoever.

The first mathematical model developed to design ADVs was presented by Ormondroyd and Den Hartog (1928). According to Silva et al. [17], these devices with viscous dampers are difficult to construct and are generally only used as a comparison of mathematical models in the study of vibration control.

Equation (2) describes the movement of the spatial model to a system with two or more degrees of freedom, as represented by Fig. [1a]:

$$[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(t)\} \quad (2)$$

Where [M], [C] and [K], are the mass, damping and rigidity matrices respectively; {F (t)} is the vector of the forces of excitation and x (t) is the vector of the displacements of the coordinates of interest. The modal matrix of an undamped system consists of all its eigenvectors. Using the mass matrix for normalization, we obtain the modal matrix [Φ].

$$[\Phi] = \begin{bmatrix} \phi_{11} & \phi_{12} \\ \phi_{21} & \phi_{22} \end{bmatrix} \quad (3)$$

where if fixed would have no effective effect whatsoever.

In which represents the element associated with the “n” mode and the “p” position.

For a harmonic excitation $f(t) = F_0.e^{\wedge(i\omega.t)}$ where ω is the frequency and F_0,is the amplitude, the system response can be represented by $x(t) = X.e^{\wedge(i(\omega.t+\theta))}$, where X is the amplitude of the displacement and θ is the phase between the excitation and the response.

Assuming proportional damping and employing the orthogonality properties of eingvectors result:

$$\Phi^T.[M].\Phi = [I] \quad (4.1)$$

$$\Phi^{T.}[C].\Phi = [c] \quad (4.2)$$

$$\Phi^T.[K].\Phi = [\omega_n^2] \quad (4.3)$$

Where: [I] is a unitary matrix and [c] and $[w_n]$ are the diagonal matrices of damping and eigenvalue respectively, with .

With this information, we can derive the matrix of receptance $[\alpha(\omega)]$, whose elements are defined by Eq. (5):

$$[\alpha(\omega)]_{pq} = \alpha(\omega)_{pq} = \sum_{n=1}^{N} \frac{\{\phi_n\}_{p.}\{\phi_n\}_q}{\omega_n^2 - \omega^2 + 2\xi_n\omega_n\omega i} \quad (5)$$

Where , is an element located in its line “p” and column “q”, representing the response (displacement) of the position “p”, x (t, x = p) due to the application of a force in position “q”, F (t, x = q).

Knowing that the relation between output and input of a linear system is given by transfer function H(w). We have $[\alpha(\omega)]_{pq} = [H(\omega)_{pq}$.

In 1968 Snowdon [18] constructed mathematical models which used a viscoelastic material of better constructional characteristics, represented by figure [1b], instead of spring and viscous element.

Soeiro [19] proposed a simplified viscoelastic model, which consists of a viscous damping model with the frequency-dependent damping constant (the term “constant” here refers only to time). It is assumed that this viscous damping constant is of the form:

$$c(\omega) = {}^{d(\omega)}/_{\omega} \quad (6)$$

Equation (6) is equivalent to the hysteretic damping model, solid or structural, where the parameter “d” is called the hysteretic damping coefficient.

The frequency response function for a system of one degree of freedom, with harmonically excited hysteretic damping, can be expressed by:

$$H(\omega) = \frac{X(\omega)}{F(\omega)} = \frac{1}{(-\omega^2 m + i\omega c + k)} = \frac{1/m}{\left(\omega_n^2 - \omega^2 + i\eta\omega_n^2\right)} \quad (7)$$

Where is known as loss factor (PF).

The dynamic behavior of a structure with any number of degrees of freedom can be characterized by its frequency response functions (FRF). The matrix , formed by the FRFs, makes it possible to obtain all the dynamic characteristics of the system, and can be represented by Eq. (8):

$$\{H(\omega)\} = \frac{\{X(\omega)\}}{\{F(\omega)\}} = \frac{1}{([K] - \omega^2 [M]) + i\omega[D]} \quad (8)$$

Where: [M], [D] and [K] are mass matrices, hysteretic damping and rigidity respectively of the primary system.

2.3 Experimental modal analysis (EMA)

According to Soeiro [19], modal analysis is a process by which a structure is described in terms of its natural characteristics, which are natural frequencies, damping factors and modes of vibration, that is, its dynamic properties.

The classical technique for identification of modal parameters EMA (Experimental Modal Analysis) uses the measurement, with the aid of accelerometers, of excitation signals (artificially performed) and response of the structure . Later, the estimation of the parameters is made with the use of specific algorithms, which allows the use of techniques in the frequency or time domain (HOLANDA, [6]). To facilitate the analysis of the response of the system, the input and output signals are converted from the time domain to the frequency domain using the Fast Fourier Transform (TRF), due to the ease of manipulation of the equations in this domain.

Some of the fundamental assumptions of EMA are:

- The structure is invariant in time (modal parameters are constant) and linear, that is, the structure response to any combination of simultaneous forces is the sum of the individual responses of each of the forces acting alone;
- The structure in test can be adequately described by a discrete model (INMAN, 1994 [8]);
- Special attention with linear behavior which limits applied forces (INMAN, 1994 [8]).

The EMA procedure used in this work is the one proposed by Holanda [7] which uses impulsive forces to exc ite the analyzed structure.

Table 1
Constructive parameters of the concrete beam

Dimensions	Proportion	Supporting condition	Longitudinal armature	Transverse reinforcement
10 x 20 x 200 (cm)	1 : 2 : 4	Bi-supported	Steel bars 8 mm diameter	Steel bars 5 mm diameter

3. Description of the experiment

3.1 Methodology

The methodology used in this work consisted of:
An Experimental Modal Analysis EMA with impulsive excitation of a bi-supported concrete beam with the objective of estimating the natural frequencies and modal forms of the structure. The natural frequencies are used for the tuning of the ADVs. The modal forms are used for optimizing positions of the ADVs. In this work, three control modes were selected.
Considering the natural frequencies associated with the vibration modes to reduce the vibration movement, two dynamic vibration absorbers with different loss factors were tuned to one of the natural frequencies, to evaluate experimentally which design configuration has more effectiveness in reducing vibration moviment.
Once the design configuration was selected, two other tuned devices were built in the other two natural frequencies of interest. With the devices installed a new EMA was performed for performance evaluation (reduction of accelewhere if fixed would have no effective effect whatsoever.ration levels).

3.2 Experimental tests

Initially, a reinforced concrete beam was constructed with the characteristics shown in Table [1].
The data acquisition system was assembled using the equipments and sensors described below:
- IEPE piezoelectric miniature accelerometer, integral cable connector, insulated base with the following characteristics: frequency band: 1 to 20,000 Hz; Mass: 1,5 grams; Sensitivity: 10 mV/g;
- Data Acquisition Module, 4 simultaneous inputs, 24 bit
- AD resolution, 512 K/S /S sampling;
- Instrumented Impact Hammer 22.7 mV/N;
- Notebook, cables and wax for fixing.

A photo of the beam, with the experimental apparatus, is shown in Figure [2].
During the measurement procedures the concrete beam, supported on two supports, was maintained with a free span of 1.80 meters.
The span was subdivided into ten parts spaced 18 cm apart, with position "1" and position "11" occupying the ends of the free span, according to figure [3]. In order to perform the SMA, acceleration data from fifteen acquired impact signals at each point, with a sampling frequency of 6400 Hz were used. During all the tests the accelerometer was in position 2. All data were acquired in the time domain through the use of a routine developed with the aid of Labview® software.

4. Results and discussions

Using a routine developed with the Scilab software the signals obtained in the time domain tests were transformed to the frequency domain using the Fast Fourier Transform (FFT). From the fifteen acquisitions performed, ten were selected, for each of the eleven positions analyzed. The mean of the acceleration response signal was obtained for each position, as well as the respective input signal represented by the force.
Using the input and output signals, the frequency response function (FRF) was calculated as described in equation (8). Analyzing the FRFs, the vibration modes associated with the natural frequencies of 453 Hz, 727.5 Hz and 1031 Hz were selected to be controlled using ADVs. For example, figure [4] shows the FRF amplitude at position 2, where the accelerometer was fixed, with the natural

Figure 2
Bi-supported concrete beam, with installed data acquisition system

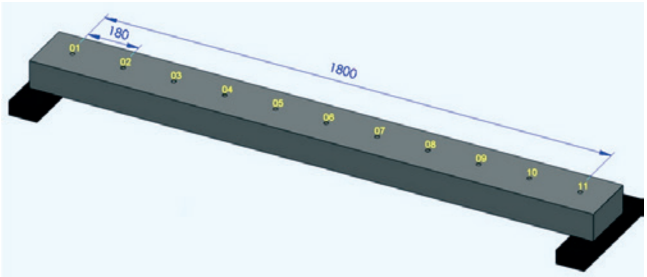

Figure 3
Concrete beam with indication of measuring positions

frequencies from 0 to 2000 Hz marked. The fundamental frequency of 28.75 Hz associated to the first flexural mode is identified, as well as six other resonance frequencies, two of which are related to torsion modes: 72.5 Hz and 99.38 Hz and the others to flexural. Considering the beam position, the flexural modes move the largest section of the structure with the highest displacement amplitude, consequently displacing a larger volume of air, by which the sound is irradiated. Therefore, minimizing the movement of the bending modes, the sound pressure level will be reduced. Figure 5 shows the bending modes of the bi-supported beam, associated with their respective resonant frequencies. In this figure it is possible to identify the positions where ADVs were fixed near the anti-nodes.
In order to determine the best design configuration of the ADVs, through the most efficient parameter of the loss factor of the viscoelastic material, the vibration mode associated with the natural frequency of 727.5 Hz was selected for the tuning of the dynamic vibration absorber. For its construction aluminum for the base, steel to obtain the mass parameter and two types of materials with different viscoelastic properties were used.

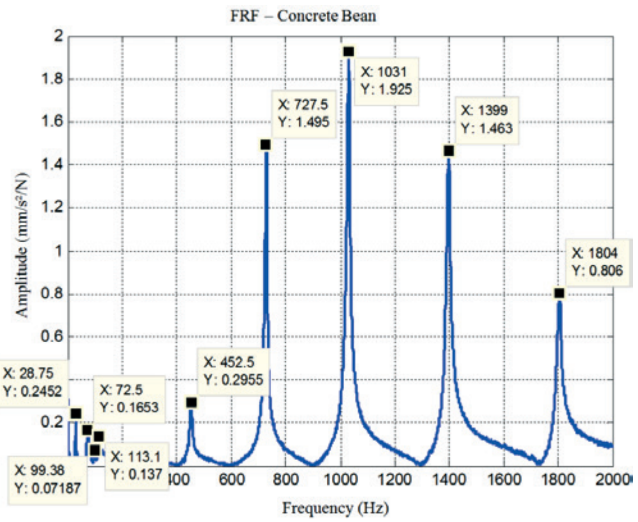

Figure 4
Frequency response function of the reinforced concrete beam

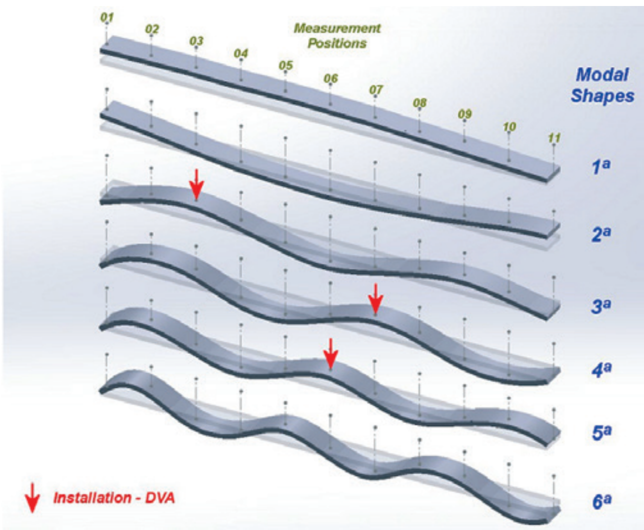

Figure 5
Flexion modes of the reinforced concrete beam

The devices were fixed on an inertial table and tested with impact loading. Figure [6] shows the photo of the experimental apparatus, consisting of impact hammer, accelerometer and data acquisition system. The accelerometer was positioned on the metallic mass and the impact was on the central axis of fixation of the metallic mass. The signals were acquired in the time domain at an acquisition frequency of 2048 Hz and the EMA was used for the estimation of the FRFs.
Figure [7] shows the FRF of the two devices tested. In the figure, the slender curve with the sharp peak corresponds to the lowest loss factor ADV. The flatter curve represents the ADV with the highest loss factor.
The value of the loss factor of the devices was obtained by the band of half power or energy method. According to Magalhães [10] it is a method of one degree of freedom to make local estimates of modal frequency and damping. The method is based on observation that the system response reaches a point of maximum amplitude (peak) near the natural frequency. The frequency value where the extreme value is observed is called the resonance frequency ω_n and is a good approximation of the natural frequency.
The method has the following characteristics: good estimate for the damping factor for materials with low and high damping

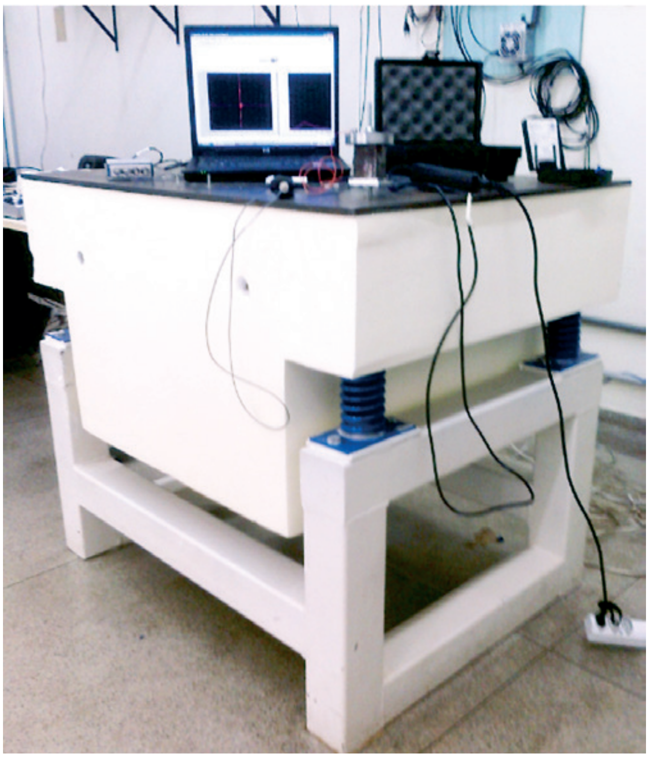

Figure 6
Inertial table with experimental apparatus for AME realization

provided there are adequate number of points in the half power band; easy computational implementation; and better results are obtained when the modes are well spaced.
The damping can be estimated by finding the points ω_1 *and* ω_2, on both sides of the FRF peak, which correspond to half of the amplitude of (3dB below) the resonance frequency. The points ω_1 *and* ω_2 are called half-power points determine a frequency band known as a half-power band. It can be obtained graphically (figure [8]), or by the loss factor determined by Eq. 9:

$$\eta = \frac{\omega_2^2 - \omega_1^2}{2\omega_n^2} \tag{9}$$

This method is particularly sensitive to spectral resolution and significant errors can be introduced in low resolution FRFs where there is a high probability of the peak value of the modes being between two spectral lines. For a good estimation of the loss factor the bandwidth of the half power band ($\Delta\omega = \omega_1 - \omega_2$) of the analyzed resonance must have at least five points in frequency.
Table 2 shows the constructional characteristics of the ADVs represented by FRFs (figure [7]), referred to as “A_1” the higher loss factor device and “B_1” the lower loss factor device, in addition to the natural frequency of the Mode selected by the FRF of the concrete beam (figure [4]). Among the constructive parameters are: the mass of each ADV, the percentage ratio of mass ADV / VIGA and the resonance frequency of the ADVs.
Two new tests were made on the concrete beam with the installed devices, to evaluate which loss factor showed the highest attenuation in the vibratory movement for the selected mode (727.5 Hz). Figure [9] shows the FRFs of the beam with the devices and without the devices installed.
For the device with a higher loss factor, a reduction in the vibration mode amplitude related to the natural frequency of 727.5 Hz was 75.32% (from 1.495 to 0.369 mm / s² / N). However, it is clear from the graph showed in figure [11] the influence on mode associated to 453.1 Hz natural frequency. Considering a frequency band of 400 to 900 Hz, the vibration reduction was 50.81% for a device that added only 1.06% in weight to the original structure.
For the device with the lower loss factor, the vibration mode vibration related to the natural frequency of 727.5 Hz disappeared, dem-

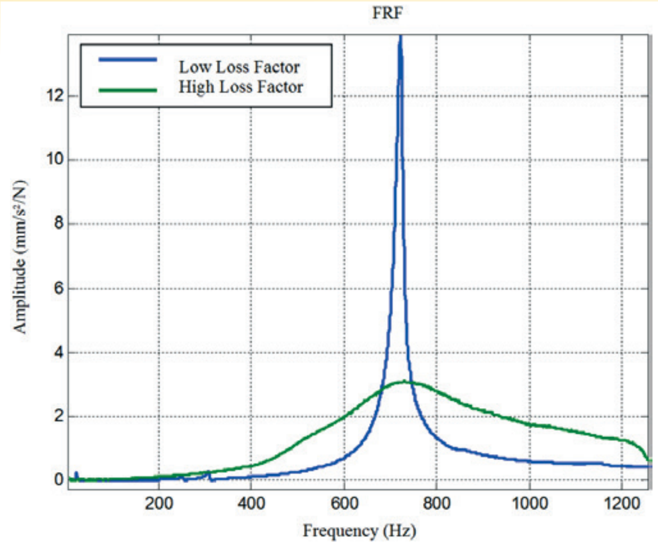

Figure 7
Frequency response function of the ADVs constructed

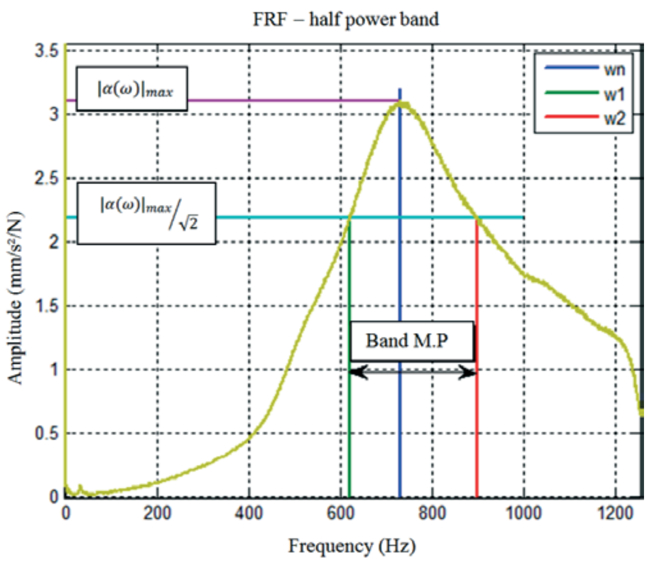

Figure 8
Determination of the loss factor through the half power band method (M.P.)

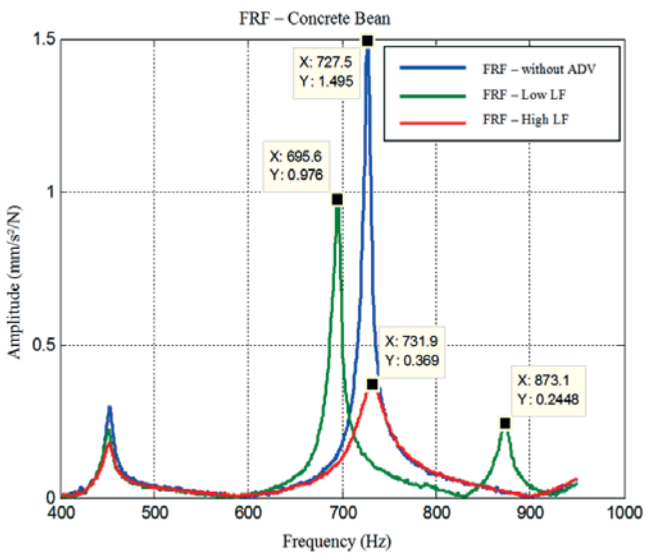

Figure 9
Concrete Beam FRFs with and without installed devices - influence band from 400 to 900 Hz

Table 2
Constructive parameters of ADVs

ADV	MASS (gr)	% ADV/BEAM	FREQ. SINT. (Hz)	FREQ. ADV (Hz)	LOSS FACTOR
A_1	1060,3	1,06	727,5	729,4	0.3926
B_1	1256	1,26		721,9	0.0185

Table 3
Constructive parameters of ADVs

ADV	MASS (gr)	% ADV/BEAM	FREQ. SINT. (Hz)	FREQ. ADV (Hz)	LOSS FACTOR
A_1	2319,9	2,32	453,1	463,1	0.6955
A_2	1060,3	1,06	727,5	729,4	0.3926
A_3	816,3	0,82	1031	975,6	0.2947

onstrating that the reduced loss factor of the viscoelastic material makes the operation of the dynamic vibration absorber similar to the operation of a dynamic vibration neutralizer built with non-damping steel spring. The similarity can also be observed with the appearance of two new resonant frequencies (695.6 Hz and 873.1 Hz) in the structure located on the sides of the eliminated mode . The graph of figure [9] shows the influence on mode associated with a natural frequency of 453.1 Hz. Considering a frequency band of 400 to 900 Hz the reduction of the vibratory movement was 27.06% for a device that added only 1.26% in weight to the original structure.
Considering that the main objective of this work is to reduce the vibratory movement in the structure and not only in a vibration mode, three high loss factor ADVs, with frequencies of resonance close to the natural frequencies already selected were constructed. The constructive parameters of the ADVs: mass, the mass ratio ADV / VIGA, resonance frequency and the loss factor are shown in table [3].
The loss factors of the materials used in the construction of the ADVs are very distinct in order to obtain low secondary/primary structural mass ratio. Even so, it is found that device A_1 - which has the lowest loss factor among the three ADVs constructed – has a loss factor sixteen times greater than the ADV without viscoelastic material which has a loss factor of 0.0185.
If the intention was to control only one mode of vibration the device should be tuned to the natural frequency associated with the mode. However as the control was for three modes, the alteration of the matrices of mass, stiffness and damping with the addition of new degrees of freedom represented by the ADVs altered the natural frequencies of the vibration modes of the beam without the ADVs. Therefore, in order to obtain the reduction in the vibration amplitude, it is sufficient that the ADVs have a natural frequency close to the natural frequencies of the selected modes of the structure.
Table [4] shows the natural frequencies associated to the selected modes and the peak values of the FRFs in acceleration of the eleven positions in which the beam was subdivided without the ADVs installed.
The highest acceleration amplitudes associated to each of the three natural frequencies associated with the vibration modes selected for control purpose are highlighted in Table [4]. The locations of these points are showed in figure [10].
An EMA was performed on the concrete beam with the devices installed, to evaluate the reduction of the vibration movement obtained in the selected modes and in a frequency band of influence from 400 Hz to 1550 Hz. Figure [11] shows the punctual FRFs with and without the devices.
Analyzing figure [11], the selected modes of 453.1 Hz, 727.5 Hz and 1031 Hz show a reduction in the vibration mode amplitude related to the resonance frequency of 79.75% , 69.9% and 79.5%, respectively. Considering the 400-1500 Hz bandwidth the reduction of the vibratory motion was 36,03% with the three devices, that added only 4.2% in weight to the original structure. Den Hartog [4] suggests a relationship between the masses of ADVs and the primary structure of 10% (literature indicates up to 25%). This significant reduction in the mass ratio validates all the methodology proposed in this work.

Table 4
Peak values of the FRFs in the eleven points of the beam

Position	Frequency (Hz)		
	453.1	**727,5**	**1031**
1	0.3184	0.7146	0.4283
2	0.2140	1.331	1.898
3	0.4532	1.077	0.1459
4	0.2038	0.8675	2.048
5	0.2605	1.584	0.0825
6	0.5127	0.07203	2.131
7	0.266	1.650	0.057
8	0.1942	0.9427	2.102
9	0.4265	0.9955	0
10	0.2124	1.312	1.724
11	0.2905	0.6042	0.3243

5. Conclusions

Dynamic vibration absorbers constructed with viscoelastic material whose loss factor is high provided greater reduction in the vibratory

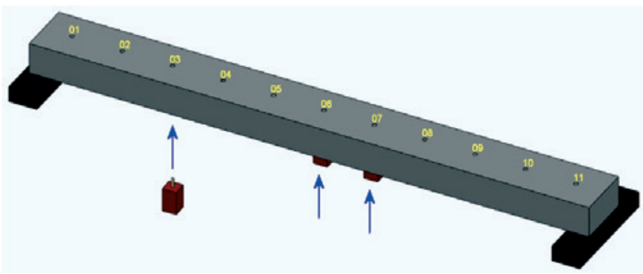

Figure 10
Concrete beam design with indication of fixation positions of the devices

movement of the structure to which they were fixed. In the present study a reduction in the vibration mode amplitude related to the natural frequency of 727.5 Hz of 75.32% (from 1.495 to 0.369 mm / s² / N) was obtained when only one ADV was fixed.

When using a single ADV with the purpose of attenuating the vibratory movement of a structure in a specific vibration mode there is influence in neighboring vibration modes. When an ADV was adjusted to control the vibration mode associated with the 727.5 Hz frequency, there was a reduction in the vibration mode associated with the natural frequency of 453.1 Hz. When considering a frequency band of 400 to 900 Hz, the reduction of the vibratory movement was 50.81%, for a device that added only 1.06% in weight to the original structure.

Devices with a lower loss factor have performance on the structure similar to that of a dynamic vibration neutralizer. The vibratory movement related to the tuned natural frequency disappears and two new resonance frequencies emerges in the structure around the vibration mode eliminated. The vibration mode of the beam associated with the natural frequency of 727.5 Hz disappeared, and two new resonance frequencies in the structure at 695.6 Hz and 873.1 Hz emerged. As in the high loss factor device, influence in the vibration mode associated with the natural frequency of 453.1 Hz also occurred. When considering a frequency band of 400 to 900 Hz, the vibration reduction was 27.06%. The device added only 1.26% in weight to the original structure.

Therefore, if the objective is to eliminate a certain mode of vibration, for example, due to the coincidence of the natural frequency of this mode with the frequency of excitation of an equipment installed on the structure, the use of devices with low loss factor is feasible. Dynamic vibration neutralizers tend to be damaged through fatigue caused by lack of damping. In spite of low loss factor, ADVs constructed with viscoelastic material with this characteristic have damping able to dissipate in the form of heat the vibratory movement, increasing the life of the device.

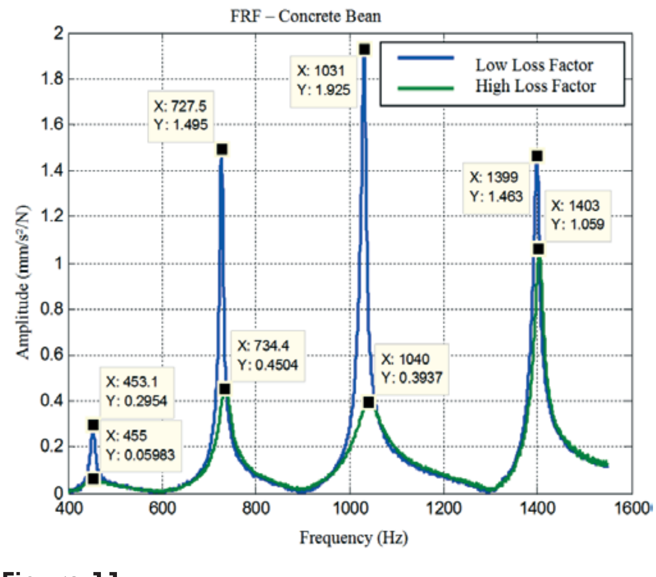

Figure 11
Concrete beam FRFs with and without the three installed devices - influence band 400 to 1550 Hz

But if the aim is to reduce the vibration of the structure in a wide frequency band, for example, with the aim of reducing noise transmitted by the structural route, the use of devices with high loss factor produce better results.

The three dynamic vibration absorbers promoted an average reduction up to 76.4% in the maximum peak acceleration tuned resonances. In 400-1550 Hz frequency bandwidth, with an extra not tunning mode of 1399 Hz natural frequency, the reduction was 36.03%. Although an optimization procedure for selecting the resonance frequency of the ADVs added to the structure was not used, a high reduction in the vibration of the beam subjected to impact loading was obtained.

The devices produced have resonant frequencies different from the natural frequencies of the selected modes, the difference varies from 0.27% for the 727.5 Hz mode and 5.37% for the 1031 Hz mode. By adding mass to the structure through the devices, the structural configuration of the beam changes as well as the values of the natural frequencies. Hence the ideal tuning frequency will hardly be one of the original natural frequencies of the structure without the devices. For this reason the frequency tuning does not have to be the same, only that they are close, so that the devices are effective in reducing the vibratory movement.

The three constructed devices have a total weight of 4,196 kg, corresponding to an increase of 4.2% to the weight of the structure, which is less than that suggested in literature, which can be up to 25% of the weight of the primary structure.

6. Acknowledgments

The authors would like to thank the Federal Institute of Education, Science and Technology of the State of Goiás - IFG and the Foundation for Research Support of the State of Goiás – FAPEG.

7. References

[1] BATTISTA, R. C.; VARELA, W. D, Medidas Corretivas para Vibrações de Painéis Contínuos de Lajes de Edifício. In: XXX Jornadas Sul-Americanas de Engenharia Estrutural, 2002, TRB0282,Brasilia, DF.

[2] BISTAFA, S. R. Acústica Aplicada ao Controle de Ruído. São Paulo: Edgard Blucher, 2006.

[3] DAYOU, J. Fixed-Points Theory for Global Vibration Control Using Vibration Neutralizer. Journal Sound and Vibration, v. 292, p. 765-776, 2006.

[4] DEN HARTOG, J. P. Mechanical Vibrations, 4th edition. Mcraw-Hill, New York, 1956. ESPÍNDOLA, J. J. Passive Control of Structure with Viscoelastic Device, 1999, Congresso Brasileiro de Engenharia Mecânica, vol. 01, PP. CD-R.

[5] ESPÍNDOLA, J. J.; BAVASTRI, C. A.; LOPES, E. M. O. On the Passive Control of Vibrations with Viscoelastic Dynamic Absorbers of Ordinary and Pendulum Types. Journal of The Franklin Institute, v. 347, p. 102-115, 2010.

[6] HOLANDA, R. V. Uma Metodologia de Atenuação de Ruído Transmitido por Lajes Utilizando Absorvedores Dinâmicos de Vibração. 2012. 125 f. Tese de Doutorado, Universidade Federal de Uberlândia, Uberlândia.

[7] HOLANDA, R. V.; DUARTE, M. A. V.; PENA, J. L. O.;

OLIVEIRA, E. L. Um Estudo Sobre a Utilização de Análise Modal Operacional com Excitação Transiente. 2010, VI Congresso Nacional de Engenharia Mecânica, CONEM 2010. Campina Grande - PB.
[8] INMAN, D. J. Engineering Vibration. Englewood Cliffs Prentice Hall, 1994.
[9] LONG, M. Architectural Acoustics. Burlington: Elsevier Academic Press, 2006.
[10] MAGALHÃES, M. C.. Fundamentos de Acústica Estrutural. São Paulo: All Print Editora, 2013.
[11] MARTINS, L. A. C.; SAHB, C. A. S.; NETO, M. F. F., Conforto Acústico de Apartamentos Residenciais Quanto ao Ruído de Impacto. Conferência Latino Americana de construção Sustentável, 1, Encontro Nacional do Ambiente Construído, 10, Anais, São Paulo, 2004.
[12] NASHIF, A. D.; JONES, D. I. G.; HENDERSON, J. P. Vibration Damping. Wiley, New York, Chap. 1, 1985.
[13] NUNES, M. A. A.; DUARTE, M.A. V. Transient Analysis of Great Concrete Structure Using the Finite Element Technique, 2005, Internoise Environmental Noise Control, Rio de Janeiro.
[14] NUNES, M. A. A.; DUARTE, M.A. V. Estudo Sobre a Transmissão de Ruídos em Lajes de Apartamentos Residenciais. 2002, Congresso Nacional de Engenharia Mecânica, CONEM 2002. João Pessoa - PB.
[15] RINDEL, J. H. Sound Insulation of Building. The International Congress and Exposition on Noise Control Engineering, 36, Istambul, 2007.
[16] SANT'ANNA, R. Análise de Acidente com Rotor. Rio de Janeiro: PUC-Rio, 2007.
[17] SILVA B. E. P., PEREIRA J. A., FREITAS T. C. Influência dos Componentes Harmônicos na Análise Modal Operacional de um sistema massa-amortecedor-mola de 5 GL(s), 7° Brazilian Conference on Dynamics, Control and Aplication, UNESP, 2008.
[18] SNOWDON, J. C. Vibration and Shock in Damped Mechanical Systems. New York: John Wiley and sons Inc., 1968.
[19] SOEIRO, N. S. Analise Modal Experimental , Apostila, UFPA, 2001, 61 f..