HYBRID VIBRATION CONTROL APPLIED TO STRUCTURES UNDER BROADBAND EXCITATION USING VISCOELASTIC NEUTRALIZER AND ADAPTIVE FILTER

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Abstract. *Due to the constant technological advances, the need for controlling vibratory phenomena is more and more present in modern society, demanding techniques of increasing level of sophistication. This work, tuned to that need, deals with the development and practical implementation of a hybrid vibration control system implemented over a metallic beam excited by a broadband signal. The system is called hybrid because it consists of two subsystems: a passive one and an active-adaptive one. The passive subsystem consists of a viscoelastic dynamic vibration neutralizer (DVN), whereas the active-adaptive subsystem comprises sensors, actuators and a digital signal processor (DSP), on which a control strategy based on a LMS algorithm is developed. Both the viscoelastic neutralizer, designed to optimal action at 25 ºC, and the active-adaptive subsystem work all over the frequency bandwidth in which the beam is excited. The adaptability of the hybrid control system is verified by varying the temperature at which the beam is tested. This variation, between 5 and 35ºC, affects directly and considerably the DVN performance. That demands the active-adaptive subsystem to adapt, in order to compensate the DVN loss of performance and maintain an overall level of vibration control. It is verified, along the current work, that a hybrid vibration control system, based on viscoelastic neutralizers and adaptive filters, is a powerful and flexible tool, since the advantages of each subsystem can be associated towards a more reliable and less expensive control action.*

Keywords*: Adaptive Filter, Dynamic Vibration Neutralizer, LMS Algorithm, Vibration Control, Viscoelastic Material.*

1. INTRODUCTION

The interest in hybrid (passive + active) vibration control systems has increased fast in the last decades (Ahmad *et al*, 2009, Pu *et al*, 2010, Svensson *et al*, 2010), especially because it has been observed that passive and active control systems are well complementary (Franchek et al, 1995, Marra, 2007). In vibration control, a passive vibration control system (PVCS) contributes with robustness and high performance at higher frequencies, while an active-adaptive vibration control system (AVCS) contributes with flexibility and great performance, especially at lower frequencies. Thus, the association of these two systems permits the design of a very high performance hybrid vibration control system (HVCS), with robustness, flexibility and reliability. In this work, a hybrid vibration control system (HVCS), designed to attenuate vibrations in a broadband excited metallic beam, is introduced and discussed.

Passive vibration control techniques are based on modifications of stiffness, mass and/or damping characteristics of the mechanical system of concern (Mead, 2000). Their largest advantages are the relative low cost and the robustness. These techniques can provide great results when working in stable environments. Their disadvantage is, exactly, the reduced flexibility when any modifications or adjustments are needed.

On the other hand, active vibration control techniques are commonly based on vibration cancelling strategies. They are implemented through sensors, actuators and, specially, a DSP capable to receive some samples of the excitation signal, process it, and generate an output control signal to be applied to the mechanical system, in order to cancel (or reduce) the original vibrations (Fuller *et al*, 1997). In case of any modifications in the mechanical system of concern (due to several factors), some active systems are capable of adapting their transfer functions, according to some predefined criteria, in order to better cancel the vibrations of the "modified" mechanical system (Clark *et al*, 1998). So, the advantages of those techniques are flexibility and, in some cases, adaptability.

2. FUNDAMENTAL CONCEPTS

2.1. Passive Vibration Control System (PVCS)

In the current application, the passive vibration control system (PVCS) consisted of a viscoelastic dynamic vibration neutralizer (DVN). As seen in Figure 1, the DVN is a secondary system (in this case, with mass m_2, stiffness k_2 and damping c_2), added to a primary mechanical system (with mass m_1 stiffness k_1 and damping c_1) the vibration of which is desired to be controlled. A DVN can be designed to work either in a tonal frequency or under broadband conditions.

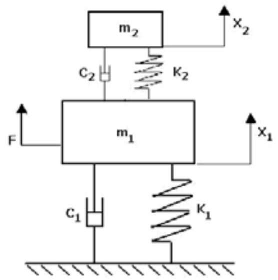

Figure 1. Schema of a dynamic vibration neutralizer (DVN)

It should be stressed that the use of a DVN transforms the primary mechanical system (plant) into a 2 degree of freedom system, the natural frequencies of which are different from the primary system, according to designer's interests. The damping of the DVN is used to determine its frequency band of action. Some design parameters must be well chosen in order to attain a good control system. Considering ω as the frequency (excitation), ω_1 as the natural frequency of the primary mechanical system, ω_2 as the natural frequency of the secondary mechanical system and the following relations:

$$r = \omega / \omega_1 \quad ; \quad \mu_m = m_2 / m_1 \quad ; \quad \beta = \omega_2 / \omega_1 \quad ; \quad \zeta = c_2 / 2 m_2 \omega_1$$

and setting $\mu_m = 0.25$ and $\beta = 0.8$ (optimal parameters), it can be seen in Figure 2 (Inman, 2008) the damping influence in the normalized system response. The broadest frequency band attenuation is for $\zeta = 0.27$.

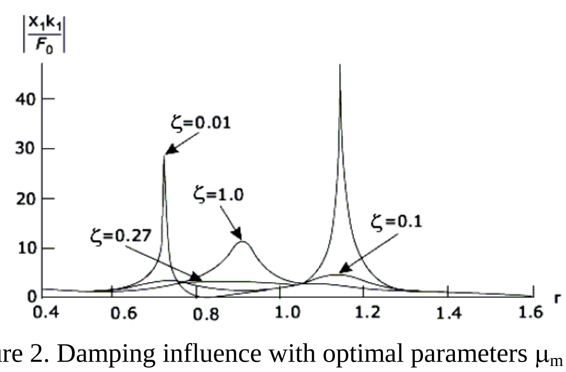

Figure 2. Damping influence with optimal parameters μ_m and β

The viscoelastic DVN characteristics of stiffness and damping are determined by the viscoelastic material utilized in the device (Bavastri, 1997). In these cases, the usual approach is to resort to the complex stiffness model, by which the viscoelastic elements of the DVN are described by a complex stiffness, which is a function of frequency and given by

$$k_c(\omega) = k(\omega)[1 + j\eta(\omega)] \tag{1}$$

where $k(\omega)$ is denominated dynamic stiffness and $\eta(\omega)$ is the loss factor.

The stiffness $k_c(\omega)$ is complex and frequency dependent because the moduli of elasticity of viscoelastic materials are also complex and frequency dependent. In fact, they are functions of temperature as well. Each modulus of elasticity can be written in terms of a dynamic modulus and the corresponding loss factor, in the same way as the complex stiffness. Figure 3 shows the typical behavior of the dynamic shear modulus and the corresponding loss factor of a viscoelastic material as functions of frequency and temperature. The way the complex moduli of viscoelastic materials can be modeled by fractional derivatives and then experimentally determined is found at Espíndola *et al*, 2005.

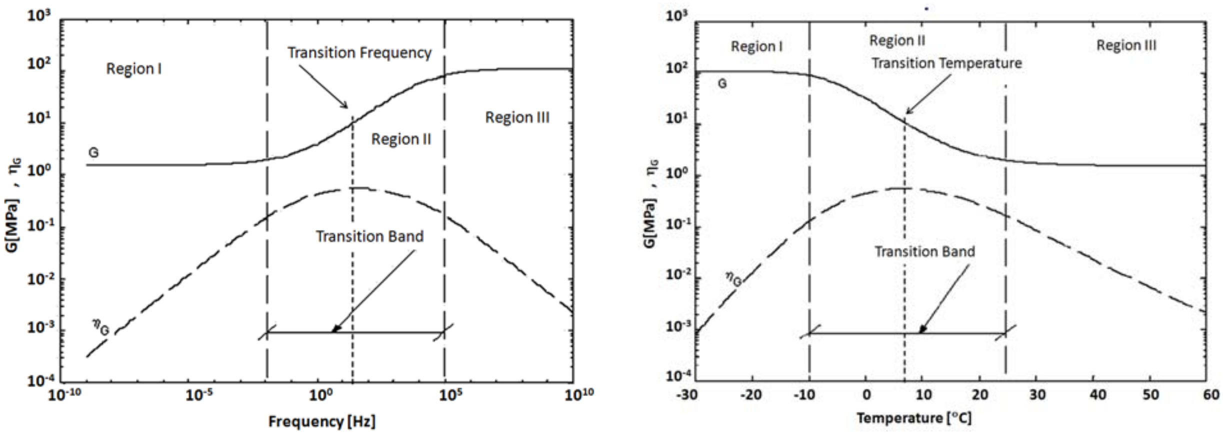

Figure 3. Dynamic shear modulus and loss factor as functions of (a) frequency and (b) temperature

Considering these features, the viscoelastic DVN is designed to work in the so called “transition region”, where the loss factor is maximum. That implies larger frequency band and vibration attenuation (Espíndola *et al*, 2010).

2.2. Active-adaptive Vibration Control System (AVCS)

In this work, the active-adaptive vibration control system (AVCS) was implemented using digital filters, the objective of which was to model the plant (primary mechanical system) behavior, in order to generate an efficacious control signal that had a broadband action over the primary system’s vibrations.

An AVCS can be implemented in different ways, regarding the architecture, control algorithm, type of sensors and actuators, available signals and operation restrictions (processing time, objective function and so on). As to the architecture, two basic approaches can be listed: feedback and feedforward, which was employed in the present work.

The basic architecture of a feedforward control comprises a sensor, which captures the input signal, a control unit, which processes the input signal according to a given algorithm and generates the output signal, and an actuator, which applies the output signal. The control unit can be thought of as a set of weight filters which, in this case, were finite impulse response (FIR) filters, as opposed infinite impulse response (IIR) filters.

2.2.1 FX-LMS Algorithm and FIR filters

FIR filters have the important characteristic of stability. They have no poles in their transfer function and its output signal vector is given by (Farhang-Boroujeny, 1998)

$$y(n)=\sum_{i=0}^{N-1} w_i(n)\,x(n-i) \tag{2}$$

where n is the discrete time, w is the main filter coefficient vector, x is the input signal vector and N is the main filter size. As FIR filters are non-recursive structures, they can demand an excessive number of coefficients to model systems with complex transfer functions, such as vibratory systems with resonances within the frequency band of excitation.

The popularity of FIR filters can also be justified by the fact that its objective function can be derived as a quadratic function, with an unique point of minimum. This point can be found in a prompt way through an adaptive algorithm. In the case of IIR filters, the objective function may have several local points of minimum and the algorithm may converge to any of those local points of minimum, instead of the global minimum (Marra, 2007).

According to the design theory of Wiener filters (of stochastic origin), the optimal coefficients for FIR filters can be obtained by the mean square error (MSE) minimization as

$$\xi=E\left[\,|e(n)|^2\right] \tag{3}$$

where e is the error signal and E is the expected value. The error, in this algorithm, is computed from

$$e(n) = d(n) - y(n) \tag{4}$$

where *d(n)* (the desired signal) is the plant response to the excitation signal *x(n)* or, in other words, the signal to be canceled. The signal *y(n)* refers to the control signal generated by the control algorithm.

The LMS (least-mean-square) algorithm is the most used algorithm for adaptive filtering, due to its simplicity and robustness. The FX-LMS (filtered-x LMS) algorithm is based on the minimization of the error signal variance. This algorithm takes the instantaneous square error as an estimative of the mean square error (MSE). A considerable disadvantage of the LMS algorithm is its high dependence of the input (excitation) signal power spectral density (PSD), so that the more uniform the input signal PSD, the faster the algorithm's convergence.

With some considerations and mathematical manipulations (Marra, 2007), the coefficient's adaptation equation for the FX-LMS algorithm, implemented with FIR filters, can be expressed by

$$w(n+1) = \upsilon . w(n) + 2.\mu . e(n) . xfs(n) \tag{5}$$

where n is the discrete time, w is the main filter coefficient vector, υ is leaky factor, μ the adaptation step size, e is the error signal and xfs is the excitation signal filtered by the feedback and secondary filters.

The control architecture employed by the authors is schematically shown in Figure 4, where $\hat{\mathrm{F}}(\mathrm{z})$ is the feedback path estimation filter, W(z) is the main filter coefficient vector and $\hat{\mathrm{S}}(\mathrm{z})$ is the secondary path estimation filter.

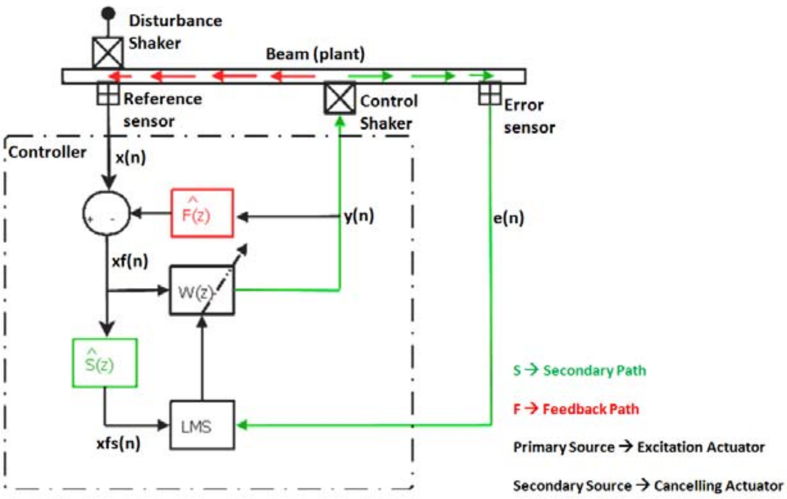

Figure 4. Schema of the active-adaptive vibration control system

The consideration of the feedback and secondary paths is of fundamental importance to the success of a practical implementation of an AVCS. The feedback path estimation filter $\hat{\mathrm{F}}(\mathrm{z})$ must be inserted in the controller in order to filter the reference signal (captured by the reference accelerometer) from the physical vibration propagation from the control source to the reference sensor, through the metallic beam. Thus, a "clear" reference signal can be delivered to the controller, which is a control law requirement, according to Eqs. (2) and (5). The secondary path takes into account all the dynamics inherent to the experimental set up, which means the dynamics of transducers, A/D and D/A converters, amplifiers and analog filters. These paths are considered in the algorithm through digital FIR filters, which estimate the real path dynamics (see Figure 4).

2.3. Hybrid Vibration Control System (HVCS)

The above PVCS and AVCS were combined to form a hybrid vibration control system (HVSC). Apart from the extended performance, an HVSC like this, shown in Figure 5, can offer a greater reliability as a global vibration control system, since, in case of failure of any "subsystem", the other one can maintain some overall vibration control level.

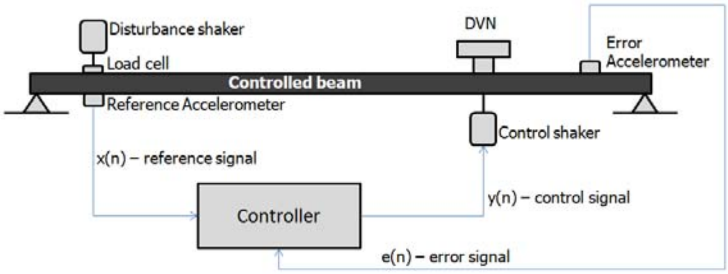

Figure 5. Schema of the hybrid vibration control system

3. METHODOLOGY

The object of control (primary mechanical system, or plant) was a 0.5 kg steel beam, whose dimensions were 930 x 23 x 3 mm. The applied excitation signal (a white noise filtered to a band of 200 to 430 Hz) is represented in time and frequency domains in Figure 6. This signal excited the plant in its 4th, 5th and 6th vibration modes, which was known by previously performing both theoretical and experimental modal analysis on the beam. A frequency response function (FRF) of the beam under the above excitation signal is shown in Figure 7.

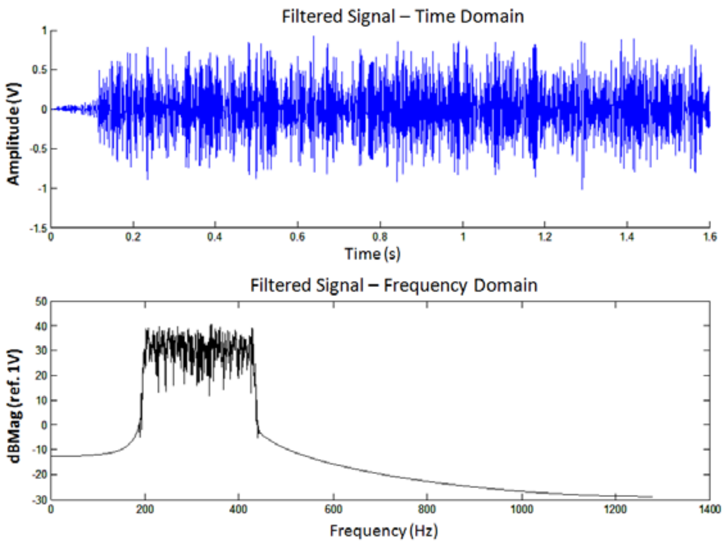

Figure 6. Excitation signal (filtered white noise, 200-430 Hz)

The PVCS, a viscoelastic DVN with a lump of mass of steel and 2 elements of butyl rubber, was optimally designed to act in the frequency band of 190 to 440 Hz and the temperature of 25ºC (Coan, 2005). It was installed at a point where there were significant displacements for the vibration modes of interest. The positioning of the viscoelastic DVN and of all the AVCS components can be schematically seen in Figure 5. The controller in Figure 5 was based on an Analog Devices board, the EZ-KIT Lite ADSP 21161N.

Before running the control tests, the identification routines for the secondary and feedback paths had to be performed, in order to have a complete FX-LMS algorithm implemented, as portrayed in Figure 4. Those paths were identified through an off-line procedure and the digital filter of each path represented its own impulsive response. Then, those filter coefficients were kept constant and reloaded into the algorithm, representing “static filters” during the control tests. It should be clear that the main FIR filter – W(z) – was not “static” like the feedback and secondary filters, but its coefficients were adjusted at each algorithm iteration (on-line, real time updating), according to Eq. (5). The use of on-line identification techniques for the feedback and secondary paths is also possible and has already been reported as advantageous (Gontijo, 2006). Those techniques, however, were not employed in this investigation.

As observed in Coan, 2005, for the implementation of an active-adaptive vibration control system, the task of tuning the FX-LMS algorithm is very important and deserves a special attention, because of its high influence on performance. The optimal tuning parameters found in this work, according to the same methodology used previously (Coan, 2005), are shown in Tab. 1, where N, Ns and NF are the number of coefficients of the main filter, the secondary path filter and the feedback path filter, respectively.

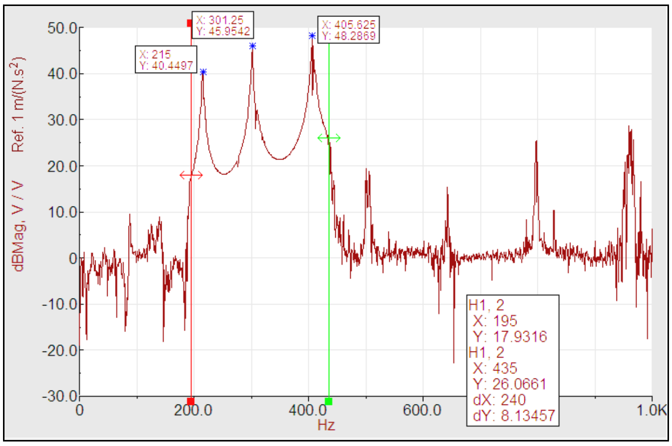

Figure 7. Beam's FRF (without the viscoelastic DVN)

Table 1. FX-LMS optimal parameters for active-adaptive vibration control

N	**Ns**	**Nf**	**υ**	**μ**
300	500	500	0,9999995	1,50E-06

The experimental set up, which allowed all the investigations regarding the vibration control systems, is displayed in Figure 8, inside a temperature chamber.

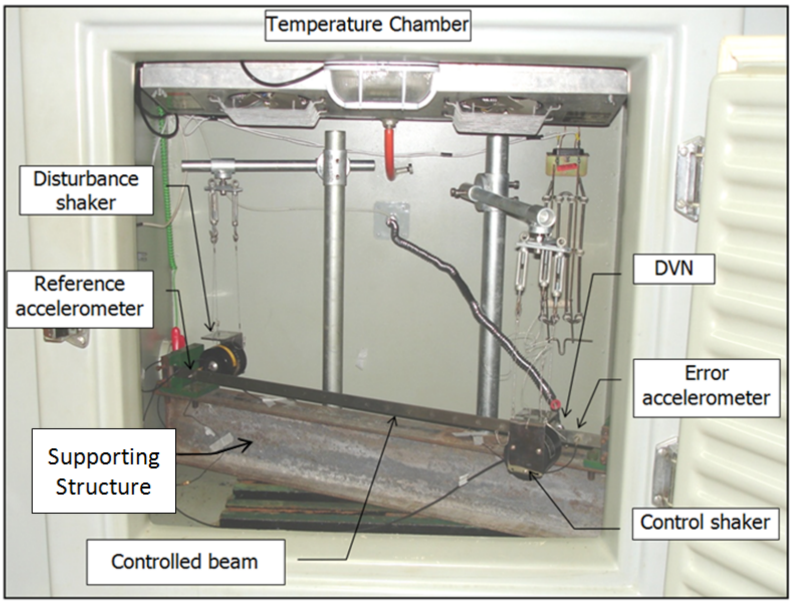

Figure 8. Experimental set up

In a first stage, for the plant alone, it was computed a ratio of the acceleration at a point of the plant to the disturbing force, in the frequency domain. That ratio was denominated a response to force ratio (RFR) [1]. After installing the DVN and maintaining exactly the same excitation signal (disturbing force) used before, it was possible to evaluate the DVN attenuation performance by computing a response to force ratio (RFR) of the compound system (plant + DVN). Then, the AVCS was turned on, acting on the compound system (plant + DVN), and its performance was evaluated by an RFR of the global system (plant + DVN +AVCS), still keeping the same disturbance force. All those tests were run at room temperature (around 25ºC, the optimal DVN design temperature).

[1] A response to force ratio (RFR) is a ratio between the response at a point in the system of concern and the disturbing force at the same or at other point, in the frequency domain (Marra, 2007). This frequency function was introduced to provide a common ground in which all the attenuation performances could be compared. When just the disturbing force acts on the system (plant alone or plant + DVN), an RFR is a frequency response function (FRF). When the disturbing force acts on the system simultaneously to another external force, in the current case, a cancelling (actuator) force, an RFR is no longer an FRF. Even though, it is revealing of the changes which take place in the system when the AVCS works on it and allow valid comparisons to be made with other RFRs, as long as exactly the same disturbance force is employed.

The next stage was to test the adaptability of the AVCS and that was carried out by placing the experimental set up inside a temperature chamber, as seen Figure 8. By varying the temperature, it was expected that the DVN performance would deteriorate and the coefficients of the main digital filter would be automatically adjusted, according to Eq. (5), such that the adaptability of the AVCS would maintain a satisfactory vibration level. To some extent, that did happen, as will be shown in the next section.

Those adaptability tests were performed at the temperatures of 5, 15, 25 and 35ºC. At each test temperature, before taking the measurements, it was considered a 30 minute interval to allow the temperature of the viscoelastic elements of the DVN to stabilize.

All the RFRs were computed between the error sensor point and the disturbance actuator point (see Figure 8). The requirements on the disturbing force were carefully kept.

The attenuations obtained in each test condition were calculated according to the Eq. (6). This equation was used to check the effectiveness of each control system in its action over the frequency band of interest.

$$\mathrm{Att}[\mathrm{dB}] = 20\log_{10}\left\{\frac{\left[\sum_j y_j^2(f_j)\right]^{1/2}}{\left[\sum_j x_j^2(f_j)\right]^{1/2}}\right\} \tag{6}$$

In the above equation, y_j represents RFR (response to force ratio) magnitudes along frequencies f_j, when any control system is ON, that is, plant with any vibration control system. As to x_j, it represents the corresponding RFR magnitudes when that control system is OFF, that is, plant without that vibration control system. In this context, the term “vibration control system” can be applied either to the passive system or to the hybrid system. In fact, Eq. (6) represents the RMS (root mean square) value of the vibration attenuation due to each control system, in the frequency band of interest.

4. RESULTS

The results were as follows.

4.1. Passive Vibration Control System (PVCS)

Figure 9 depicts the effectiveness of the DVN over the frequency band of interest, including at the resonance peaks.

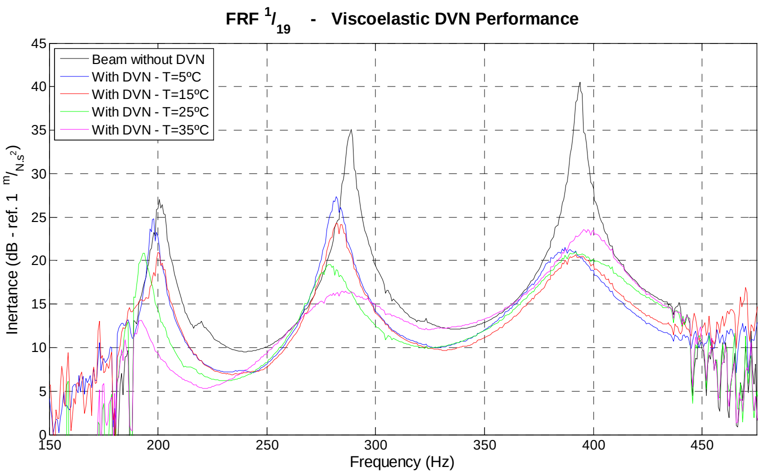

Figure 9. Passive vibration control system (viscoelastic DVN) performance

It was expected that the DVN performance would vary with the temperature, due to the behavior of the viscoelastic material (see Figure 3) and that was confirmed. In a broadband analysis, there was loss of performance when the temperature was different from the design value of 25ºC, as observed in Tab. 2. In general, the farther the temperature the worse the DVN broadband performance.

Focusing on the resonance peaks only, it is observed that the greatest attenuations, regarding each mode of vibration, did not occur at 25ºC, as recorded in Tab. 3. However, it must be remembered the viscoelastic DVN was not primarily designed for tonal action, but for broadband action. In fact, for tonal passive control, an “undamped” DVN should be used instead.

Table 2. DVN performance (vibration attenuation) – Broadband analysis

Test temperature [ºC]	RMS broadband attenuation [dB]
5	7,9
15	9,1
25	9,5
35	8,9

Table 3. DVN performance (vibration attenuation) – Resonance peak analysis

Test temperature [ºC]	Resonance peak attenuation (4th mode) [dB]	Resonance peak attenuation (5th mode) [dB]	Resonance peak attenuation (6th mode) [dB]
5	2,0	8,0	19,0
15	6,0	11,0	20,0
25	6,0	16,0	19,5
35	14,0	18,0	17,0

4.2. Hybrid Vibration Control System (HVCS)

The results obtained with the PVCS and the HVCS (comprising both passive and the active-adaptive vibration control systems) are shown in Figure 9, for various temperatures. It can be observed in the corresponding curves that the PVCS (DVN) generally produced a small displacement of the resonance peaks, apart from the characteristic "smoothing". These effects are typical in cases of damping addition. When the AVCS was turned on and the full HVCS was in place, drastic modifications were observed on the RFR curves, especially at the resonance peaks and at frequencies where antiresonances were introduced in the global system RFRs.

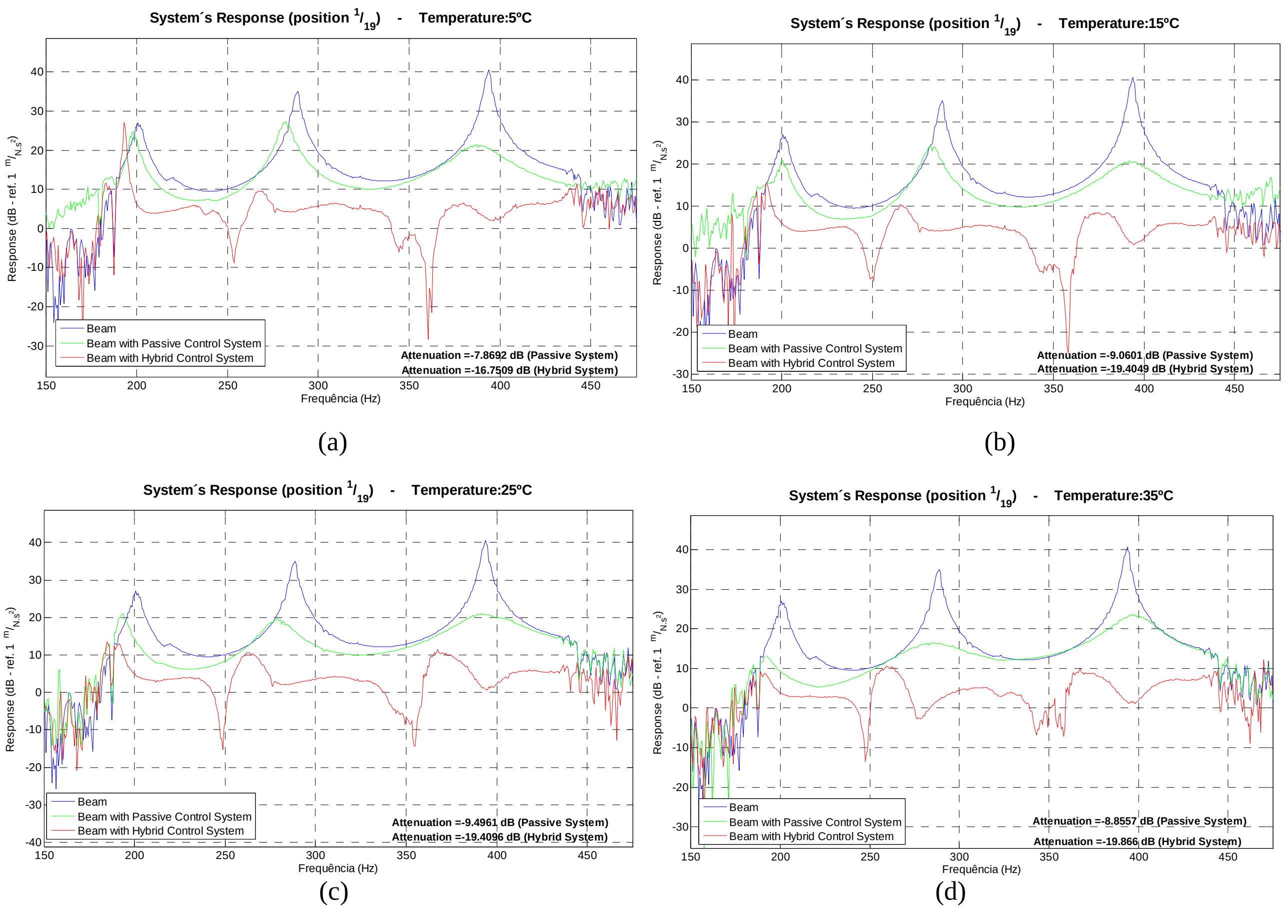

Figure 10. PVCS and HVCS performances at (a) T=5ºC, (b) T=15ºC, (c) T=25ºC and (d) T=35ºC

The "hard work" of the AVCS at the resonance peaks can be explained as follows. The control signal generation was based on a plant on-line identification method and this makes the highest energy frequency components of the system response (the resonance frequencies) to be carried to the main filter coefficients. Thus, the control signal was generated with the same characteristics, that is, the control signal had the same highest energy frequency components as the plant response. Hence, when the control signal was reintroduced into the system (see Figure 5), its action would be more efficacious exactly at the resonance peaks.

Table 4 presents the performance of each control system at each test temperature, considering the system's response at frequency band of interest, according to Eq. (6).

Table 4. PVCS and HVCS performance (vibration attenuation) – Broadband analysis

Control System / Temperature	Passive [dB]	Hybrid [dB]	Difference (HVCS to PVCS) [dB]
5ºC	7.9	16.8	8.9
15ºC	9.1	19.4	10.3
25ºC	9.5	19.4	9.9
35ºC	8.9	19.9	11.0

Considering the performance of the vibration control systems by the attenuation at the resonance peaks (see Figure 10), more expressive values were observed. In some cases, the global attenuation, when the HVCS was in place, was around 35 dB.

The behavior of the vibration control systems were also monitored through the error sensor signal, in the time domain, as displayed in Figure 11. Those curves reinforces the greater efficacy of the HVCS.

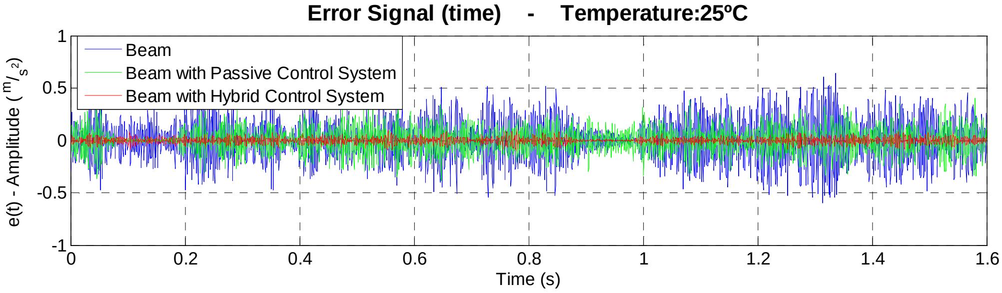

Figure 11. Error sensor time history

The adaptation capability of the AVCS, as part of the HVSC, can be inferred from the above results. However, in order to make this point even more clear and help to visualize how the LMS algorithm deals with the main filter coefficients to generate the best control signal at each test condition, the main filter coefficients were traced at each test temperature. The results are presented in Figure **12**, where it is clearly observed the adaptability of the AVCS according to the behavioral modifications of the mechanical system due to the temperature dependence of the viscoelastic DVN.

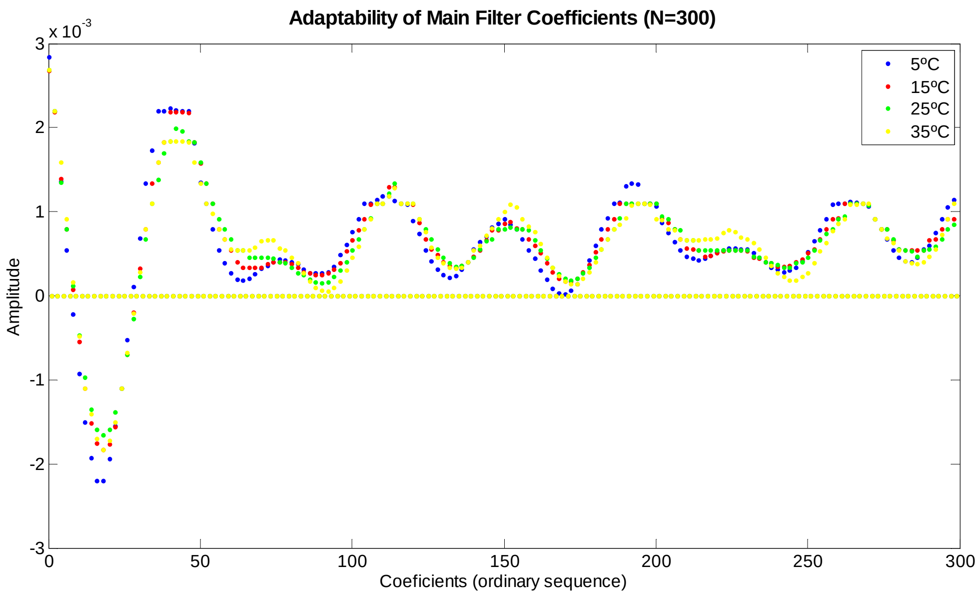

Figure 12. Adaptability of main filter coefficients

5. CONCLUSIONS

The viscoelastic DVN (PVCS) performance in terms of vibration attenuation and robustness could be verified once more, following the long and successful chain of broadband applications in which the same technique for viscoelastic DVN design was employed (Bavastri,1997, Espíndola *et al*, 2010)

The adaptive behavior of the AVCS became well characterized when there were alterations in the system parameters due to changes in temperature, confirming and extending some previous efforts of tonal nature (Coan, 2005). In most cases, the filter coefficients were satisfactorily adjusted according to the new system behavior and compensated the viscoelastic DVN loss of performance, maintaining a global attenuation at around 19.5 dB. Even when that level was not reached, a very significant effort was made by the AVCS. It is worth mentioning that the adaptive capability is very important to compensate not only system modifications but also some modeling errors or imprecision in design, which improves the applicability of the AVCS.

Finally, the hybrid vibration control system (HVCS), comprising the above PVCS and the AVCS, demonstrated to be a powerful tool in vibration control, in which the advantages of each system can be well associated in order to have an adaptable, flexible and robust global control system, of reduced cost. This association also contributes to a more reliable system, where the responsibility is shared between the control systems and some overall vibration level can be maintained in case of failure of any of the systems.

6. ACKNOWLEDGEMENTS

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8. RESPONSIBILITY NOTICE

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