Numerical and Experimental Results of Passive Control for Noise and Vibration Reduction

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Abstract
During last years many research projects have been funded by European and American Institutions with the aim to increase passengers’ acoustical and vibrational comfort, in aerospace as well as in automotive field. In order to control noise radiated and vibration levels in vehicles and structures, different approaches have been used. Hybrid active-passive damping treatments seem to be the most effective by combining robustness and reliability of passive methods with high performance of active ones. Experiences gained on using passive damping treatments, mainly on 2-D structures, have been used for studying typical truss space structures and assessing the ability of damping treatments to reduce structural vibrations. In addition, the temperature sensitivity of viscoelastic damping materials has been experimentally measured and its effect simulated in numerical models. The paper presents the characteristics of viscoelastic damping materials within “glassy” and “rubbery” behaviour’s typical temperature, and discusses the possibility to use them, in a numerical model, for simulating damping treatment’s effect in operating conditions on dynamic response of a truss structure. The results obtained during these preliminary studies are quite discouraging leading to the conclusion of the inefficient use of such damping treatments for truss-like structures. Completely different results are obtained, on the other hand, when using active control approach, which may be even technologically more efficient, for such kind of structures. As a final results a sort of hybrid truss-panel structure has been studied which may offer an alternative approach to reduce the vibrations of such typical structural architecture.

Introduction
The studies and the simulations regarding the passive approach to noise and vibration problems may be dated in the seventies and beginning of eighties. After that the interest of researchers and even industries has been essentially devoted to active controls of these unwanted phenomena. The results of this technology have, in some cases, totally suppressed the use of a passive approach, but, generally speaking, the magnified expected benefits have not been achieved and therefore some revival of previous approaches have been reconsidered. Such restored interest toward the application of passive treatments for noise and vibration control has motivated the need for an improved knowledge of the simulation of such complex energy dissipation mechanism, together with their most significant parametric variations, [1]-[2]. The added value to the studies of passive approach to the noise and vibration problems may be found on the use of the new simulation and modelling techniques which were not available in the past, [3]-[4]. Similar
motivations are also applicable to the possibility of simulating active control laws in a complex geometry framework, [5], without the need of implementing a state-space approach.

With this background it was decided to understand the applicability of both passive and active vibration control technologies to truss-like structures which are often implemented in space applications. The paper collects the experience gained in measuring the characteristic parameters of aerospace damping treatments, and using a similar approach as used in previous applications, such materials are simulated on a truss structure. The modal analysis of such structure has been initially performed. In view of future experimental correlations, such results may be useful for helping in the correct definition of the finite elements which may be used for the numerical model. Such analysis has been numerically completed, showing some differences between rod and beam dynamic behaviour of the assembled truss structure. The dynamic frequency response of the structure has been used as parameter for evidencing the differences among the models and the simulations.

The results of these different simulations are discussed herein leading to the conclusion that the passive damping application is almost useless for vibration reduction of a truss structure. By comparison the application of even simple active control strategy may be much more efficient. A concluding remark is finally outlined regarding the possibility of using a stressed skin approach which could improve the effectiveness of an hybrid vibration control strategy.

**Viscoelastic materials characterisation**

The American Society for Testing and Materials (ASTM) developed a standard test procedure based on the Oberst beam method [6] for measuring vibration damping properties of materials, including loss factor, $\eta$, Young's modulus, $E$, and shear modulus, $G$. Assuming the structure is a two layered system, namely a metal beam and a surface damping treatment, the analysis allows either to evaluate the damped response of the composite system if the damping properties of each layer are known, or, as in the present case, to estimate the damping properties of one layer if the damped response of the composite system is known. The Oberst beam technique retains the use of non-contact transducers for the system input as well as for the response acquisition.

The UNNA-DPA vibro-acoustic team, on the other hand, tried to modify these latter aspects in order to develop a modified set-up introducing some simplifications in the application of the excitation force and the measurement of the response. In Figure 1 the measurement section of the set-up is shown.
Figure 1: Damping material properties measurement set-up

Being the exciter constrained to the free edge of the bar, the non-contact requirement is not met. The introduced approximation does not affect the damping but the response of the bar, which, at least at low frequencies, could differ from the clamped-free model on which the calculation is based. The reason of this choice, apart from the practical simplicity of its application, lies on the possibility to excite the structure at higher frequencies than the non-contacting systems as far as with higher force values, permitting as a consequence to excite thicker beams, i.e. being fixed the maximum thicknesses ratio, to test thicker materials. Indeed, beam-damping treatment thicknesses ratio must be kept within certain values in order to be able to measure a typical “modal” behaviour response (with detectable resonances peaks) even once the damping treatment has been applied. However, validation tests have showed that the modified set-up presents cantilevered behaviour above 400 Hz.

The calculation technique consists in very accurate frequency response spectra extraction from both specimens in order to determine with high frequency resolution the resonance frequencies and the damping values (the latter only for the composite beam) for each vibration mode. Once these data are available, the parameters of interest are calculated via the Ross-Kerwin-Ungar (R-K-U) equations, [1], properly simplified for two layer systems and in presence of complex moduli. The viscoelastic materials properties are temperature-frequency dependent, so, for a complete characterization of such materials, measurements (and the following calculations) have to be made at different temperatures using an environmental chamber. If the damping treatment is unconstrained, as for a two layered system, a cycling bending on the composite beam produces a tension compression deformation of damping material. Under these conditions, the R-K-U analysis gives, for each test temperature, the following relations for the bending stiffness and loss factor ratios:

\[
\frac{EI}{E_1I_1} = \frac{1 + 4e_2h_2 + 6e_2h_2^2 + 4e_2h_2^3 + e_2^2h_2^4}{1 + e_2h_2} \tag{1}
\]
\[
\eta = \frac{e_1 h_1 (3 + 6h_2 + 4h_2^2 + 2e_2 h_2^3 + e_2^2 h_2^4)}{(1 + e_2 h_2)(1 + 4e_2 h_2 + 6e_2 h_2^2 + 4e_2 h_2^3 + e_2^2 h_2^4)}
\]

For a beam with a free layer damping treatment, the equations (1) and (2) allow to evaluate the properties of the damping material. If the frequencies, the loss factors and the densities of the damped and undamped beams are measured for the same modes of vibration, the following equations are obtained

\[
\left(\frac{\omega_n}{\omega_{in}}\right)^2 (1 + h_2 \rho_r) = \frac{1 + 2e_2 h_2 (2 + 3h_2^2 + 2h_2^2) + e_2^2 h_2^4}{1 + e_2 h_2}
\]

\[
\eta_n = \frac{e_1 h_1 (3 + 6h_2 + 4h_2^2 + 2e_2 h_2^3 + e_2^2 h_2^4)}{(1 + e_2 h_2)(1 + 4e_2 h_2 + 6e_2 h_2^2 + 4e_2 h_2^3 + e_2^2 h_2^4)}
\]

Substituting the values of the geometrical properties and densities into equations (3) and (4), they can be solved for \(e_2\) and \(\eta_2\).

Tests have been conducted on a 0.55x0.05x0.002 m aluminium bar within -30 and +22 Celsius degrees in the frequency bandwidth 0-5000Hz. In the following Figure 2 the curves fitting loss factor values calculated from experimental measurements at the natural flexural frequencies of the bar are presented.

Figure 2: Loss factor curves at varying temperature

Figure 3 presents the corresponding Frequency Response Functions measured in one acquisition point of the bar.
It is possible to notice how the treatment affects the damping distribution of the host structure as far as its mass and stiffness distribution: for high temperature mass effect can be noticed since natural frequencies move towards lower values if the damping treatment is applied, while for low temperature, due to the transition from rubbery and glassy behaviour of the viscoelastic layer, the stiffness effects is dominant since natural frequencies move to higher values. The achieved results in terms of storage modulus and loss factor have been later use to simulate viscoelastic layers application on the truss structure under investigation.

The FEM approach to active vibration control

The feedback control algorithm may be implemented inside the well known commercial code MSC/Nastran by using the TF data card which allows the definition of the control law with the consequent modification of the structural dynamic matrices as, [5]:

\[
[M + M_c] \ddot{x} + [C + C_c] \dot{x} + [K + K_c] x = \{F\}
\]

(5)

where the suffix c stands for parameters of the control algorithm. This equation may substitute the state space form which corresponds to

\[
[B] \ddot{z} = [A] \dot{z}
\]

(6)

with

\[
\{z\} = \begin{bmatrix} \{x\} \\ \{\dot{x}\} \end{bmatrix}, \quad [A] = \begin{bmatrix} 0 & \begin{bmatrix} M + M_c \end{bmatrix} \\ -[K + K_c] & -[C + C_c] \end{bmatrix}, \quad [B] = \begin{bmatrix} 0 \\ [M + M_c] \end{bmatrix}
\]

(7)

The simplicity and efficiency of this approach may be given by the following example on a simple cantilever beam, using, separately, a stiffness and a damping control law.
The first control algorithm adopted for the test structure, later referred to as type 1, is more complex since it is based on both stiffness and damping control, i.e.,

$$u = -K_p y - K_d \ddot{y} \quad (8)$$

Furthermore, another control strategy based on acceleration control, $u = -K_a \ddot{y}$, later referred to as type 2 has been applied.

**Test structure**

Numerical simulation have been carried out on a typical space structure, showed in Figure 5. The structure is a 3 longeron truss with 5 triangular bays. It consists of 48 Aluminium tubular struts connected at 18 nodes. The total mass is 2.58 Kg and each strut has an outer diameter of 20 mm and it is 1 mm thick.

Figure 5: The truss structure tested

**Modal analysis**

The truss finite element model has been built following two different approaches that is by using either bar or rod elements for the truss struts. The bar element is a one-dimensional bending element which is prismatic, and for which the elastic axis, gravity axis, and shear centre all coincide. The rod element includes instead only extensional and torsional properties. The modal analysis of the two FE models of the structure has been performed and its results have been analysed in the form of a numerical-numerical correlation. To correlate models well known indexes such as
the Modal Scale Factor (MSF) and the Modal Assurance Criterion (MAC), usually calculated with the aim to correlate numerical and experimental results, have been used.

\[
MSF(x, p) = \frac{\{\psi_x\}^T \{\psi_p\}^H}{\{\psi_x\}^T \{\psi_x\}^H}
\]

\[
MAC(x, p) = \frac{\{\psi_x\}^T \{\psi_p\}^H \{\psi_p\}^T \{\psi_p\}^H}{\{\psi_x\}^T \{\psi_x\}^H \{\psi_x\}^T \{\psi_x\}^H}
\]

Where \(\{\psi_x\}\) and \(\{\psi_p\}\) are the structure’s modes calculated with the different approaches.

It can be seen that the results achieved for the modes of the two models up to 2000 Hz are very similar except first two symmetric structure’s translational mode. The analysis of the FE model built with rod elements does not give at all the possibility to observe this mode. That is because by simulating truss struts with these elements is not possible to observe flexural deformations. On the other end, truss struts mainly work under axial stress and then rod element are able to simulate in the correct way their behaviour. In order to evaluate the risk of any structural failure due to the loss of a single truss strut, a modal analysis has been performed on the structure from which a strut has been previously removed.

<table>
<thead>
<tr>
<th>MODE</th>
<th>FULL TRUSS (BAR)</th>
<th>STRUT 1-11 REMOVED</th>
<th>STRUT 41-51 REMOVED</th>
<th>MODE SHAPE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>108.5</td>
<td>65.2</td>
<td>122.4</td>
<td>1\textsuperscript{st} torsion along z</td>
</tr>
<tr>
<td>2</td>
<td>270.7</td>
<td>271.0</td>
<td>273.7</td>
<td>1\textsuperscript{st} bending</td>
</tr>
<tr>
<td>3</td>
<td>318.5</td>
<td>298.7</td>
<td>360.2</td>
<td>2\textsuperscript{nd} torsion along z</td>
</tr>
</tbody>
</table>

Table 1: Modal analysis results comparison

Moreover, a model upgrading has been performed for the FE model built with bar elements. Double number of nodes has been used.
Direct frequency response

In order to perform a direct frequency response analysis of the structure a dynamic load has been applied at one of the grid points that in operation conditions has to be connected to the main structure. For the present numerical simulation white noise in the frequency range 0-2000 Hz has been used loading the degree of freedom along the truss axis. The dynamic response of the structure has been evaluated along directions x, y, z in the grid points 51-53 on the free tip of the truss.

<table>
<thead>
<tr>
<th></th>
<th>ROD</th>
<th>BAR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damaged structure</td>
<td>1-11</td>
<td>41-51</td>
</tr>
<tr>
<td>Viscous damping</td>
<td>1-11;2-12; 3-13</td>
<td>41-51;42-52; 43-53</td>
</tr>
<tr>
<td>Viscoelastic damp. –30°C</td>
<td>1-11</td>
<td>41-51</td>
</tr>
<tr>
<td>Viscoelastic damp. –10°C</td>
<td>1-11</td>
<td>41-51</td>
</tr>
<tr>
<td>Viscoelastic damp. +10°C</td>
<td>1-11</td>
<td>41-51</td>
</tr>
<tr>
<td>Active control type 1</td>
<td>1-11;2-12; 3-13</td>
<td>41-51;42-52; 43-53</td>
</tr>
<tr>
<td>Active control type 2</td>
<td>1-11;2-12; 3-13</td>
<td>41-51;42-52; 43-53</td>
</tr>
</tbody>
</table>

Table 2: Performed simulation - damaged or damped struts are highlighted

In the following figure, a comparison is presented between FRF calculated with using the structure FE models already illustrated from which can be seen that using of bar elements provides a stiffness increase to the structural model.

Figure 7: FRF at a truss tip (grid 51) in direction z –bar (continue) vs rod (dotted) model

Direct frequency analysis on the damaged structure leads to FRFs as the ones presented in the following figure from which can be clearly seen that a possible damage gives rise to later modes, so that the structure’s behaviour changes
dramatically. Herein the cases of two possible failures are presented, respectively related to missing of strut 1-11 and strut 41-51.

Figure 8: FRF at a truss tip (grid 51) in direction z –original vs damaged

Passive damping treatments effectiveness on the test structure seems not to be very effective, especially at low frequencies. This may be justified by the dissipation mechanism of viscoelastic materials (VEM). VEM are mainly dissipating vibration energy by working in bending mode while for truss structures the stress is transmitted in an axial way via the aluminium structural elements. Furthermore, classical methods for FE simulation of VEM, that work properly with plane elements, don’t work in case of linear elements. Instead, by applying on some struts a viscoelastic layer 1 or 5 mm thick, almost no improvement is achieved in numerical FRF from the damping point of view, even at temperatures and frequencies for which the loss modulus, given by the product between storage modulus and loss factor and proportional to the dissipating energy, is very high.

If in the FE model the CVISC element that defines a viscous damper, with a 10% damping at the first structural vibration mode, is used the accomplished damping level is neither very high. In Figure 9, the FRFs calculated while applying each time 3 viscous dampers at 3 struts in different positions are presented.
Active damping control strategies applied to the structure show an higher efficiency and, as it can be seen in Figure 10, even though with only 3 active struts, good results are achieved.
Hybrid truss-panel structure

In order to verify on a plane structure effectiveness of passive damping simulation, a FE model of such a structure has been built on which a viscoelastic layer has been applied. The encouraging results achieved (see Figure 11) have induced the authors to investigate the applicability of passive control on stressed skin like structures. In this kind of structures, the stability is ensured by panels rather then by the cross struts.

![Figure 11: FRFs –original vs damped at varying temperature](image)

With this aim, further tests have been carried out on a modified FE model that differs from the original one because of cross struts removal and application of aluminium panels 1 mm thick within each truss bay. The damping effect of viscoelastic layers bonded on each panel or only on few of them has been later simulated, considering that in aerospace applications the degree of damping is limited by thickness and weight restrictions. The achieved results seem to be quite encouraging, so that further investigation is needed in this field with the aim to optimise damping treatment parameters as material choice, thickness and positioning. The FE simulation of the treatment has been performed introducing in the input file values previously calculated from experimental acquisitions. In the following Figure 13 can be seen the damping effect of the VEM layer applied at different temperatures.

Conclusions

Performed studies have clearly showed that the passive damping design needs a deep knowledge of the damping material properties and their behaviour with temperature and frequency. The accomplished experimental procedure gives the opportunity to easily extract parameters needed for numerical simulations of such treatments but further studies are needed to better understand their applicability.
Use of a feedback broadband control algorithm have proved its effectiveness on a truss like structure. The validity of the approach, without being exclusive with respect to the state space approach, has the possibility to consider other FE applications with the study of control algorithms. Experiences done with fluid-structure interaction problems may be easily considered with reference to both passive and active control tools, maintaining the feeling of the structural analyst with the physical understanding of the parameters in use and the related degrees of freedom. Such approach may combine both approaches and may revitalize the use of past methods and materials with new numerical tools which may result very useful for the designer in studying system for passive and active noise and vibration control. In addition the possibility to modify the architecture of a truss-like structure comes from the implementation of a stressed-skin approach, removing the diagonal members and using very thin panels for the stability of truss structures. For such structure the passive damping approach appears to recover its inherent dissipation properties. The stressed skin solution may also offer advantages other than the application of damping foils, and it may be worthy to consider such configuration inside a multidisciplinar optimisation approach.
List of symbols

\[ E = \text{Young's Modulus} \]
\[ I = \text{Second moment of area} \]
\[ H = \text{Thickness} \]
\[ e_2 = \frac{E_2}{E_1} \]
\[ \eta = \text{Loss factor} \]
\[ \eta_n = \text{Loss factor of the composite beam in the nth mode} \]
\[ h_2 = \frac{H_2}{H_1} \]
\[ \omega_n = \text{nth circular frequency of the composite beam} \]
\[ \rho = \text{Density} \]
\[ \rho_r = \frac{\rho_2}{\rho_1} \]

subscripts 1 and 2 refer to metal and damping material respectively

References
