Recent Developments on the Optimization of Viscoelastic Coating

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Viscoelastic coating is widely used to attenuate both vibrations and acoustic radiation of vibrating structures. A non-exhausting number of papers have been published in the past to show the general performance of the technique. However, the effort made by manufacturers to build lighter and more powerful forces people to use the technique in an optimal manner. In fact, in many fields such as aerospace and aeronautics, the addition of extra weight due to the application of viscoelastic layers plays a critical role since it always drives to additional energy costs. Furthermore, the structures in these fields being frequently subject to a wide range of temperature and frequency variations, the optimization procedure should take these effects into account. In other more conventional industry applications, although the added weight is not necessarily a problem of primary concern, an overall layer setup may lead to the overheating of the machinery. This paper aims to illustrate such an issue using a simple vibrating structure.

The structure that is studied in the present paper is a rectangular thin plate covered symmetrically on both sides by rectangular sections of viscoelastic material (see Fig.1). For simplicity, The boundary conditions of the plate are supposed to be simply-supported along the edges. The Love-Kirchoff theory for thin plates under small displacements is used and its displacement along the z-axis is approximated according to the Rayleigh-Ritz method by a polynomial expansion over the x and y-axis. For sinusoidal excitations applied perpendicularly to the plate's surface, this formulation leads to a classic system of second-order linear differential equations. The response of the system as well as the damping factor of each mode can be easily solved by standard numerical methods.

Considerable research is being done in the area of analytical modeling on the stiffness and damping properties of viscoelasic materials. Several curve-fitting formulas derived directly from experiments made on commonly used viscoelastic sheets are available. These formulas give the storage modulus and the loss factor of the material as a function of temperature and frequency. In this work, the formulas for the viscoelastic layer of Soundcoat-D have been taken.

Depending on the thickness and the configuration of the viscoelastic material coverage, both the modal damping factors of the system and its response to external forces over a given frequency range will be optimized. To define the coverage configuration, the plate is first divided into a certain number of small sections and the components of a configuration vector $c(i,j)$ are assigned a value of 1 or 0, depending whether the corresponding section (i,j) is covered or not.

An optimization algorithm is applied to the system (see Fig. 2). The particularity of the problem is to avoid overlapping of the layers. Consequently, we have the layers' location as an optimization variable that is not a continuous quantity. To this end, a simple trivial routine is developed, which proceeds with a given percentage of the plate to cover and a fixed thickness of viscoelastic material. Two objective functions can be analyzed. The first one is the damping factor of a particular mode and the second one is the response of the system over a given frequency range. The algorithm starts off with a randomly chosen initial configuration and calculates the objective function. Then it makes a slight change in the configuration and compares the objective function obtained by this coverage with the previous one. If the result is better, the new configuration is kept. Otherwise, another change is made starting from the previous configuration. This routine is repeated until an optimum is reached. Note that a verification of the convergence of the solution has been made during its calculation to make sure that a global optimal solution was obtained.

A simply supported 30 cm by 45 cm by 3,175 mm aluminum plate is taken for the analysis. The response of the plate alone and the response of the plate entirely covered on both sides by a 1,5875 mm sheet of Soundcoat-D are presented in Fig.3. The response is in fact the mean quadratic velocity of the plate. Note the reduction of the vibration level and the shift of the resonance peaks to the left due to the effect of the added mass and stiffness to the system. This result is given for a room temperature. The so-called **gain** is defined as follows:

$$Gain = \frac{Undamped\ Area - Damped\ Area}{Undamped\ Area} \times 100\%$$

where the area is the area under the curve of mean quadratic velocity of the corresponding system (damped or undamped).

The optimization routine has been applied on the minimization of the mean quadratic velocity of the plate for forced vibration problems. A typical result is presented on Table 1. A force of 1 N was applied at x=0 cm and y=8,7 cm. Table 1 shows the results for three different frequency bands. For each frequency band of interest, the last column of the table gives the optimal configuration. Again the 1/4 coverage shows itself more effective than the full coverage. The optimization process enhances even more the damping performance, reaching a damping increase varying from 78.0% to 88.3% compared to the bare plate, which is approximately twice the performance of the full coverage and about 50% higher than the non-optimized ones. It is interesting to note that, depending on the dominating mode involved for each frequency band and on the location of the exciting force, the obtained optimal arrangement is a combination of the ones previously obtained for each single mode.

To better illustrate the additional weight to the structure, the gain is expressed as a function of the percentage of weight added for the frequency range of 50 -300 Hz in Fig.4. The weight of the plate alone is used as the reference. Note that the maximal thickness of the coverage is limited to 3,175 mm (the thickness of the plate) since our model was made with the assumption of thin structures. Therefore, the results obtained in this range of thickness can be considered very accurate. For the same amount of viscoelastic material, the quarter coverage with optimal shape reaches a reduction of 78 % while the full coverage gives only a reduction of 42 %. To get the same reduction level, the full coverage requires an additional weight of 45 %, compared to 18 % for the quarter coverage. The half coverage gives an intermediate performance between the full coverage and the quarter coverage.

The aforementioned results related only to a certain number of selected configurations. As illustrated, the applied viscoelastic layers mainly bring three effects to the system (mass, stiffness and damping) to form a complex coupled system with the plate, so that an optimal result is a compromise based on these factors. Moreover, the details of the mechanical properties and applied excitations can also affect the optimization results. Generally speaking, modal damping optimization results seem to confirm the commonly used rule of applying the layers at the anti-nodal portion of the structure with optimally shaped arrangement. However, for the forced response, in addition to the variation of mechanical properties of the viscoelastic materials due to the temperature variation, the excitation position and the boundary conditions of the structures are the main factors affecting the energy partition over the structure. Consequently the optimization result on the response depends on a good knowledge of these factors. The methodology developed in this paper permits a systematic handling of the problem with all these factors taken into account. It should be pointed out that the formulation developed in the present work can be easily extended to treat more realistic structures in practical applications. A combination of the optimization procedure and the finite element method seems to be a reasonable way to tackle more complex systems.

Acknowledgment: The author wishes to thank Robby Lapointe for his contributions.

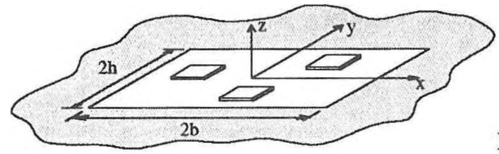

Fig.1

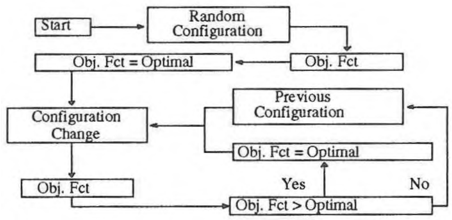

Fig.2

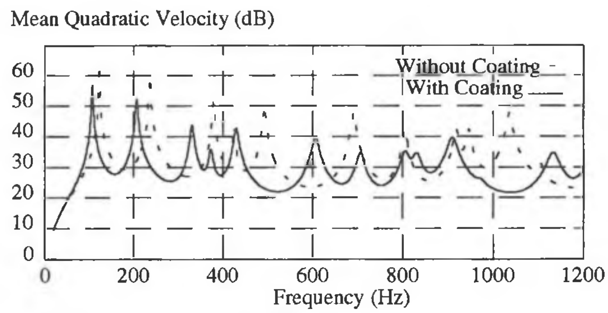

Fig. 3

	Gain (Bare Plate as Reference)		
Frequency Band	Full Coverage	1/4 Coverage	Optimized 1/4 Coverage
50 - 300 (Hz)	42,8 %	59,7 %	78,0 %
300 - 550 (Hz)	44,0 %	49,4 %	88,3 %
550 - 800 (Hz)	31,3 %	55,6 %	86,9 %

Table.1

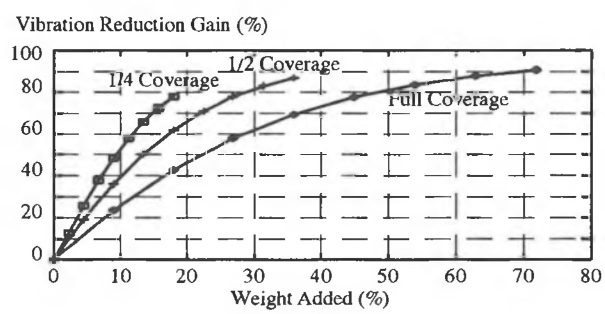

Fig.4