Simulation of beaded and curved panels with multi-layer damping treatment

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Abstract

By using equivalent material parameters and a special equivalent shell element, multi-layer damping treatments can be integrated into simulations of structural vibrations without significant increase of either modelling or computing efforts. The treatment representations are used to conduct classical single-layer FE calculations and determine the frequency response functions of the structure in which the individually treated panels (non-flat) are now included. The procedure presented here does not increase the number of active degrees of freedom, so that it is possible to include the effect of these treatments in large system level models. Two representative examples have been numerically investigated and a practical application of the procedure to a real car floor has been conducted (comparing simulation with measurement). These examples confirm the accuracy, the efficiency and the flexibility of the procedure.

1. Introduction

The use of viscoelastic damping materials is known in many applications for solving noise and vibration problems appearing in automotive, engine, office, computer, military, aerospace and transportation systems. Concretely in automotive systems, panels such as floor and dash, or around the engine and trunk compartment, even the roof or door are often treated with viscoelastic damping layers to reduce vibration and therefore noise radiation.

A damping system may consist of a single layer of damping material glued on the panel or may be between two metallic layers (i.e. constrained damping). Additionally, there may be adhesive films between the layers, which finally lead to a 3 to 5 (or even more) layer configuration. These damping systems introduce damping effects but at the same time stiffen the panel and increase the mass of the system. In addition to the complexity of the system and its effects, the material parameters of a viscoelastic material are often slightly frequency and heavily temperature dependent. To represent such a system with a multi-layer FE model needs consequently much modelling and calculation effort, and is sometimes not feasible in complex models.

In order to simulate such a complex multi-layer system in an efficient and effective manner, strong efforts have been devoted to an equivalent material parameter procedure in the last years. Various sophisticated techniques such as Oberst theory [1], Ross-Ungar-Kerwin equations [2] and single element modelling based on variational asymptotical theory [3] have been developed to describe multi-layer damping treatments. These methods can handle complex constructions with arbitrary layers. But the application has been limited to flat panels until now.

Rieter has developed a procedure for calculating the equivalent material parameters using a transfer matrix procedure called "SIMOKELL" [4] to calculate bending stiffness and corresponding loss factor, from which equivalent parameters of a single layer representing a flat multi-layer system can be derived. By using the transfer matrices approach, the procedure is applicable to configurations with unrestricted numbers of layers. Due to the fact that direct frequency response methods are applied within the procedure, any frequency and/or temperature dependencies of individual layer materials can be also taken into account correctly. These equivalent parameters from SIMOKELL for bending deformations together with analytical solutions for in-plane deformations can then be implemented into finite element models to solve structural vibration problems of real non-flat panels, with damping treatment.

In this paper, the following conventions are used:

Known material parameters for the ith layer

E_i	Young's Modulus
h_i	Thickness
η_i	Loss Factor
ρ_i	Density
υ_i	Poisson's Ratio

Equivalent material parameters for *n*-layer system

E_{eq}^B	Equivalent Young's Modulus (Bending)
η_{eq}^B	Equivalent Loss Factor (Bending)
E_{eq}^M	Equivalent Young's Modulus (Membrane)
η_{eq}^M	Equivalent Loss Factor (Membrane)
ρ_{eq}	Equivalent Density
h_{eq}	Equivalent Thickness
υ_{eq}	Equivalent Poisson's Ratio.

2. Viscoelastic damping materials

The material parameters of viscoelastic materials are usually highly temperature dependent and also slightly dependent on frequency. Both an *increase in temperature* at constant frequency and a decrease in frequency at constant temperature lead to a softening of the material and a change in loss factor. The structural properties of materials can be represented by the storage modulus and the loss factor, which are usually presented in the form of a nomogram (reduced frequency representation) [6, 7]. Before starting any calculation of multi-layer systems, the properties of the individual materials must be known. For viscoelastic materials, several measurement methodologies exist, of which the so-called Oberst method – based upon resonance evaluations of treated beams – is well established in the field of automotive applications. By using a sophisticated version of the Oberst method with a highly precise curve-fitting algorithm, reliable material data can be obtained in the frequency range between approx. 50 and 500 Hz for temperatures between –40 and +100 °C. By introducing these measurement results into the reduced frequency representations together with a careful selection of the nomogram model, the procedures enable finally the inter- and extrapolation of material parameters for any chosen frequency, even beyond the experimentally measurable range.

3. Methodology

The theory of plates and shells [8] assumes that sections normal to the middle plane remain plane during deformation and the normal stresses in z direction are small (Fig. 1), hence strains in that direction can be neglected. With these two assumptions it is easy to see that the total state of deformation can be described by u_0 and w_0 of the middle surface (z=0) and a rotation θ_x of the normal (for simplicity, the deformation in the y direction will not be described in the following). Thus the local displacements in the direction of the x and z axes are

$$\begin{aligned} u &= -\theta_x z + u_0 \qquad &\text{with} \quad u_0 &= u_0(x) \\ w &= w_0 & w_0 &= w_0(x) \\ & & \theta_x &= \theta_x(x) \end{aligned} \tag{1}$$

Writing the appropriate constitutive relations the stresses σ_x and τ_{xz} can be evaluated and hence the stress resultants are obtained as

$$M_x = -\int_{-t/2}^{t/2} \sigma_x z dz = -\frac{Et^3}{12}\frac{\partial \theta_x}{\partial x} \tag{2a}$$

$$P_x = Et\frac{\partial u_0}{\partial x} \quad (2b)$$

$$S_x = \beta Gt(-\theta_x + \frac{\partial w_0}{\partial x}) \quad (2c)$$

where E and G are Young's and shear moduli, β a constant respectively.

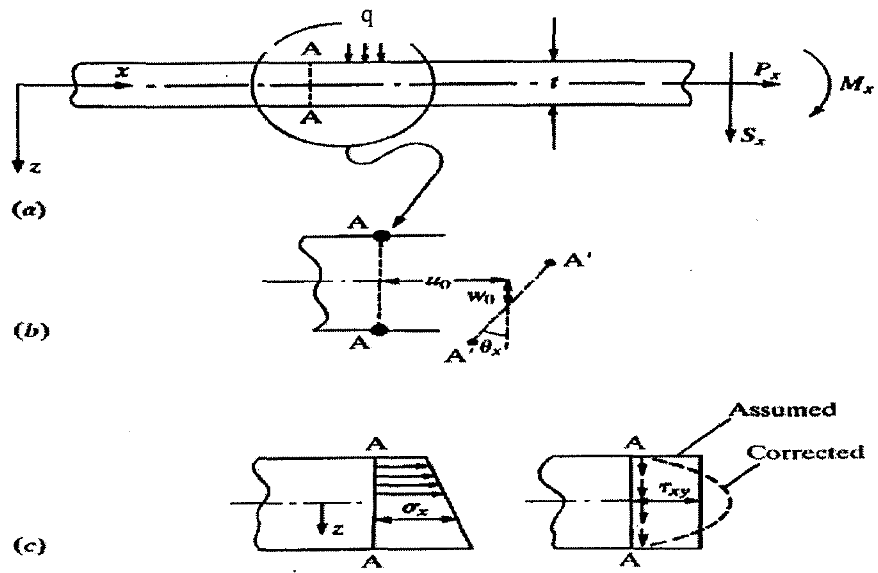

Figure 1 Displacements and stress resultants for a typical plate

Three equations of equilibrium complete the formulation:

$$\frac{\partial M_x}{\partial x} + S_x = 0 \quad (3a)$$

$$\frac{\partial S_x}{\partial x} + q = 0 \quad (3b)$$

$$\frac{\partial P_x}{\partial x} = 0 \quad (3c)$$

In the elastic case of a plate, it is easy to see that the in-plane displacements and forces u_0 and P_x decouple and the problem of membrane deformations can be dealt with separately. This is the basic principle that a single equivalent layer can be used to represent a multi-layer system. Such a single equivalent layer has totally different parameters for bending and membrane deformations.

3.1. SIMOKELL for bending deformation

Using the transfer matrix formalism, in which shear and bending waves are properly included for all layers, SIMOKELL [4] calculates the fundamental natural frequency and related modal loss factor of an infinite flat multi-layer panel, for which the boundary condition is to be pinned by a periodic rectangular grid. By tuning the bending stiffness and material loss factor of a single-layer panel with the same boundary condition such that the same resonance response is obtained, the bending stiffness (with real component B^B_{sys}) and loss factor (η^B_{sys}) of the equivalent system are derived. By making the grid finer or coarser, the modal frequencies can be shifted up or down and the evaluation of the equivalent material parameter is extended to a full frequency dependent function. As mentioned before, SIMOKELL can consider an arbitrary number of layers (panels, damping or constrained layers), of arbitrary thicknesses and can be temperature and frequency dependent. Each layer must, however, be isotropic.

By choosing an equivalent thickness h_{eq} and an equivalent Poisson's ratio υ_{eq} for both bending and membrane stiffnesses, the equivalent Young's modulus of the material for bending is derived,

$$E^B_{eq} = \frac{12(1-\upsilon^2_{eq})}{h^3_{eq}} B^B_{sys} \tag{4a}$$

For the equivalent thickness h_{eq}, the panel thickness of the original body-in-white model is usually chosen. The loss factor of the material η^B_{eq} is identical to the loss factor η^B_{sys} of the bending deformation of the multi-layer panel

$$\eta^B_{eq} = \eta^B_{sys} \ . \tag{4b}$$

3.2 Analytical solution for membrane deformation

As has been described at the beginning of this section, the total deformations can be represented appropriately by a superposition of pure bending deformations and pure in-plane deformations. For the in-plane deformations, the in-plane stiffness and dissipation can be integrated over all layers, yielding the following expressions:

$$E^M_{eq} = \frac{1}{h_{eq}} \sum_{i=1}^{n} E_i h_i \ . \tag{5a}$$

$$\eta_{eq}^{M} = \frac{1}{E_{eq}^{M} h_{eq}} \sum_{i=1}^{n} \eta_i E_i h_i \ . \tag{5b}$$

The equivalent density ρ_{eq} is specified as follows,

$$\rho_{eq} = \frac{\sum_{i=1}^{N} \rho_i h_i}{h_{eq}} \ . \tag{5c}$$

3.3 A special equivalent shell element

In sections 3.1 and 3.2, it is shown that seven independent parameters are necessary to describe the equivalent layer, which represents a multi-layer system. In the general case:

E_{eq}^{B} , η_{eq}^{B}	(calculated from SIMOKELL)
E_{eq}^{M} , η_{eq}^{M} , ρ_{eq}	(analytical solution)
h_{eq} , υ_{eq}	(chosen).

The MSC/NASTRAN [5] shell element allows different Young's moduli to be assigned to the three deformation types, membrane, bending and shear. However, this possibility is not extended to the loss factor , i.e. if membrane stiffness and loss factor are specified, this loss factor will also be assigned to all other deformations. On the other hand, the loss factors for bending and membrane can differ considerably in the case of an equivalent layer formulation as shown above. This problem can be solved by duplicating the shell elements, at least for the non-flat treated panel areas. The procedure results in a formulation with 2 superimposed shell elements, each carrying stiffness and loss factor for one deformation type (bending or membrane). Despite the increase of shell elements as compared to the bare mesh, this method does not increase the number of active degrees of freedom of the model and therefore requires considerably less computational resources than the 3D-approach with solid elements. The shell element mesh from the untreated panel can thus be used for the equivalent layer approach with minimal modifications. The frequency dependency of material parameters can be taken into account – when necessary – by using a direct solution technique. Unfortunately, MSC/NASTRAN allows only one table to be defined for the complex stiffness values of viscoelastic materials as a function of frequency. For analysis, which includes more than one frequency dependent material, an easy-to-use procedure has been developed to consider this effect.

4. Examples for beaded and curved panels

In order to verify the accuracy of the methodology, two examples of a two-layer system have been calculated using the MSC/NASTRAN structural FE software and the results compared with a full multi-layer solid-element modelling approach. One is a beaded panel (Fig.2) and another a cutaway section of a sphere (a curved panel, Fig. 6). Figure 10 shows the locations for output of results of both examples.

4.1 A beaded panel

A beaded panel (Fig. 2) with clamped edges is loaded at its centre with a unit force in the normal direction of the panel. The steel layer has a thickness of 0.65 mm, and the damping layer 3.0 mm. The following four calculations are performed:

a) Bare (untreated) condition, i.e., neglecting the damping treatment, classical simulation using plate elements with Young's modulus of steel and estimated loss factor (η=0.002) [in figures below with legend **Bare**].

b) Single-layer equivalent simulation using plate elements with Young's modulus and loss factor obtained with SIMOKELL applied to both bending and membrane deformation [**Up-to-now**]. This is the method up to now.

c) Exact multi-layer simulation of the 2-layer system by modelling the damping layer with 3-D solid elements and the steel panel with plate elements [**Exact**];

d) Single-layer equivalent simulation using plate elements with separate Young's moduli and loss factors for bending deformation from SIMOKELL and for membrane deformation from the analytical approach [**Method here**].

4.2 A curved panel

A curved panel (a cutaway section of a sphere, Fig. 6) with clamped edges is loaded at its centre with a unit force in the normal direction of the panel. The steel layer has a thickness of 0.3 mm, and the damping layer 3.0 mm. Four calculations as described in the first example above are also performed.

4.3 Results and observations

In all the above calculations, direct frequency response analyses are performed with solution 108 in MSC/NASTRAN. Figures 3 and 7 (node 100) show the mobility curves for the beaded and curved panels in a frequency range from 5 to 500 Hz. For two other locations (nodes 101 and 102, see Figure 10), the results of frequency response functions for the four cases considered are presented in Figures 4 and 5 for the beaded panel and Figures 8 and 9 for the curved panel.

When considering specifically the shifts in natural frequencies, it becomes evident from Figures 3, 4 and 5 (beaded panel) that the damping treatment stiffens the panel considerably. Comparing the cases [**Bare**] and [**Exact**], upward shifts of 30 Hz to 100 Hz are observed. This stiffening is even more strong than shown by the frequency shifts when one bears in mind that the mass effect of the treatment alone would have shifted the modal frequencies down by approx. 25 %. For the curved panel (Figure 7), both upward and downward shifts of modal frequencies are observed. The lower modes seem to be mainly affected by the mass effect of the damping treatment whereas the higher modes perceive a much stronger stiffening.

Of particular interest is that the proposed method with separate data for bending and membrane deformations [**Method here**] gives more or less exactly the correct frequency shifts as calculated with the multi-layer solid-element approach [**Exact**] for all modes in both cases. So both the mass and the stiffening effects are correctly considered. The results are however highly unsatisfactory when it is neglected that membrane deformations are both less stiffened and less damped by the treatment than bending deformations, which is implicitly the case when the bending data are applied to all deformations of the panel [**Up-to-now**]. The stiffening can be strongly over-estimated as can be seen in the fundamental mode of the beaded panel. This effect becomes more and more pronounced when the degree of beading of the panel is increased. In other words, beading may stiffen a panel very positively, but at the same time the potential for reducing resonance amplitudes by applying damping treatment is reduced. This is in agreement with experimental observations that damping treatment is more effective in flat panel sections than in beaded areas.

It should be pointed out that the change of the stiffness due to damping treatment may have considerable contribution on global modes when a system includes such a panel. In addition, neglecting damping treatment and its effects [**Bare**] yields erroneous modal frequencies and unreasonably high response amplitudes.

Not only the modal shifts are correctly predicted by the proposed method. Also the levels of the response functions are nearly identical with the exact multi-layer solid-element approach. The deviations are usually less than 2 dB in the frequency range up to 500 Hz. This is in favourable contradiction to both the bare solution which yields far too high responses or to the approach using bending related data only [**Up-to-now**], which yields far too low levels.

The great advantages of the proposed method are:

1) The method is highly efficient and effective. The part model can be defined in terms of single-layer shell or plate elements and is thus easily to model. The number of active degrees of freedom is not increased in comparison to the body-in-white (BIW) models. In other words, the calculations need the same memory and CPU time as a BIW-analysis, but the different damping treatments on different panels are still properly included. This makes it possible to work with large and complex structures such as a complete vehicle body.

2) The treatment can be taken into account locally. Subsections of a vehicle body (or a main component such as a complete vehicle floor), which may be treated differently, are represented just by assigning different equivalent materials.

3) An optimization of treatment selection and treatment distribution is easy to perform using available optimization loops. The model remains always the same. The change from one treatment configuration to another is represented by a change of the related modification of the assignments of equivalent materials to the different subsections of the vehicle body or part.

5. Application to a real floor of a car *

In addition to the numerical verifications reported in the previous section of the paper, an experimental verification has been performed as well.

A floor segment was cut out from a car body by two vertical planes and two horizontal planes. Damped sheet metals (1.5 mm metal sheet with 3.0 mm damping) were welded to the cut lines in the front and the rear edges of the floor, and the holes at the A-pillar and B-pillar locations were also closed with damped sheet metal. This floor section was then been fixed by welding the side-sill weld flange on both sides with two thick metal plates (2.3 mm thickness, damped), which are "rigidly" connected to a heavy frame.

This floor panel was excited by a shaker located at the left reinforcement (side-member) of the floor underside. The excitation was in the vertical direction. The measurements were performed with a frequency bandwidth

* The data used are the results of a joint project with Daihatsu Motor Co., Ltd.

of 1.25 Hz up to 500 Hz. Third octave vibration measurements were performed on a large number of measurement points properly distributed along the surface. Figure 11 shows the vibration maps in dB and the distribution of the surface velocity of the floor in the case of 3.0 mm damping treatments.

Finite element meshes for the floor were generated by roughly cutting out the structure from a finite element model of the body-in-white. The model of the floor was then exactly cut as described above (Figure 12). Different damped metal sheets were modelled and connected to the cut lines with grid by grid conjunction according to the actual welding points. The boundary conditions of the whole model were defined by the two thick metal plates (2.3 mm thickness, damped) being clamped at the edges, and fixed to a heavy frame. The heavy frame is not included in the simulation model.

The model excited by a shaker has been computed by using the direct method in MSC/NASTRAN for the bare condition and with the damping treatments. Obviously, narrow-band simulation results had to be integrated over third-octave frequency bands in order to obtain data which are comparable to the measurement results. In the following, for example, only the simulation results for the 3.0 mm damping treatment case are presented. To compare with measurement, only results at the points where measurements were made have been shown in Figure 13.

A comparison of the simulation results (Figure 11) with those from the measurements (Figure 13) shows that they are in quite good agreement with respect to levels and spatial distributions. The simulation correctly predicted the velocity distribution on most of the floor surfaces, and the high velocities in both measurement and simulation results are located almost in the same places. The results allow the optimisation of treatment distributions only by using simulation. The target is thereby not to reproduce the quantitative values of the measurements exactly at such high frequencies with such a complicated model, but to predict relative changes for different treatments.

From the results above, the sound power levels in dB radiated by the floor can also be calculated and compared with measurements:

	250-315 Hz	315-400 Hz	400-500 Hz
Simulation [dB]	80.9	76.7	75.9
Measurement [dB]	78.3	79.5	78.9

The differences between simulation and measurement are less than 3 dB.

6. Conclusions

Damping treatment is a common and widely used method to reduce vibration and therefore noise radiation. The effects of damping treatments on the vibrational behaviour of panels are strong and can not be neglected in structural and acoustic analysis. The paper describes a simple and reliable procedure for simulating the vibrations of panels (non-flat) with damping treatments.

The two representative examples and the practical application have confirmed that the proposed method works well, is easily integrated into complex structural and acoustic analysis, yields accurate results and allows efficient and effective design optimization.

7. References

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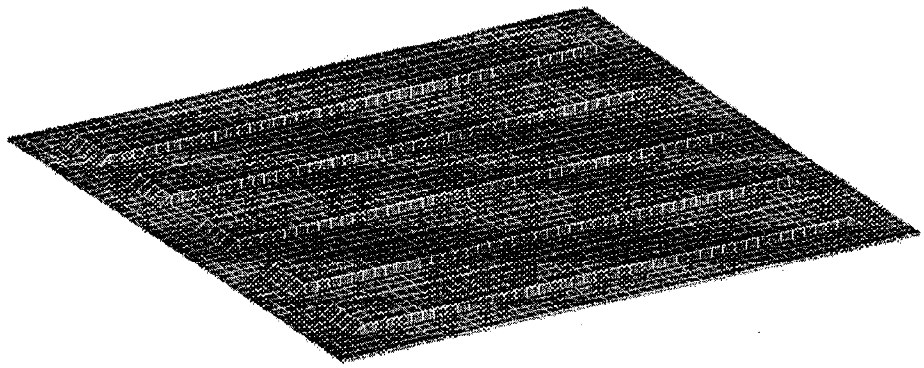

Figure 2 A beaded plate with damping treatment

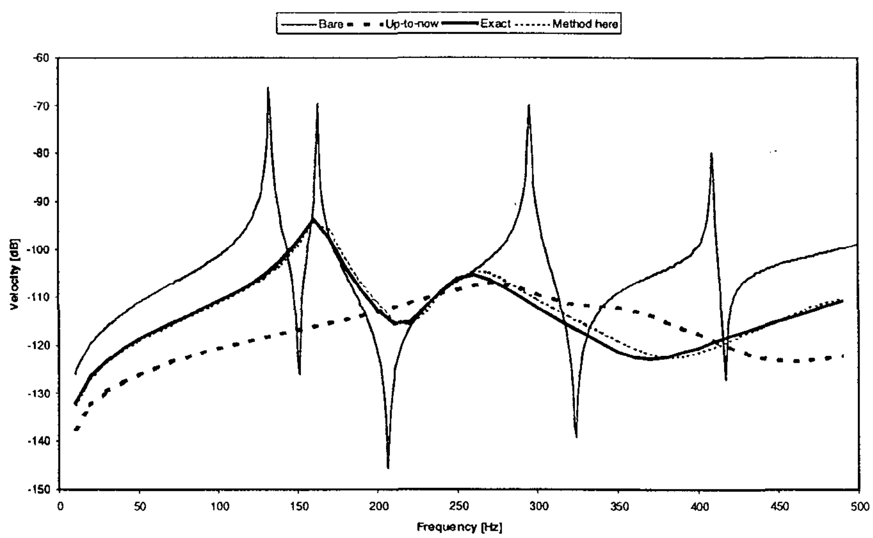

Figure 3 The velocity of a beaded plate at the centre [node 100]

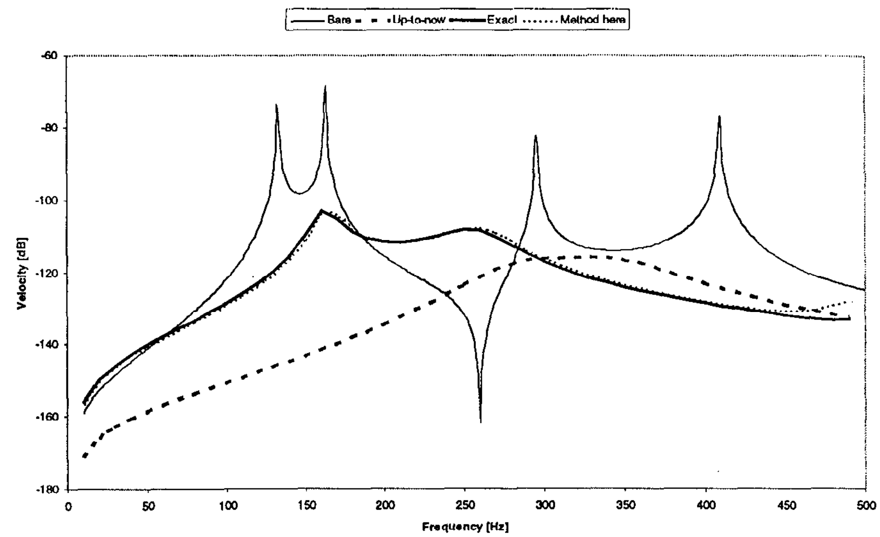

Figure 4 The velocity of a beaded plate at the corner [node 101]

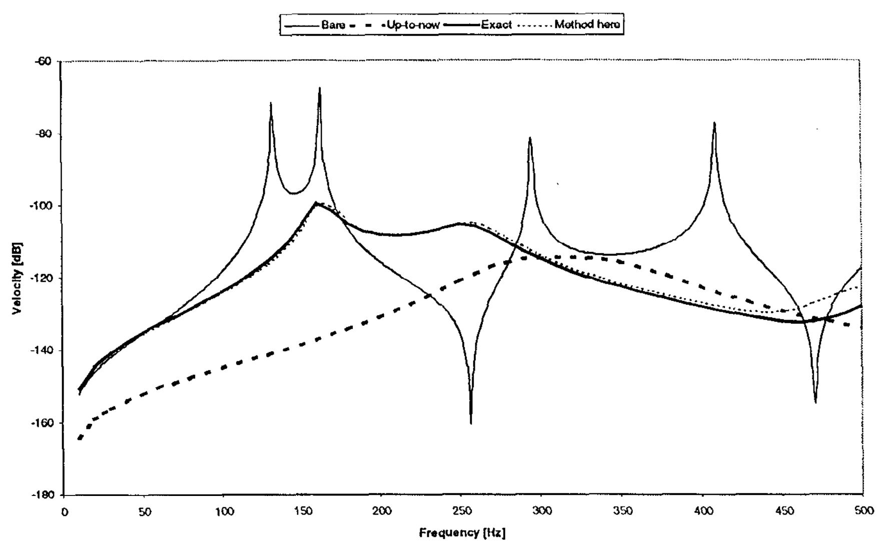

Figure 5 The velocity of a beaded plate at the edge [node 102]

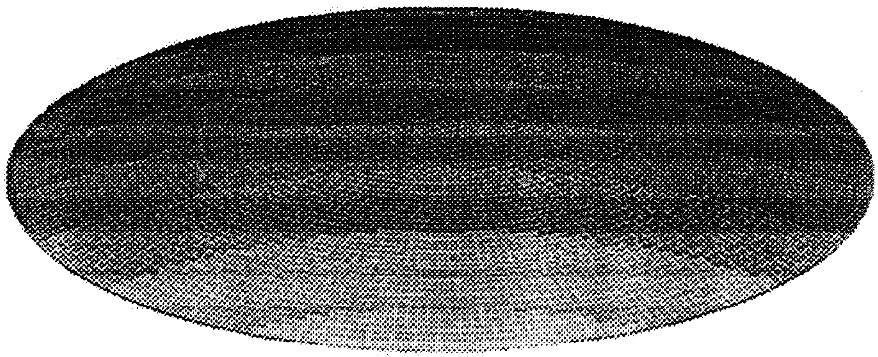

Figure 6 A cutaway section of a sphere with damping treatment

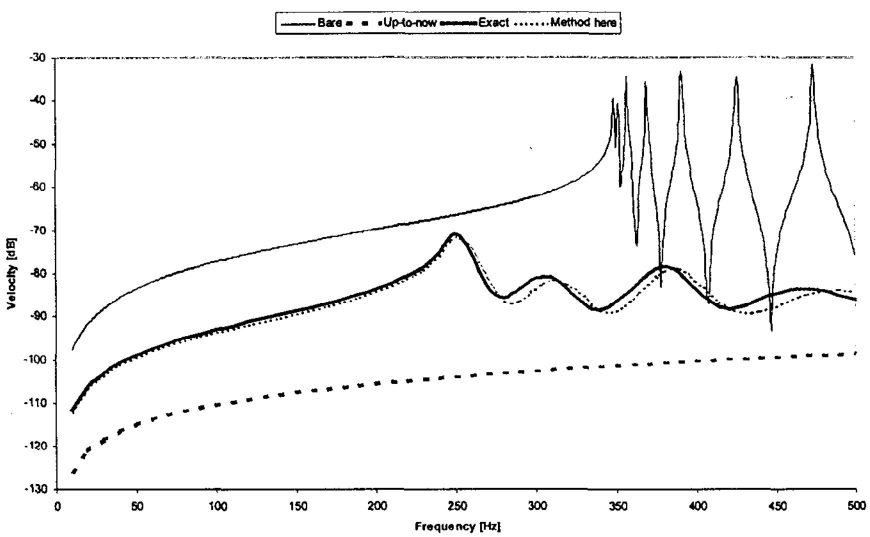

Figure 7 The velocity of a curved plate at the centre [node 100]

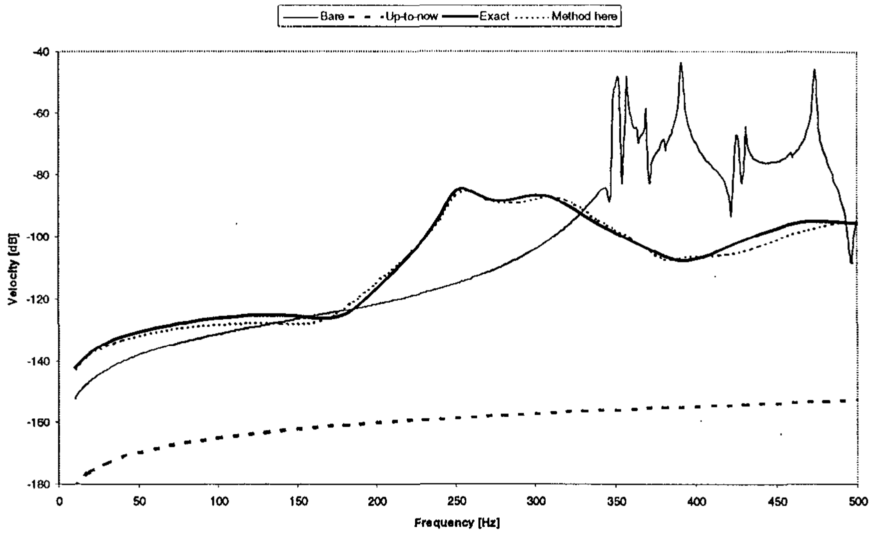

Figure 8 The velocity of a curved plate at the halfway [node 101]

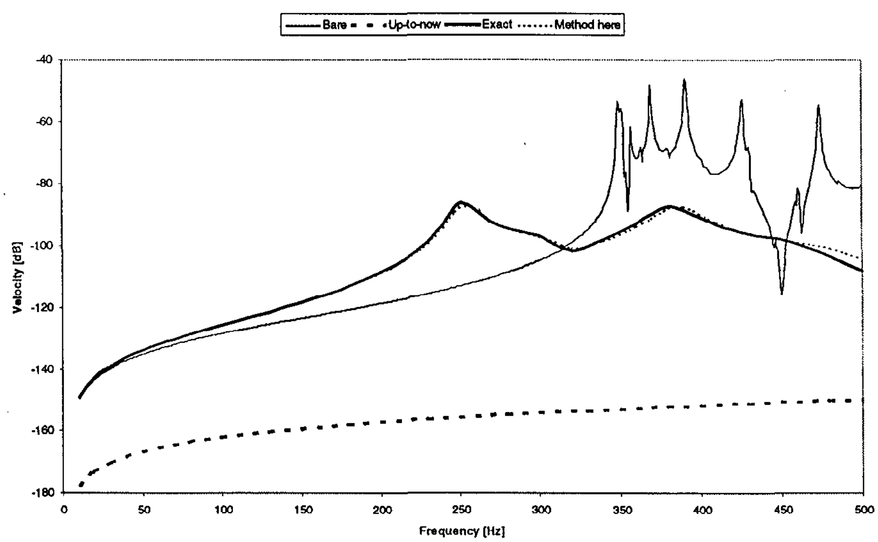

Figure 9 The velocity of a curved plate at the edge [node 102]

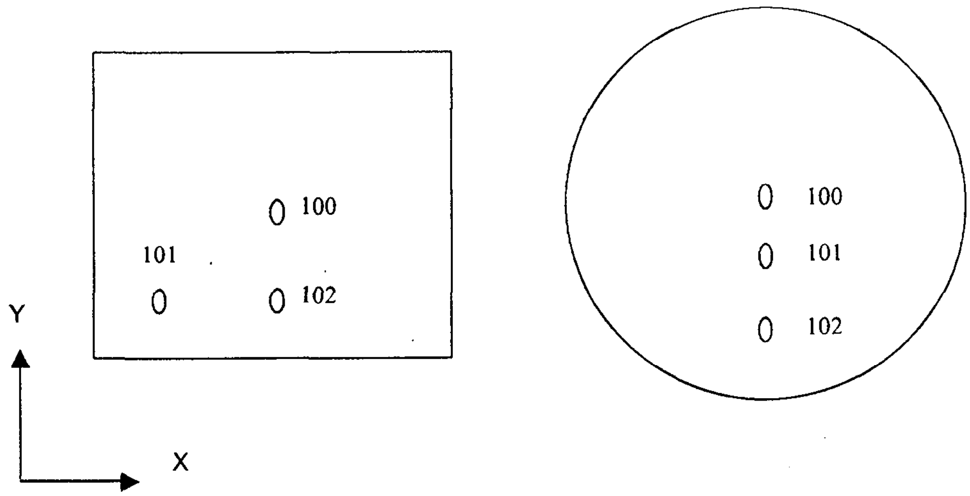

Figure 10 A sketch of output locations for both examples

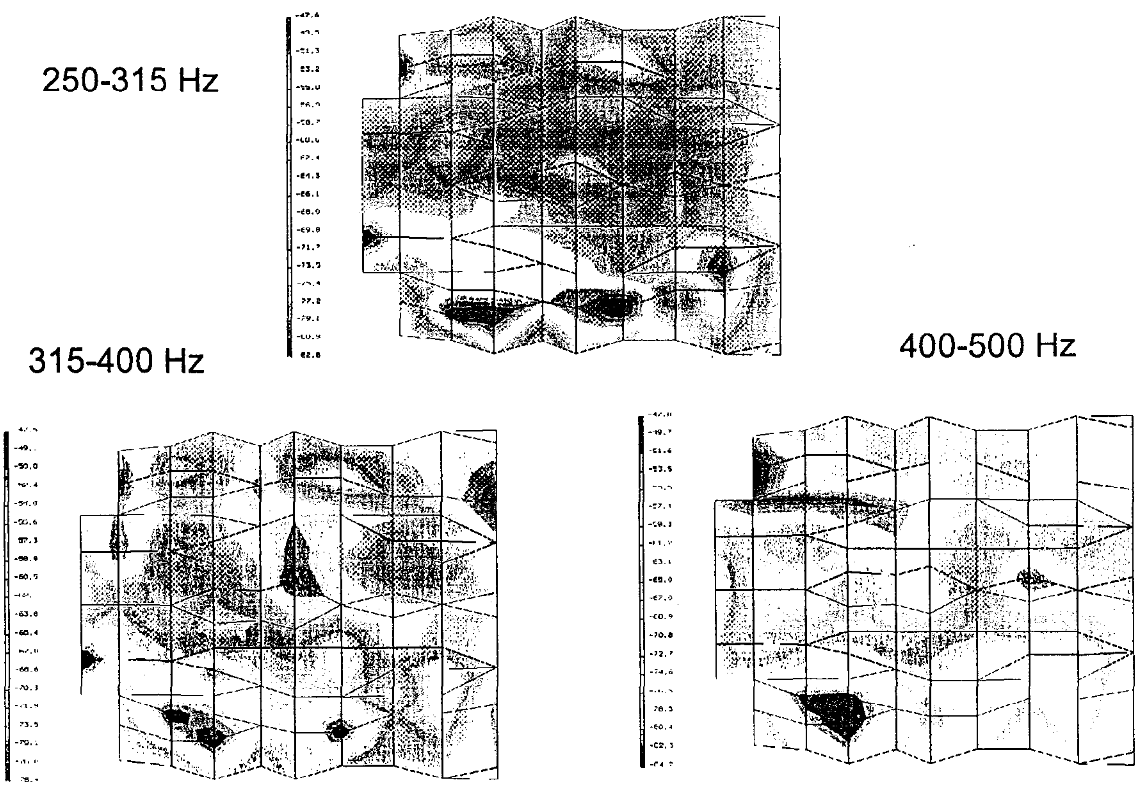

Figure 11 Velocity of floor surface in dB (measurement)

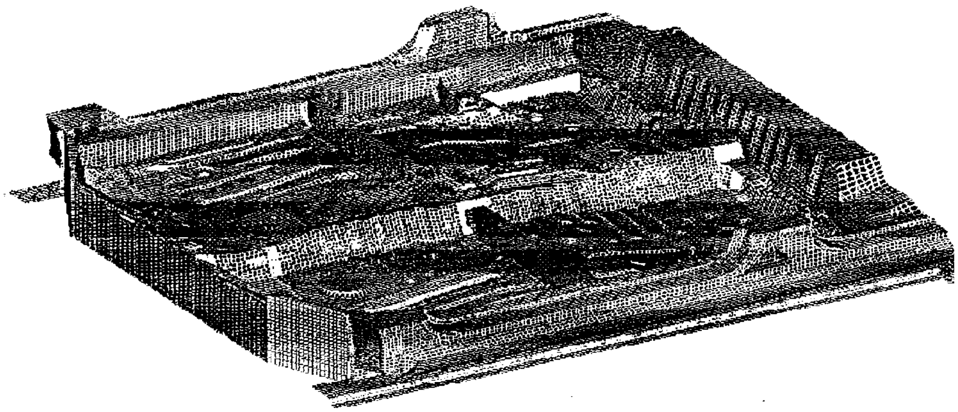

Figure 12 Simulation model of floor

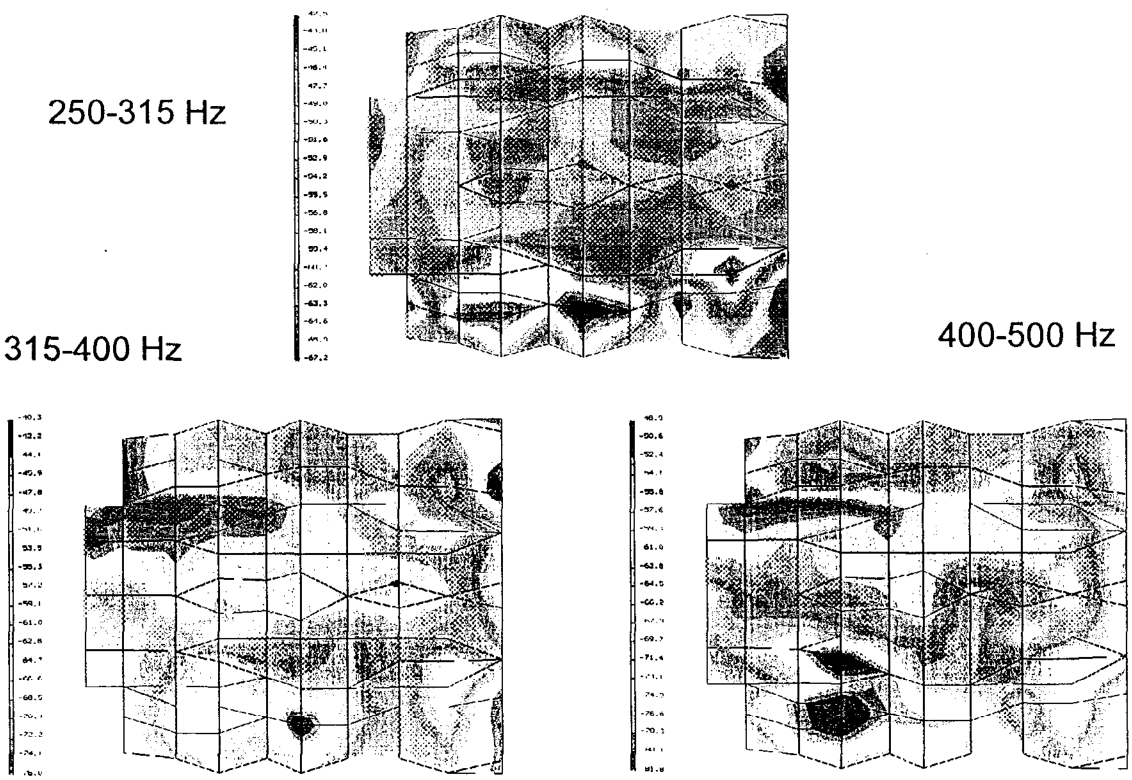

Figure 13 Velocity of floor surface in dB (simulation)