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PRACTICAL INDUSTRIAL METHODS OF INCREASING STRUCTURAL DAMPING IN MACHINERY

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INTRODUCTION

Industrial machinery is made of plates, shells and beams attached together either by welding or by bolting. The damping level of these connected structures is usually very high and often of the order of 0.01. However, the ringing vibrational responses are very much dependent on the damping of the structure. Therefore in order to reduce the sound radiation from these structures, damping level of the structure components has to be further increased.

Damping by attached viscoelastic layers has been used successfully for many years and can be very effective for light beams and plate-like structures. The method consists of attaching a layer of high damping viscoelastic material with a very high modulus of elasticity to the surface of the structure. However, the major set back of this kind of damping treatment is that it is expensive and sensitive to hostile environment. It is ineffective on the thick sections of typical machinery structures without resorting to impractically thick damping layers.

Vibration absorbers can also be attached onto the structure maximum displacement points to reduce the vibration. This system is ideal for structures having basically one dominant resonant frequency or a group of frequencies very close together. Usually the damper's effectiveness is limited to resonant frequencies concentrated in a band less than one octave. This type of spectrum is not, however, very common in machine structures. Impulsive forces and high modal densities lead to a uniform distribution of the vibratory energy throughout the audio spectrum.

Granular materials such as sand can also be used to increase the structure components loss factor. It is useful for high temperature application and where cavities of components can be filled. Small vibrational motion (low strain condition) leads to very little macroscopic slip between grains and the absorption of energy is within the grains themselves [1]. If the vibrational energy is increased (large strain), the energy dissipated at the surfaces of grains by friction predominates. Loss factors approaching 0.1 can be achieved over a broad frequency range. However, this method is practical only if the internal cavities are available for filling.

Another damping treatment, which is the squeeze-film damping method, can also be used to increase the structure components loss factor. This method is particularly useful to reduce the vibration of stiff machinery panels and the treatment will neither impede the working part of the machine nor will be damaged by hostile environment.

The following section will discuss the basic mechanism of the squeeze-film damping at work. The use of light fluid air and the heavy fluid oil in relation to the damping level will be discussed.

The squeeze-film damping mechanism with air

The damping is generated by attaching an auxiliary plate parallel to the surface of the machinery component, thereby trapping a thin layer of air; relative vibration of the plates pumps this air at high velocities resulting in energy loss due to the air viscosity. With reference to figure 1, the applied plate and the air are assumed to move with the same wavenumber k_p, of the free wave in the thick plate. By using an impedance approach and assuming that the applied plate has a higher wavenumber than the thick plate, the impedance of the applied plate $\bar{Z}_2$ is mass controlled.

By using the Navier-Stokes and continuity equations and setting $\bar{v}_2 = \underline{0}$, the impedance per unit area of the air layer for both compressible flow ($\bar{Z}_b$) and the incompressible flow ($\bar{Z}_b'$) are calculated [2]. The treatment impedance of the air layer and the applied plate can then be calculated as

$$\bar{Z}_t = \frac{\bar{Z}_2 \bar{Z}_b}{\bar{Z}_2 + \bar{Z}_b} \quad \text{and} \quad \bar{Z}_t' = \frac{\bar{Z}_2 \bar{Z}_b'}{\bar{Z}_2 + \bar{Z}_b'} \qquad (1a \;\&\; b)$$

These curves are shown in figure 2. As can be seen, at low frequencies $\bar{Z}_t'$ and $\bar{Z}_t$ are the same. The air is pumped back and forth as if incompressible. At the thick plate critical frequency, the velocity in the air is high and $\bar{Z}_t$ is a maximum Above the critical frequency, the air is compressed and $\bar{Z}_t$ becomes small.

The appropriate model of the system is shown in figure 3. The spring accounts for the compressibility of the fluid. The mass and dashpot account for the inertia and viscosity of the air. The loss factor of the system can be calculated as

$$\eta = \frac{\text{Energy dissipated/radian}}{\text{Kinetic energy of thick plate, air layer and thin plate}}$$

$$= \frac{\mathrm{Re}\ \{\bar{Z}_t\}}{\omega m_1 + \left|\frac{\bar{Z}_t}{\bar{Z}_t'}\right|^2 \mathrm{Im}\ \left\{\bar{Z}_t'\right\}} \qquad (2)$$

where m_1 is the mass per unit area of the thick plate.

Above the critical frequency the applied plate can vibrate with its own free bending wavenumber thus representing an additional source of energy storage and dissipation. The critical frequency of the attached plate is higher than that of the thick plate and so has larger loss factors at high frequencies. By using a two component SEA model, the loss factor of the coupled system can be calculated as $\eta_{tot} = (\eta_1 \epsilon_1 - \eta_2 \epsilon_2)/(\epsilon_1 + \epsilon_2)$, where ϵ_1 and ϵ_2 are the time averaged energies of the thick and applied plates, η_1, is the previously calculated thick plate loss factor, η_2 is the loss factor of the back plate due to squeeze-film damping, radiation and material damping. Figure 4 shows the comparison of the measured and the estimated loss factor and the agreement is good. Figure 5 shows that by further increasing the mass of the applied plate the high frequency performance is improved.

The squeeze-film damping mechanism with oil

Referring back to figure 1, it is assumed that the applied plate and oil layer move with the same wavenumber k, the combined fluid and free wave in the thick plate. This can be approximately calculated from

$$k^4 = \frac{\omega^2(m_1 + m_2 + \rho d)}{B} \qquad (3)$$

where m_1, m_2 are the mass per unit area of the thick and applied plate, ρ is the oil density, d is the fluid gap, and B is the flexural rigidity of the thick plate. The impedance per unit area of the applied plate (including the oil layer) is

$$\bar{p}/\bar{V}_2 = \bar{Z}_2 = i\omega\,(m_2 + \rho d) \qquad (4)$$

where $\bar{p}$ is the pressure and $\bar{V}_2$ is the velocity of the applied plate.

The impedance per unit area of the oil layer (assuming incompressible and of parabolic velocity profile), calculated by setting $\bar{V}_2 = 0$ can be shown equal to

$$\bar{Z}_b{}' = \frac{1}{k^2 d}\left[\frac{12\mu}{d^2} + i\omega\rho\right] \tag{5}$$

where μ is the fluid dynamic viscosity. This impedance has a mass and dashpot behaviour. The loss factor of the system is then calculated as

$$\eta = \frac{\mathrm{Re}\left\{\bar{Z}_t{}'\right\}}{\omega m_1 + \omega(m_2 + \rho d)\left|\frac{\bar{Z}_t{}'}{\bar{Z}_2}\right|^2 + \mathrm{Im}\left\{\bar{Z}_b{}'\right\}\left|\frac{\bar{Z}_t{}'}{\bar{Z}_b{}'}\right|^2} \tag{6}$$

Figure 6 shows the comparison of the measured and the predicted values including the radiation and material losses of the thick plate. The agreement confirms that there is little advantage of using oil alone to increase the loss factor. However, a further investigation shows that it is the ratio of the fluid dynamic viscosity and its fluid density which determines the level of the losses. It is possible to obtain high loss factor by using a high density fluid provided that the fluid viscosity can be greatly increased.

According to the above finding, a polyester foam with many small cells (average 1 mm diameter cell size) was immersed into the oil layer to increase the viscous losses. The measured loss factor shows a substantial increase compared to that with oil alone. Theoretical modelling of the impedance $\bar{Z}_b{}'$ of the foam-oil layer for two-dimensional consideration can be shown equal to

$$\bar{Z}_b{}' = \frac{1}{k^2 d}\,\frac{e\left[\frac{8\mu}{a^2} + i\omega\rho\right]}{g} \tag{7}$$

where a is the average radius of the cells in the foam, g is the porosity of the foam and e is the macroscopic structural factor ($e = \frac{1}{\cos^2\theta}$ with θ the inclined angle to the x-direction) which accounts for the tortuousity of the cells. The foam is not glued onto the plates and the shear stiffness of the foam is not considered.

For a very high porosity foam, equation (6) is still valid in calculating the total loss factor of the system. The loss factor is found to the extremely sensitive to three parameters a, ϵ and d [3]. Figure 7 shows the measured and predicted loss factor. The predicted values are calculated using $\epsilon = 1$ (horizontal cells) and the material and radiation losses of the thick plate are included. The agreement is encouraging.

EXPERIMENTAL WORK

ExperimE97 was conducted using 0.5 m x 0.5 m glass plates, 6 mm thick polyester high density open cell foam (density = 40 kg/m^3) and Essolube hdxplus 20 W/50 diesel engine oil. The loss factors were measured from the decay of third-octave noise.

CONCLUSIONS

High loss factors are obtained by making the air gap as thin as possible and making the applied plate as heavy and flexible as possible. By using the heavy fluid like oil, the increase in loss factor is insignificant if oil is used alone. However, high loss factors can be obtained by making the viscosity losses as high as possible by artificial means (e.g. putting foam into the oil layer).

REFERENCES

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[3] L.C. Chow and R.J. Pinnington, 'Theoretical and experimental investigation of oil layer damping of plates', To be published in the Proceeding of Inter-noise in July 1986, U.S.A.

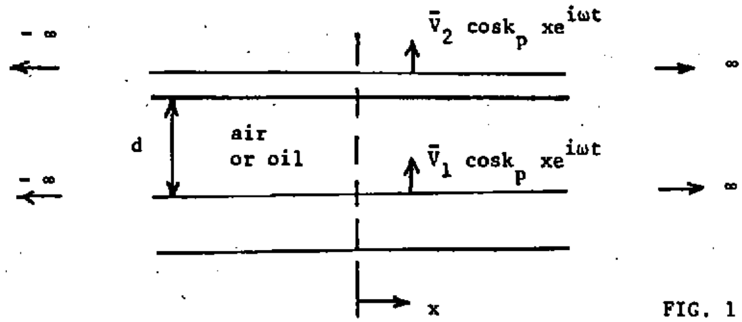

FIG. 1

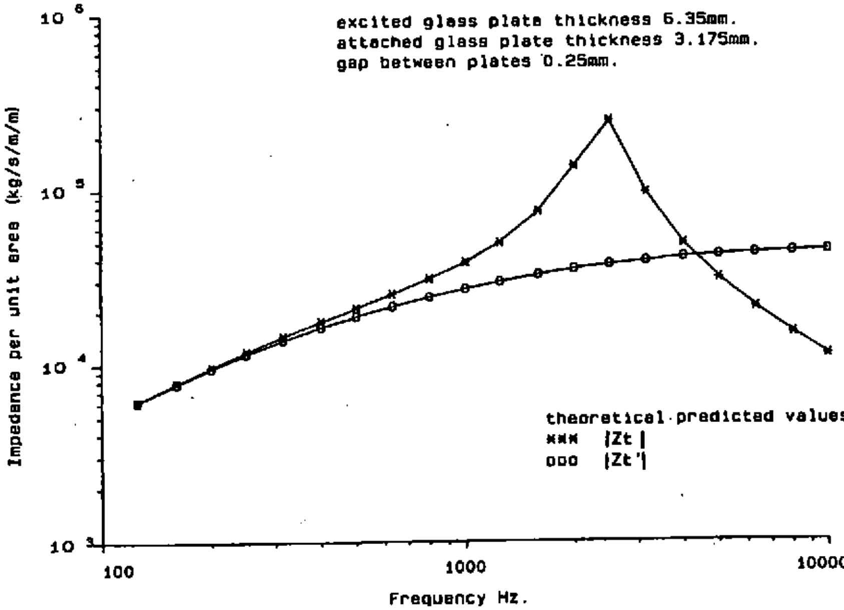

FIG. 2

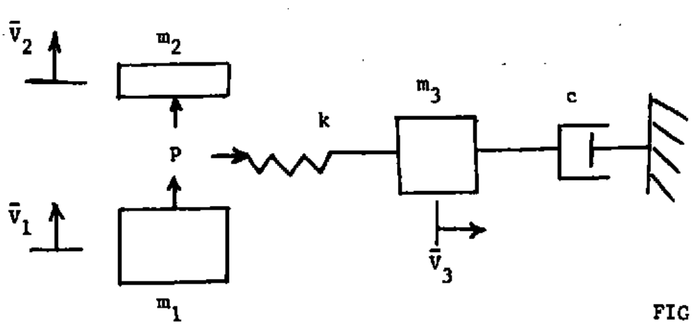

FIG. 3

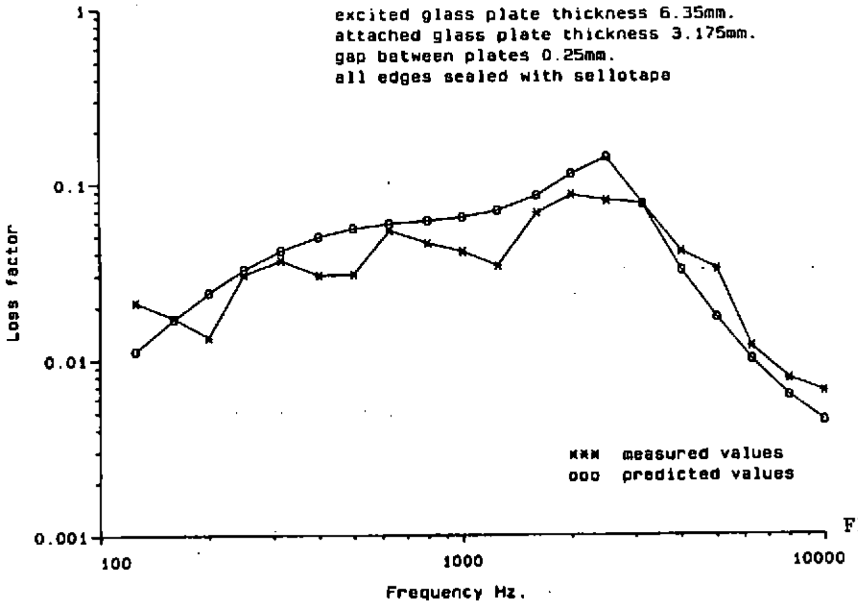
excited glass plate thickness 6.35mm.
attached glass plate thickness 3.175mm.
gap between plates 0.25mm.
all edges sealed with sellotape
xxx measured values
ooo predicted values
Loss factor
1
0.1
0.01
0.001
100
1000
10000
Frequency Hz.

FIG. 4

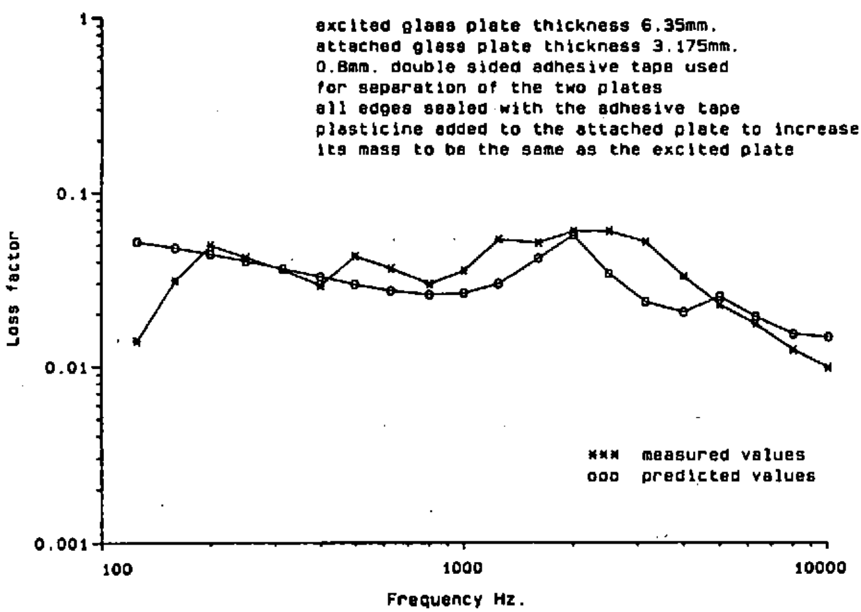
excited glass plate thickness 6.35mm.
attached glass plate thickness 3.175mm.
0.8mm. double sided adhesive tape used
for separation of the two plates
all edges sealed with the adhesive tape
plasticine added to the attached plate to increase
its mass to be the same as the excited plate
xxx measured values
ooo predicted values
Loss factor
1
0.1
0.01
0.001
100
1000
10000
Frequency Hz.

FIG. 5

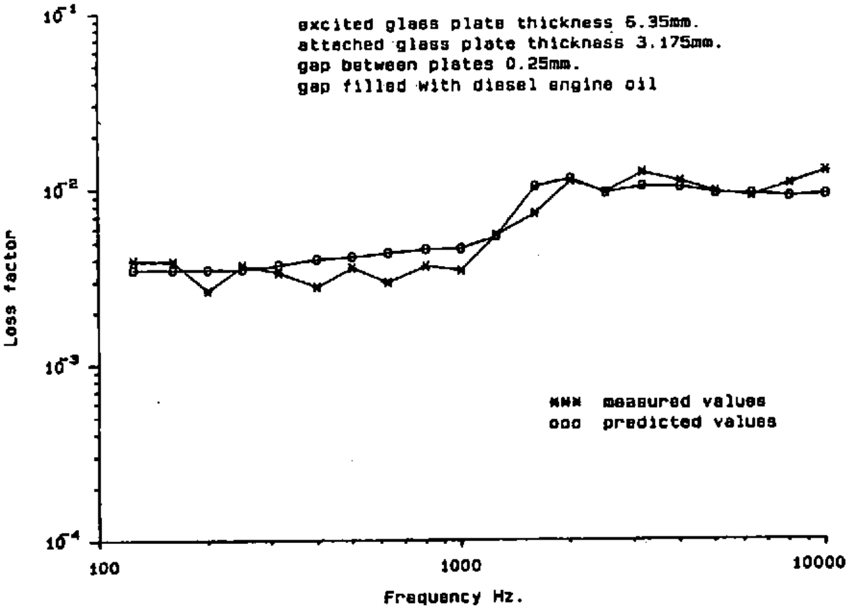
excited glass plate thickness 6.35mm.
attached glass plate thickness 3.175mm.
gap between plates 0.25mm.
gap filled with diesel engine oil
*** measured values
ooo predicted values
Loss factor
Frequency Hz.
100
1000
10000

FIG. 6

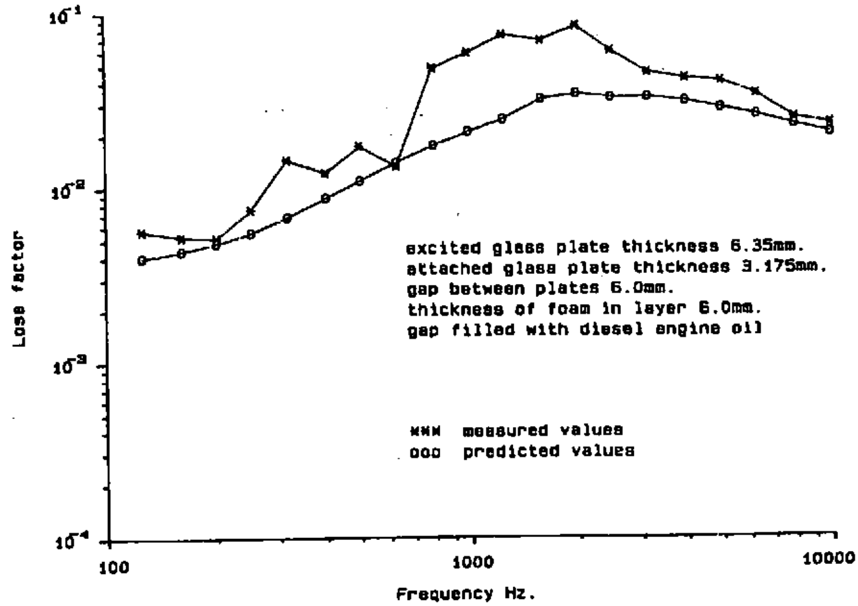
excited glass plate thickness 6.35mm.
attached glass plate thickness 3.175mm.
gap between plates 6.0mm.
thickness of foam in layer 6.0mm.
gap filled with diesel engine oil
*** measured values
ooo predicted values
Loss factor
Frequency Hz.
100
1000
10000

FIG. 7