Innovative Passive Viscoelastic Damper to Suppress Excessive Floor Vibrations

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ABSTRACT:

Due to changes in construction methods and new building layouts in recent years, human activities such as walking, dancing and aerobics have become major sources of floor vibrations. Excessive floor vibration is a cause of annoyance and discomfort to occupants. Excessive floor vibrations are realised after completion of construction or following structural modifications. Hence rectification measures would be required to minimise floor accelerations. This paper proposes a new innovative passive viscoelastic damper to reduce floor vibrations. This damper can be easily tuned to the fundamental frequency of the floor and can be designed to achieve various damping values. This paper discusses the analytical development of the damper and further experimental results are presented to demonstrate the effectiveness of the developed damper.

Keywords: Floor vibrations, viscoelastic damper, tuned mass dampers.

1. Introduction

Mechanical dampers can be installed more cheaply than structural stiffening and are often the only practical mean of vibration control in existing structures (Webster and Vaicaitis 1992). One form of mechanical damper is the Tuned Mass Damper (TMD) system as illustrated by Figure 1. In such systems the damper mass (m_2) and spring stiffness (k_2) are tuned to have the same natural frequency as the primary system. The addition of damping (c_2) to the TMD reduces the overall response of the combined system as shown in Figure 2. TMD systems are typically effective over a narrow frequency band and must be tuned to a particular natural frequency. They are not effective if the structure has several closely spaced natural frequencies and sometimes they increase the vibration if they are off-tuned (Webster and Vaicaitis 1992). A TMD splits the natural frequency of the primary
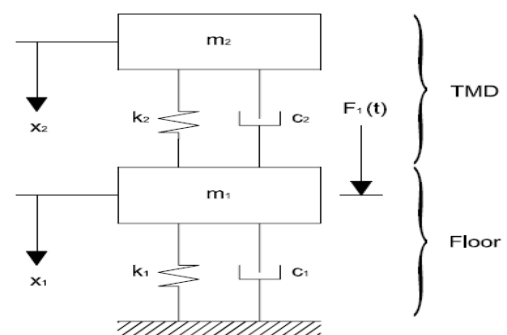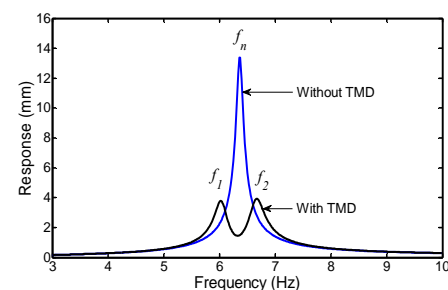

Fig. 1 Systematic two DOF

Fig. 2 Example showing the effects of attaching a TMD to SDOF system

system into a lower (f_1) and higher frequency (f_2) as shown in Figure 2. If there is zero damping then resonance occurs at the two undamped resonant frequencies of the combined system (f_1 & f_2). The other extreme case occurs when there is infinite damping, which has the effect of locking the spring (k_2). In this case the system has one degree of freedom with stiffness of (k_1) and a mass of (m_1+m_2). Using an intermediate value of damping such as ζ_{opt}, somewhere between these extremes, it is possible to control the vibration of the primary system over a wider frequency range (Smith 1988). In the optimum damper the values of damper's natural frequency and damping ratio (ζ_{opt}) are specified to obtain minimum and equal height peaks at f_1 & f_2 (Puksand 1975).

2. Viscoelastic Damper Design

Floor vibrations due to human excitation typically produce very small displacements (less than 1 mm). This small displacement would require a highly sensitive viscous damper to reduce the floor vibrations to the accepted level. It would be very difficult to produce a practical viscous damper that provides a reasonable level of damping with such small displacements and consequently an alternative equivalent viscoelastic damper has been investigated.

A common and effective way to reduce transient and steady state vibration is to add a layer of viscoelastic material, such as rubber, to an existing structure. The combined system often has a higher damping level and thus reduces unwanted vibration (Inman 1996). The simplest form of a viscoelastic damper is a constrained viscoelastic layer in a beam. This could be made of two constraining metal plates bonded together with high damping rubber as shown in Figure 3. In this composite
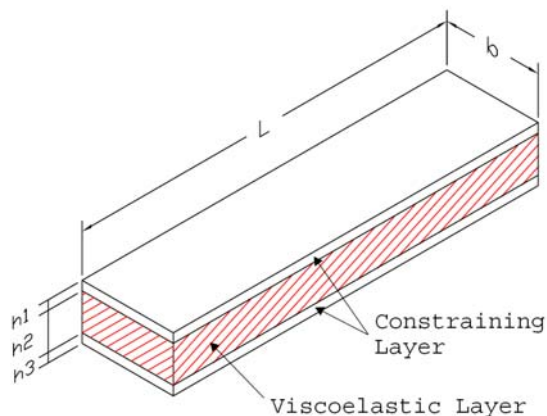

Fig. 3 Typical sandwich beam

sandwich beam, the viscoelastic material experiences considerable shear strain as it bends, dissipating energy and attenuating vibration response (Mace 1994).

There are many factors which affect the damping performances of viscoelastic materials including temperature, thickness and bonding. The damper discussed in this paper is for interior use so the variation in the temperature is not very significant whilst the resin used for bonding the layers must not allow slip to occur at the interface of layers. The two main variable factors to be taken into the account for the design of the damper are the viscoelastic material type and thickness.

An analytical method is used to estimate the overall dissipation loss factor of the composite system (η) based on the dissipation loss factor of the viscoelastic material (β), thickness of viscoelastic layer, geometric parameters and Young moduli of the top and bottom plates constraining the viscoelastic material. The analytical model considers that the core resists the shear but no direct flexural stress, shear strains in the face plates are negligible, transverse direct strains in both core and face plates are neglected and no slip occurs at the interfaces of the core and face plates (Mead and

Markus 1969). Boundary conditions have no significant effect on the loss factor and the following method can be applied to any composite beam configuration such as simply supported beam, cantilever etc. The overall loss factor of the composite system can be estimated by using Equation 1;

$$\eta = \frac{\beta g Y}{1 + g(2 + Y) + g^2(1 + Y)(1 + \beta^2)} \tag{1}$$

where β is the dissipation loss factor of the rubber and Y is a geometric parameter calculated as;

$$Y = \frac{(E_1 A_1)(E_3 A_3)d^2}{(E_1 A_1 + E_3 A_3)(E_1 I_1' + E_3 I_3')} \tag{2}$$

where E_1 and E_3 are the Young moduli of top and bottom constraining plates respectively, A_1 and A_3 are the cross-sectional area of the top and bottom constraining plates respectively, I_1' and I_3' are the moment of inertia of top and bottom constraining plates about their neutral axes respectively and d is the distance between the centroids of top and bottom constraining plates.

The shear parameter (g) can be written as;

$$g = \frac{Gb}{h_2 K_B^2}\left(\frac{1}{E_1 A_1} + \frac{1}{E_3 A_3}\right) \tag{3}$$

where G, b and h_2 are shear modulus, width and thickness of the damping layer respectively and K_B is the wave number of beam.

When the core shear stiffness is very low, the constraining layers dominate the flexural stiffness of the beam and the sandwich beam vibrates in the same mode as an Euler-Bernoulli beam (Mead and Markus 1970).

The wave number K_B is related to the flexural rigidity of the constraining layer (EI), mass per unit length (ρ) and natural frequency of the beam (ω) and for a system without an end mass the wave number can be calculated using Equation (4);

$$K_B^2 = \frac{\omega^2 \rho A}{EI} \tag{4}$$

For a system with a thick and thin constraining layer, it can be assumed that the thick layer dominates the flexural rigidity of the system and beam vibrates in the same mode of the thick layer. An expression was derived from first principles of Euler-Bernoulli beam for a cantilever beam with an end mass to calculate the wave number. The assumption used in the derivation of Equation 5 is that the slenderness ratio of the beam is ≥ 10;

$$\frac{m_{end}\left(Cos\left(K_B L\right)+Cosh\left(K_B L\right)\right)}{1+Cos\left(K_B L\right)+Cosh\left(K_B L\right)}=\rho A L + m_{end} \tag{5}$$

where ρ, A, L and m_{end} are mass density, cross-sectional area, length of the thick constraining layer and end mass respectively. Equation 5 can be numerically solved to obtain the value of the wave number. The flexural rigidity (EI_{total}) of the composite viscoelastic system can be calculated using Equation (6);

$$\frac{EI_{total}}{E_1 I_1' + E_3 I_3'} = 1 + \frac{gY\left(1 + g\left(1 + \beta^2\right)\right)}{1 + 2g + g^2\left(1 + \beta^2\right)} \tag{6}$$

The natural frequency of the viscoelastic cantilever beam damper can be estimated as;

$$f_d = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{7}$$

where k and m are the effective stiffness and mass of cantilever beam respectively and can be calculated as;

$$k = \frac{3EI}{L^3} \tag{8}$$

where EI and L are the flexural rigidity and length of composite viscoelastic cantilever beam respectively. The effective mass of a uniform composite cantilever beam can be calculated as;

$$m = \frac{33}{140}\rho A L + m_{end} \tag{9}$$

where ρ, A and L are density, cross-sectional area and length of composite cantilever beam respectively and m_{end} is a mss at the tip of the composite beam.

3. Optimum Viscoelastic damper

To illustrate the effectiveness of the proposed tuned mass viscoelastic damper, a prototype was built for use on a reinforced concrete T-beam. The 9.5m long beam is simply supported at the ends and simulates a proportion of a typical long span floor construction, with a cross-section shown in Figure 4. The properties of the bare T-beam are: effective mass m_1 = 3091 kg,

Fig. 4 RC T-beam cross-section

natural frequency f_n = 4.2 Hz and the damping ratio ζ = 2.88%. The properties of the optimum viscous damper with a mass ratio (μ) of 1% are m_2 = 30.91 kg and $\zeta_{opt.}$ = 6.02%.
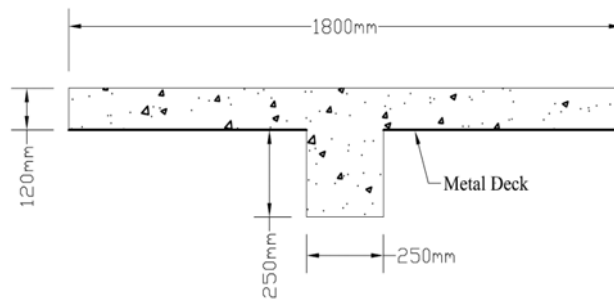
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A commercial rubber with a dissipation loss factor (β) of 0.2 was used to develop the viscoelastic damper. The geometry, natural frequency and dissipation loss factor of the damper were calculated using Equations (1-9) and the properties of damper are listed in Table 1. The dissipation loss factor of this rubber is not sufficient to provide the optimum damping ratio of 6% for the TMD with the thickness, width and length of the rubber and plates (a rubber with a higher dissipation loss factor would be needed to increase the damping). Table 2 shows a very good correlation between the predicted damper properties and experiment results whilst Figure 5 shows both the damper with the end mass and the time history response of the damper under a pluck test.

Table 1 Viscoelastic damper properties

Length (L)	510mm
Width (b)	100mm
Thickness of top constraining layer (steel)	6mm
Thickness of rubber (h_2)	38mm
Thickness of bottom constraining layer (steel)	6mm
Dissipation loss factor of rubber (β)	0.2
Rubber shear modulus	527331 N/m^2
End mass (m_{end})	30 Kg
Natural frequency of damper	4.1 Hz
Damping ratio (ζ) of damper	4.5%

Table 2: Validation of analytical model

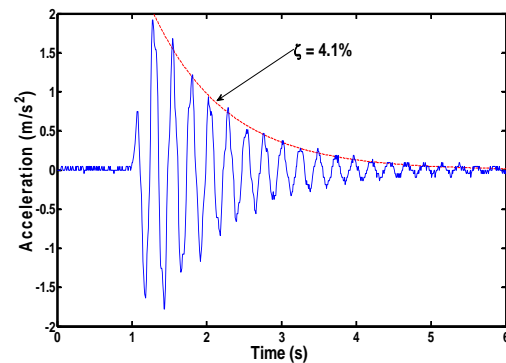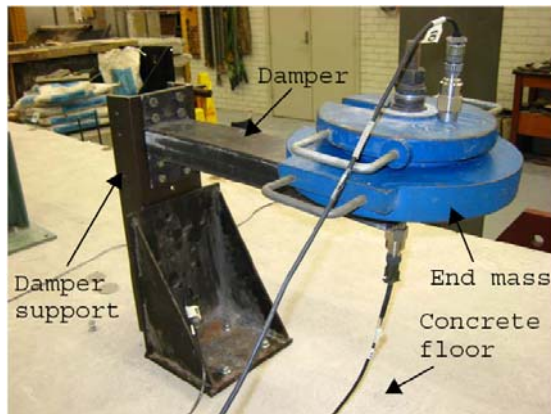| Parameter	Analytical Model	Experiment
Natural frequency of damper (f_d)	4.1 Hz Eq. 7	4.2
Damping ratio (ζ) ($\zeta = \eta/2$)	4.5% Eq. 1	4.1%

Fig. 5 Proposed three layers viscoelastic damper

4. Floor with Viscoelastic Damper Attached

The bare floor was excited by heel drops to measure the natural frequency, response acceleration and damping ratio. The viscoelastic damper was attached to the floor using a steel bracket as shown in Figure 5. The damper was tuned to the natural frequency of the floor of 4.2 Hz.

Table 3 Comparison of results

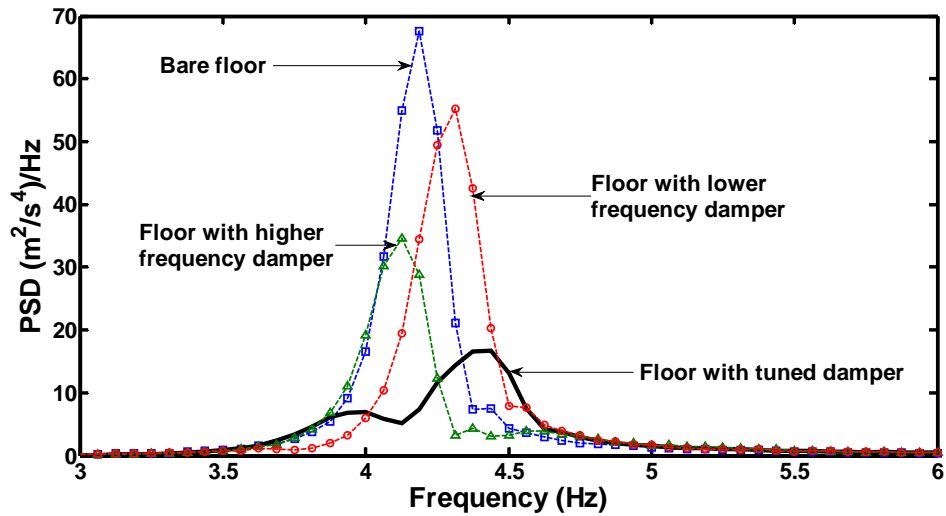| Measure	Analytical model	Experiment
Reduction in PSD	3.6	4
Damping ratio (ζ)	5.5%	6.1%

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Fig. 6 Floor response due to heel drops excitation

Figure 6 shows the power spectral density (PSD) due to the heel drop excitation for both cases with and without damper. The units of the PSD depend on the transformed data and for the acceleration Fast Fourier Transform (FFT) the units are $m^2/s^4/Hz$ i.e. square of the acceleration per Hz. Figure 6 shows that the reduction factor in PSD is about 4 which means the actual value of reduction in the acceleration is 2. Based on the heel drop test and idealising the response of the T-beam to a SDOF system, the overall damping ratio of the T-beam with damper was found to be 6.1% using the logarithmic method. Figure 7 clearly shows the increase of damping in the time history response of the T-beam whilst the experimental and analytical results were also found to be in good agreement as shown by Table 3.
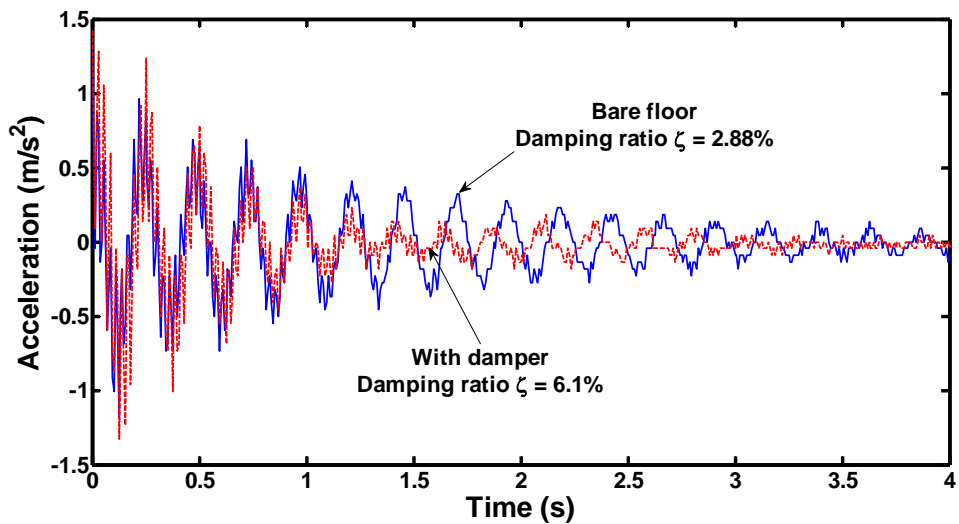

Fig. 7 Time domain decay of bare and with damper attached under heel drop excitation

The sensitivity of the viscoelastic damper to a variation in its natural frequency was also investigated. Figure 6 shows that the efficiency of the damper was considerably affected by increasing the frequency by 10% by modifying the end mass.

5. Concluding Remarks

This paper has presented a summary on the development of a viscoelastic tuned mass damper for floor applications. The proposed damper is based on the concept of a sandwich beam with the energy dissipation by shearing of a constrained rubber layer.

A prototype was built for use on a reinforced concrete T-beam with 9.5m span. An analytical model was used to predict the required optimum damper for this T-beam and the predicted properties of damper were in excellent agreement with the test result on the damper in terms of its natural frequency and damping ratio.

The T-beam was tested under heel drops and walking excitation with and without the viscoelastic damper. Without the damper, the T-beam had a measured damping ratio of 2.9% whereas the overall response was halved with a corresponding increase in damping to 6.1% with the addition of the TMD. The measured reduction in response due to the damper was in excellent agreement with the analytical prediction.

With success of this viscoelastic damper, the researches are currently extending the technology to a multi damper configuration.

6. References

Inman, D. J. (1996). Engineering Vibration, Prentice-Hall, Inc.

Mace, M. (1994). "Damping of beam vibrations by means of a thin constrained viscoelastic layer: Evaluation of a new theory." Journal of Sound and Vibration 172(5): 577-591.

Mead, D. J. and S. Markus (1969). "The Forced Vibration of a Three-Layer, Damped Sandwich Beam With Arbitrary Boundary Conditions." Journal of Sound and Vibration 10: 163-175.

Mead, D. J. and S. Markus (1970). "Loss Factor and Resonant Frequency of Encastre' Damped sandwich Beam." Journal of Sound and Vibration **12**(1): 99-112.

Puksand, H. (1975). "Optimum conditions for dynamic vibration absorbers for variable speed systems with rotating or reciprocating unbalance." The International Journal of Mechanical Engineering and Education. **3**: 145-152.

Smith, J. W. (1988). Vibration of structures application in civil engineering design, Chapman and Hall.

Webster, A. C. and R. Vaicaitis (1992). "Application of Tuned Mass Dampers To Control Vibrations of Composite Floor Systems." Engineering Journal/American Institute of Steel Construction Third Quarter/1992: 116-124.