Floor vibrations due to human excitation - damping perspective

***Ibrahim Saidi*[1]*, Nick Haritos*[2]*, Emad F Gad*[1 2]*, John L Wilson*[1]**

[1] Swinburne University of Technology

[2] University of Melbourne

Abstract

High levels of vibrations can occur in floor systems due to excitation from human activities such as walking and aerobics. In building floors, excessive vibrations are generally not a safety concern for building floor systems but a cause of annoyance and discomfort. Excessive vibrations typically occur in: (a) light weight floors; (b) floor systems with low stiffness where the floor dominant natural frequency is close to the excitation frequency; and (c) floors with low damping. While the floor mass and stiffness are normally constant during the life of the structure and can be estimated with a high degree of accuracy, damping is more difficult to predict because it is mostly associated with non-structural components such as partitions, false floors, suspended ceilings and ducts as well as furniture such as filing cabinets and bookshelfs. Current trends in the building industry associated with using lightweight materials, long-span open-plan floors and adoption of the electronic office, give rise to the importance of understanding floor vibrations and specifically damping. This paper provides a summary of factors affecting floor vibrations and discusses available damping systems which can be used to reduce vibration levels in floor systems.

Keywords: Floor vibrations, damping, human excitation, dampers.

Introduction

Annoying levels of floor vibrations due to human movements such as walking and running have become more common during the last two decades. The main factors contributing to this problem are a decrease in the floor mass resulting from the use of high strength building materials and composite systems; decrease in the floor natural frequency due to longer floor spans; an increase in the number of rhythmic human activities such as aerobics; and decrease in damping due to fewer partitions and items of furniture and other contributing factors (Setareh 2006).

Floors in office or apartment buildings are subject to the dynamic forces induced by people when they walk and occasionally, run, jump or dance. The latter three apply especially when an office building contains facilities such as running tracks on roofs, exercise rooms, dance floor or gymnasia. In corridors or long floors, running could be contemplated, but this will only occur in isolated instances (Bachmann et al 1995). Live loads are produced by the use and occupancy of a structure and in general, human live loads are classified into the two broad categories of in situ and moving. Periodic jumping to music, sudden standing of a crowd, and random in-place movements are examples of in situ activities whilst walking, marching, and running are examples of moving activities (Ebrahimpour & Sack 2005).

Human excitation

Occupants excite floors from their activities such as walking, dancing and jumping. Such forces are particularly problematic because they cannot be easily isolated from the structure and they occur frequently (Hanagan & Murray 1997). Walking pedestrians can induce considerable vertical and horizontal rhythmic impulsive dynamic loads that are dominated by the pacing rate. Typical pacing rates for walking are between 1.6 and 2.4 steps per second, i.e. 1.6-2.4 Hz (slow-fast walk) whilst for jogging the pace rate is about 2.5 Hz and running occurs at pace rates up to about 3 Hz (Collette 2004).

Although the load from pedestrians is dominated by the pacing rate, it also includes higher harmonic components caused by the impulsive nature of the load with frequencies corresponding to an integer multiple of the pacing rate. One pedestrian walking at a pacing rate of 2 Hz will therefore load the floor with a force composed of harmonic components at 2 Hz (1st harmonic), 4 Hz (2nd harmonic), 6 Hz (3rd harmonic), etc. A floor may be prone to resonance induced by pedestrian walking, if one or more of its natural frequencies are within the ranges 1.6-2.4 Hz (1st harmonic), 3.2-4.8 Hz (2nd harmonic) and 4.8-7.2 Hz (3rd harmonic). Higher harmonics components for walking seldom induce unacceptable vibrations. Since the annoying vibration amplitudes are caused by a coincidence of the natural frequency (f_n) of the floor with one of the harmonics of the walking excitation, the problem can be avoided by keeping these frequencies away from each other. This strategy is called High Tuning Method (HTM), which for a high damped floor system ($\zeta \geq 5\%$), the lowest f_n of the floor should be above the frequency range of the second harmonic (i.e. above 4.8 Hz) and for a floors with low damping ($\zeta \leq 2\%$), the lowest resonance frequency should be above the third harmonic (i.e. above 7.2 Hz). To allow for some scatter in the accuracy of estimating the parameters, $f_n \geq 7.5$ Hz should be targeted. This HTM is a simple and effective method for design and remedial measures but may be unnecessarily conservative since it does not take account of damping explicitly or the effect of a large participating mass. As a consequence, some floors with a fundamental frequency less than the 7.5 Hz criterion can perform quite satisfactory to walking (Bachmann et al 1995). On the other hand, composite floors with very low damping ($\zeta \leq 2\%$), can experience high levels of vibration even if their first natural frequency is above 7.5Hz (Haritos et al 2005).

The lowest natural frequency can be evaluated using a number of rational methods. AISC Steel Design Guide Series No 11 Chapter 3 details methods for estimating the natural frequency. For a concrete slab supported by simply supported steel joists, the natural frequency can be estimated by calculating the natural frequency for the beam or joist panel mode and for the girder panel mode separately and then combining the two using the Dunkerley relationship given by Equation (1):

$$\frac{1}{f_n^2} = \frac{1}{f_j^2} + \frac{1}{f_g^2} \qquad (1)$$

where f_j = beam or joist panel mode frequency and f_g = girder panel mode frequency (Murrary et al 1997).

Acceptance criteria for human comfort

The reaction of people who feel vibration depends very strongly on what they are doing. People in offices or residences are disturbed at peak acceleration of about 0.5% of the acceleration of gravity (g) whereas people taking part in an activity will accept acceleration levels 10 times greater (5% g or more) (Murrary et al 1997). People’s perception is also affected by the characteristics of the vibration response including frequency, amplitude and duration (Hanagan & Murray 1997). Figure 1 shows the recommended acceptable peak acceleration for different environments and their variation with frequency. Comfort studies for automobiles and aircraft have found that in the frequency of 5 to 8 Hz humans are especially sensitive to the vibration. This is explained by the fact that many organs in the human body resonate at these frequencies (Alvis 2001) whilst outside this frequency range, people accept higher vibration acceleration levels (Murrary et al 1997).

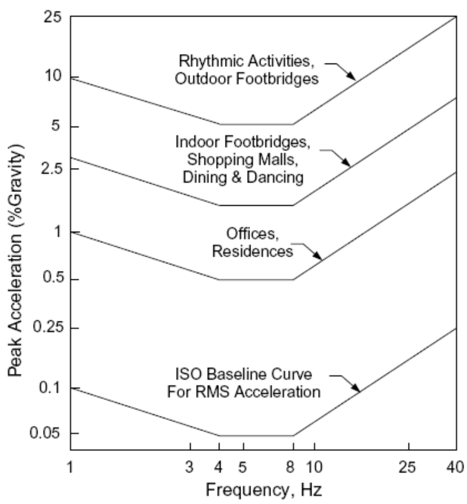

Fig 1: Acceptance Criteria

Determination of damping level

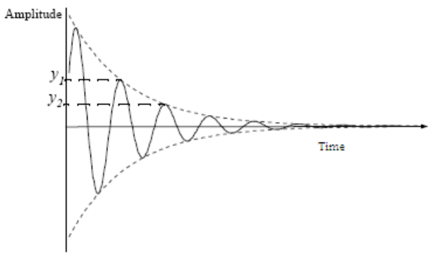

Fig 2: Logarithmic decrement method

Damping in a vibrating structure is associated with dissipation of mechanical energy, generally by conversion into thermal and sound energy. In most cases, the structural mass and stiffness can be evaluated rather easily, either by simple physical consideration or by generalised expressions. On the other hand, the basic energy-loss mechanism (damping) in practical structures is seldom fully understood; consequently it usually is not feasible to determine the damping coefficient by means of corresponding generalised damping expression. For this reason, the damping in most structural systems must be evaluated directly by experimental methods (Clough & Penzien 1975). There are different methods of estimating the damping ratio using either time or frequency domain analysis. Logarithmic Decrement Analysis (LDA) can be used in the time domain analysis and Half Power Bandwidth (HPB) can be used in the frequency domain analysis.

In the LDA analysis (see Figure 2), the decay in vibration amplitude (δ) which is defined as the natural log of the ratio of the size of two peaks, m cycles apart, can be estimated using Equation (2)

$$\delta = \frac{1}{m} \ln \frac{y_n}{y_{n+m}} \qquad (2)$$

where y_n is the amplitude of nth cycle and y_{n+m} is the amplitude of the n+mth cycle. The damping ration can then be found from Equation (3).

$$\zeta = \frac{\delta}{2\pi} \qquad (3)$$

The half-power bandwidth method (see Figure 3) is commonly used in estimating damping in the frequency domain. The dynamic Transfer Function which is defined as $T^2(f)$ is related to the spectrum for force, SF(f), as expressed by Equation (4):

$$T^2(f) = \frac{S_X(f)}{S_F(f)} = \frac{\chi_m^2(f)}{k^2} \qquad (4)$$

where χ_m (f) is the structure magnification function and k is the equivalent stiffness. The structure magnification function $\chi_m(f)$ for a single degree of freedom (SDOF) oscillator can be described in terms of the natural frequency f_n and damping ζ via Equation (5):

$$\chi_m^2(f) = \left[(1 - (\frac{f}{f_n})^2)^2 + (2\zeta \frac{f}{f_n})^2 \right]^{-1} \qquad (5)$$

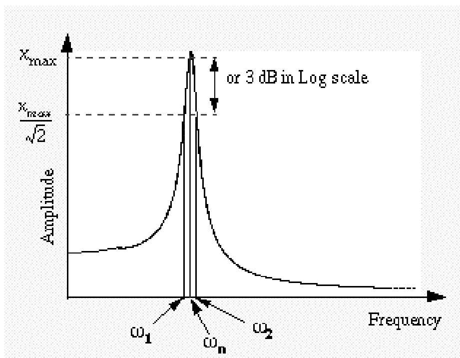

Fig 3: Half power bandwidth method

The half power bandwidth method uses the transfer function (or Frequency Response Function) trace of the structure to estimate the amount of damping for each mode. In this method, the transfer function amplitude of the system is obtained first. Corresponding to each natural frequency, there is normally a peak in the transfer function amplitude as shown in Figure 3 and at the root mean square (RMS) of the peak (ω_n) there are two points corresponding to the half power value (ω_1 and ω_2). The higher the damping, the larger the frequency range between these two points. Half-power bandwidth is defined as the ratio of the frequency range between the two half power points to the natural frequency at this mode. Haritos (1993) investigated an alternative optimised method to obtain the damping level. The "equivalent area" was tested and compared to the "peak value" in the frequency domain and half power bandwidth. The basic concept of the "equivalent area method" is to equate the area under the measured transfer function trace. The reason behind the use of this concept is by conducting such an integration the influence of "noisiness" is minimised because the integration is a form of smoothing operation.

$$A = \int_0^\infty \chi_m^2(f)\,\mathrm{d}f = \frac{\pi f_n}{4\zeta}\frac{1-2\zeta}{\sqrt{1-\zeta^2}} \approx \frac{\pi f_n\,(1-2\zeta)}{4\,\zeta} \approx \frac{\pi f_n}{4\zeta} \tag{6}$$

where A is the area under $\chi^2_m(f)$, $\chi_m(f)$ is the structure magnification function, f_n is the natural frequency and ζ is the damping. For light damped SDOF system the contribution made to the area under $\chi^2_m(f)$ is dominated by A_r, the area associated with the resonance bandwidth (i.e. $f_n/\sqrt{2} < f < \sqrt{2}\,f_n$) so that:

$$A_r = \int_{f_n/\sqrt{2}}^{\sqrt{2}f_n} \chi_m{}^2\ (\mathrm{f})\mathrm{df} \approx \frac{\pi f_n}{4\zeta}\frac{1-2\zeta}{\sqrt{1-\zeta^2}} \tag{7}$$

A_r can be determined by using standard numerical integration such as Simpson's rule. Haritos (1993) used a Monte Carlo style simulation to identify the statistical characteristics of predicted damping levels of a SDOF. The equivalent area method is considered more than satisfactory for determination of damping levels below about 8%.

The accuracy of the estimated level of damping may vary depending on the prediction method. The accuracy is influenced by number of factors in particular the "noisiness" of the data. It is reported that the equivalent area methods produces sufficiently accurate estimates for system with low damping (Haritos 1993).

Dampers

Although structural engineers have some design guideline for evaluating floor vibration before construction, there are still many floors that exhibit excessive vibrations. There are few options available to correct a floor with excessive levels of vibration. The relocation of the vibration source is the cheapest corrective method such as placing the vibration source (eg a gym) on the ground slab or placing sensitive equipment near columns or walls where the vibrations are less severe than at mid-bay (Koo 2003). Increasing the floor stiffness can reduce human induced vibration because it increases the natural frequency of the floor; however, in many instances there is physically not enough space to introduce new structural elements. Adding mass can reduce the vibration level but in most cases it is not practical as it may create overstress in structural members. Adding nonstructural elements such as full-height partitions with the aim of increasing damping and stiffness in most cases is not possible due to architectural requirements (Setareh 2006). Passive, semi active and active dampers can be used effectively to reduce excessive vibrations. Mechanical dampers can be installed more cheaply than structural stiffening and are often the only practical mean of vibration control in existing structures (Webster & Vaicaitis 1992).

Passive Dampers

Tuned Mass Dampers (TMD) and viscoelastic materials represent typical passive dampers. The first use of TMD for floor vibration application was reported by Lenzen who used small TMDs with a total mass of about 2% of the floor mass. The TMDs were made of steel hung by springs from the floor beams and dashpot to provide damping. Lenzen reported floors with annoying vibration characteristics became entirely satisfactory by tuning the TMDs to a natural frequency of about 1.0 Hz less than that of the floor and using a damping ratio of 7.5% (Setareh 2006).

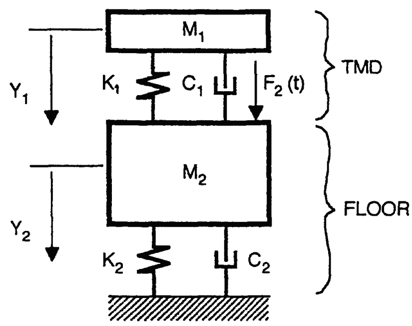

Fig 4: Two DOF (TMD)

Generally, a modern TMD consists of a mass, spring, and dashpot, as shown in Figure 4, and is typically tuned such that when large levels of motion occur, the TMD counteracts the movements of the structural system. The terms M_1, K_1, C_1, Y_1 represent the mass, stiffness, damping and displacement of the TMD, while M_2, K_2, C_2, Y_2 represent the mass, stiffness, damping and displacement of the floor and $F_2(t)$ represents the excitation force. As the two systems move relative to each other, the passive damper is stretched and compressed, reducing the vibrations of the structure by increasing its effective damping. 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 & Vaicaitis 1992). The natural frequency f_n of a TMD and floor can be obtained from Equation (8):

$$\omega_n = 2\pi f_n = \sqrt{(k/m)} \qquad (8)$$

where k = stiffness and m = mass. The optimum damping ratio (ζ_{opt}) of the vibration absorber (TMD) corresponds to

$$\zeta_{\text{opt}} = \sqrt{\frac{3(m_1/m_2)}{8(1+m_1/m_2)^3}} \qquad (9)$$

It should be noted that one TMD can only damp one mode of vibration. If damping of several modes is necessary the arrangement becomes quite complex (Backmann et al 1995).

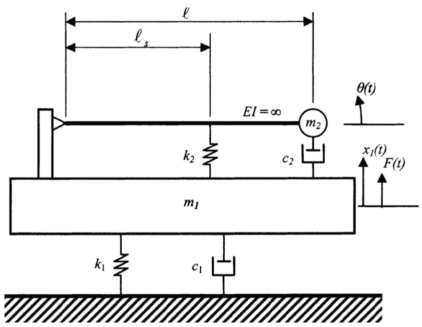

Fig 5: PTMD

The Pendulum Tuned Mass Damper (PTMD), as shown in Figure 5, is an innovative new TMD designed and manufactured by ESI Engineering. The mass is provided by steel plates distributed along the PTMD arm. This is done to minimise the PTMD vertical dimension such that it can be installed within the floor plenum. The springs are movable along the PTMD arm, so that the PTMD natural frequency can be fine-tuned and the dampers are attached to the end of the PTMD arm to maximise the damping force. PTMDs are in general tuned to a set of floor dynamic parameters and therefore if these parameters change over time the PTMD can become off-tuned and not be able to reduce the floor vibrations effectively. The main source of off-tuning is variations in the floor mass which is mainly due to the fact that the weight on the floor changes with variation in live loading over time (Setareh 2006).

Damping using visco-elastic materials: Visco-elastic materials (VEM) offer the advantage of reducing vibrations over a broader range of frequencies compared with TMDs. However, similar to TMDs, visco-elastic damping works optimally only for a specific mode of vibration. Nevertheless use of VEMs is a cheap method of increasing the damping if incorporated during construction (Ljunggren 2002).

An example of visco-elastic damping is the Resotec product which was developed by Arup in collaboration with Richard Lees Steel Decking to provide additional damping to modern composite floor construction. The Resotec system improves the dynamic performance of composite floors by dissipating energy through shearing of the visco-elastic damping layer during low-level vibrations. This product as shown in Figure 6 comprises a thin layer of high-damping visco-elastic material sandwiched between two thin steel plates; the overall thickness of the product is about 3mm. Resotec is placed on top of the top flange of a steel beam for a proportion of the beam near each end. The steel decking is placed normally over the beam (on top of the Resotec) and shear studs are fixed in the central zone of the beam only. The concrete slab is then cast in the usual manner. In the completed floor the visco-elastic layer is effectively sandwiched between the steel beam and the concrete slab to create a constrained layer damping mechanism.

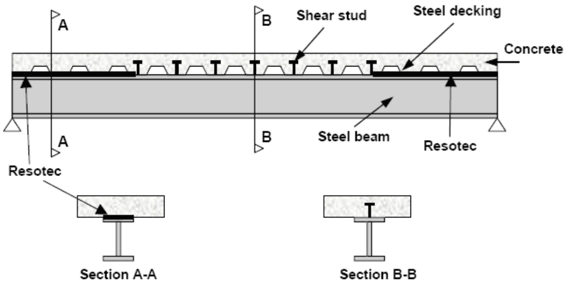

Fig 6: Resotec product installation

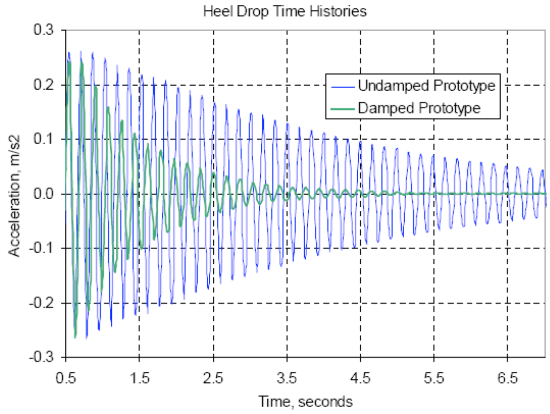

Fig 7: Performance of Resotec product

The steel beam is therefore fully composite with the floor slab only over a portion of its length centred at midspan. The product could be provided over the entire length of the beam (which would develop a large amount of damping), but this would make the entire beam non-composite, which would adversely affect its strength and stiffness. The effectiveness of Resotec is sometimes limited by the floor layout. The system works best for regular layouts where identical secondary beams have the same parallel lines of support. Where the ends of the beams are staggered due to curved or angular edges to the floor, composite and non-composite sections of adjacent beams are positioned next to each other, and constrained layer damping will be less effective. Example acceleration traces (recorded at mid-span) for two prototypes with and without Resotec are shown in Figure 7. It is reported that the damping of a fitted out floor is typically doubled by the incorporation of Resotec (Willford et al 2004). However, this product needs to be incorporated within the floor, during construction.

Semi-active control dampers

During the 1980s, the automotive industry researched, developed and tested various types of semi-active shock absorbers. That research produced a new type of control actuator that has applications in civil, mechanical, and aerospace engineering. The term semi-active describes a system that consists of a variable actuator that requires very little power to operate. The power required for the semi-active dampers (SADs) is that necessary to modulate the valve position and is typically many orders of magnitude less than that required to achieve a similar performance by fully active dampers (FADs).

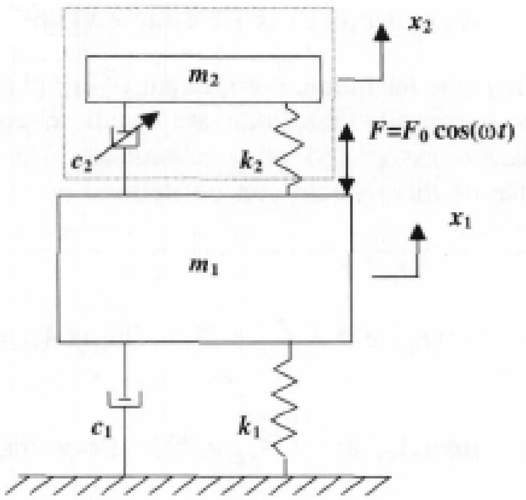

Fig 8: GHTMD

Setareh (2002) and Koo et al (2004) reported the use of a class of semi-active tuned mass dampers called ground-hook tuned mass dampers (GHTMD) as shown in Figure 8 which comprises a TMD with the ground hook semi active damper as a damping element. A magnetically responsive fluid damper can be used for this purpose which is a suspension of micron-sized, magnetizable particles in a carrier fluid. Altering the strength through the application of a magnetic field precisely controls the yield stress of the fluid. The alteration of the inter-particle attraction, by increasing or decreasing the strength of the field, permits continuous control of the fluid's rheological properties. Based on analytical studies, it was demonstrated that the GHTMD is more effective than its equivalent passive counterpart (for the same mass), in reducing the level of displacement when subjected to harmonic force excitation. Specifically, it was found that GHTMD can outperform its equivalent TMD by about 14% (Koo et al 2004).

Active control dampers

Hanagan and Murray (1995) developed an active electro-magnetic actuator that uses a piezoelectric velocity sensor and a feedback loop to generate control forces effectively adding damping to the supporting structure. Significant results were obtained on an office floor and a chemistry laboratory although high initial costs, maintenance, reliability, and the number of actuators needed to effectively reduce vibration levels were issues that were noted with this system. An actively controlled mass provides a larger degree of control compared with a passive device with an equivalent reactive mass. The active system is also less disruptive to the building function than most other repair measures. The active device is compact and can be installed with relative speed and ease in the ceiling cavity available in most commercial buildings. There are also disadvantages to the active control scheme. The cost of the components to provide a single control circuit was reported to be high with the hardware components costing about US$21,000 for a single control circuit. Maintenance and reliability issues also detract from the attractiveness of an active system, however as the technology advances, the cost will reduce (Hanagan & Murray 1997).

Concluding remarks

This paper has presented a summary of factors affecting floor vibrations in buildings with a particular focus on damping. Low damping is one of the primary causes of excessive floor vibrations in buildings. While designers have accurate models and tools to predict strength and stiffness, estimation and calculation of damping can be more difficult.

This paper has also reviewed passive, semi-active and fully-active dampers for floor applications. A passive tuned mass damper (TMD) can be effective in reducing floor vibrations if it is well tuned to the natural frequency of the floor. However, its effectiveness can quickly diminish or even exacerbate the problem if the TMD is "off tuned". As a form of passive damper, visco-elastic materials can be very effective because they can cover a wider range of frequencies compared to TMDs. However, such materials must be incorporated during construction and the floor must be designed to account for the reduced composite action between the slab and beams. Semi-active tuned mass dampers can be more effective than TMDs but an actuator requires power to modulate the fluid flow through the valves. Fully active dampers offer greater flexibility and can be more efficient than the passive and semi active tuned mass dampers, but they require significantly higher initial cost and on going maintenance of their associated electronic and power systems.

There is a demonstrated need for the development of simple passive multi tuned mass dampers for retrofitting applications. By using multi dampers, several modes of vibrations can be treated. Furthermore, having a distributed system would result in the individual units being physically smaller in size. Finally, a multi damper system may be more accommodating if one specific damper is off-tuned due to changes to the floor such as those associated with redistribution of live loads.

References

Alvis S, 2001, "An experimental and analytical investigation of floor vibrations", partial fulfilment of Master thesis submitted to Virginia Polytechnic institute and State University and approved by Thomas Murray et al.

Bachmann H et al, 1995, "Vibration Problems in Structures: Practical Guidelines", Birkhäuser Verlag, Berlin.

Clough R and Penzien J, 1975, Dynamic of Structures, McGraw-Hill.

Collette F, 2004, "Comfort and vibrations on floors due to walking loads", COWI Publications, The Scandinavian Vibration Society (SVIB), Stockholm.

Ebrahimpour A and Sack R, 2005, "A review of vibration serviceability criteria for floor structure", Computers & Structures Vol 83, Issues 28-30; Nov pp 2488-2494.

Hanagan L and Murray T, 1995, "Active control of floor vibration: implementation case studies", Proceedings of the 1995 American Control Conference, 21-23 Jun pp 1911-1915.

Hanagan L and Murray T, 1997 "Active control approach for reducing floor vibrations", Journal of Structural Engineering; Nov pp1497-1505.

Haritos N, 1993, "The 'Equivalent Area' Method for Estimating Damping Levels", Proceedings of the 13th Australasian Conference on the Mechanics of Structures and Materials (ACMSM13), Wollongong; July pp 341-348.

Haritos N, Gad E, Wilson J, 2005, "Evaluating the Dynamic Characteristics of Floor Systems using Dynamic Testing", Proceedings of ACAM2005, Melbourne, Australia, pp 225-230.

Koo J, 2003, "Using Magneto-Rheological Dampers in Semiactive Tuned Vibration Absorbers to Control Structural Vibrations", Dissertation submitted to the Faculty of the Virginia Polytechnic Institute and State University.

Koo J.H et al, 2004, " In Search of suitable control methods for semi-active tuned vibration absorbers", Journal of Vibration and Control Vol 10 No. 2 pp 163-174.

Ljunggren F, 2002, "Development of a new damper to reduce resonant vibrations in lightweight steel joist floors", Doctoral thesis "Floor Vibration-Dynamic properties and subjective perception, Division of Sound and Vibrations, Lulea University of Technology, Sweden.

Murray T et al, 1997, Floor Vibrations Due to Human Activity, AISC Design Guide No. 11.

Setareh M, 2002, "Floor Vibrations Control Using Semi-Active Tuned Damper", Canadian Journal of Civil Engineering Vol 29 No. 1; Feb pp. 76-84.

Setareh M, 2006, "Pendulum Tuned Mass Dampers for Floor Vibration Control", Journal of Performance of Construction Facilities; Feb pp 64-73.

Webster A and Vaicaitis R, 1992, "Application of Tuned Mass Dampers to control vibration of composite floor dampers", Engineering Journal AISC, third quarter, pp 116-124.

Willford M et al, 2006, "A constrained layer damping system for composite floors", The Structural Engineer, Vol 84 issue No. 4; Feb.