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STUDY OF A MULTIPLE-TUNED MASS DAMPER FOR EXCESSIVE FLOOR VIBRATIONS MITIGATION

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Excessive vibration of architectural structures due to human movements has become an important serviceability issue. Tuned Mass Dampers or Vibration Absorbers have been used to mitigate the problem. This paper presents studies on an easy to fabricate, assemble and tune Multiple Tuned Mass Damper (MTMD) that can alleviate vibrations due to the excitations of several modes of a floor system. Using a test floor at the Virginia Tech Vibration Testing Laboratory, a number of experiments were conducted with the MTMD active and inactive. Different excitations generated by an electrodynamic shaker, and people walking on the floor were used. The vibration tests were repeated at various times to check the consistency of the MTMD performance. The measured reduction in floor vibrations were compared to those predicted based on the analytical equations. The results have shown that the device is effective in reducing floor vibrations due to human movements in spite of its small mass compared to the mass of the test floor.

Keywords: Multiple Tuned Vibration Absorber (MTVA), Vibration serviceability, Vibration control, Civil engineering structures, Walk tests, Human vibrations.

1 INTRODUCTION

Vibration problems due to human movements such as walking in civil structures have become more prevalent in recent years due to several factors such as: reduction in the structural mass resulting from the use of higher strength construction materials, reduction in damping due to less office furniture and fewer partitions, reduction in stiffness due to the use of optimized computer-aided design approaches (which generally result in less stiff supporting members), and reduction in the natural frequency of floor system due to the requirements for fewer support members such as columns or walls.

Tuned mass dampers (TMDs) or tuned vibration absorbers (TVAs) have been used to control excessive vibrations of various structural and mechanical systems. A TMD is a vibrating system made of a mass, spring, and damper which is usually installed within a structural system. It is tuned to the natural frequency of the structure such that it counteracts the movements of the structure. Frahm (1911) proposed the first application of TMDs to an engineering problem. Ormondroyd and Den Hartog (1928), Brock (1946), and Den Hartog (1947) contributed significantly to the theory of vibration reduction of undamped systems using TMDs.

Setareh *et al.* (2006, 2007) used pendulum and semi-active TMDs for floor vibration control. In both cases a TMD consisting of a frame hinged at one end and free at the other end supporting steel plates was used. The semi-active damper consisted of a magnetorheological damping device. Varela and Battista (2011) conducted laboratory tests on TMDs made of steel plates directly hung from the supporting beams of a test floor and showed that they were effective in reducing floor vibrations.

The common feature of the devices discussed above is their relatively large size, which require them to be installed within the floor structure. Also, the assembly and tuning of the devices are not generally easy tasks and require involvement of highly technical personnel. In addition, there are usually several modes of vibration that need to be controlled in multi-bay floor systems. These result in the application of TMDs as a retrofit option to be difficult and cost-prohibitive. Therefore, this paper introduces an easy to fabricate, install, and tune multiple tuned mass damper (MTMD) to control several modes of vibration of floor systems. It presents the results of the studies using the MTMD on a laboratory test floor susceptible to excessive vibrations due to human movements.

2 THEORETICAL BACKGROUND

The theoretical aspect of the performance of TMDs or MTMDs to control vibration is well-established. However, for clarity of discussion, the theoretical development of the application of a TMD on a single-degree of freedom (SDOF) system is briefly discussed here.

Figure 1 shows a TMD represented by a mass (m_T), stiffness (k_T), and damping (c_T) connected to a SDOF structure represented by a mass (M_s), stiffness (K_s), and damping (C_s). X_s and x_T represent the movement of the structure and TMD, respectively. F is the frequency-dependent harmonic force acting on the structure.

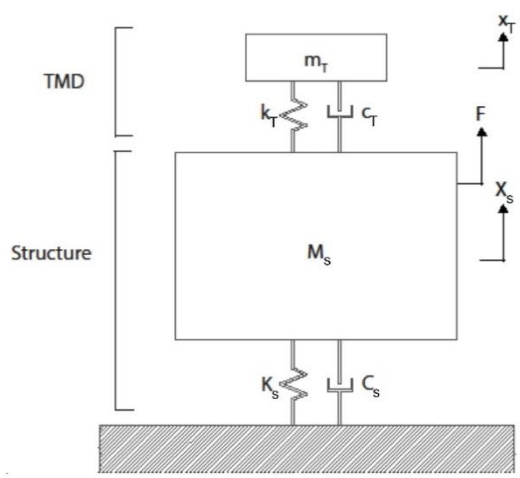

Figure 1. SDOF Model with TMD.

The acceleration response of the structure, $\ddot{X}_s$, is normalized and derived in Eq. (1) in terms of non-dimensional parameters of frequency ratio, *f* (natural frequency of the TMD to the natural frequency of the structure); mass ratio, μ (mass of TMD to the mass of the structure); force frequency ratio, *g* (excitation frequency to the natural frequency of the structure); ξ_s (damping ratio of the structure); and ξ_T (damping ratio of the TMD).

The relationship between the physical properties of the dynamic system and the above non-dimensional parameters are given below in Eq. (1):

ω_s= Natural frequency of structure (M_s)

ω_T= Natural frequency of TMD (m_T)
$f = \omega_T/\omega_s$ = Frequency ratio
$\mu = m_T/M_s$= Mass ratio
$g = \omega/\omega_s$= Force frequency ratio
$K_s=M_s\omega_s^2$= Stiffness of the structure
$k_T=m_T\omega_T^2$= Stiffness of the TMD
$C_s=2M_s\omega_s\xi_s$ = Damping of the structure
$c_T=2m_T\omega_T\xi_T$= Damping of the TMD
$\xi_s= C_s/2M_s\omega_s$= Damping ratio of the structure
$\xi_T= c_T/2m_T\omega_T$= Damping ratio of the TMD

$$R = \left|\frac{\ddot{X}_s}{\frac{F}{M_s}}\right| = g^2\sqrt{\frac{(f^2-g^2)+(2\xi_T f g)^2}{[(1-g^2)(f^2-g^2)-g^2 f(\mu f+4\xi_s\xi_T)]^2+4g^2\,[\xi_T f(1-g^2-g^2\mu)+\xi_s(f^2-g^2)]^2}} \quad (1)$$

In the above equation, R, is the normalized acceleration response of the structure. In general, for a given μ and ξ_s the TMD non-dimensional parameters, f , and ξ_T are computed to minimize the maximum response of the structure (R_{max}).

3 DESCRIPTION OF THE MTMD

The MTMD is entirely made of steel parts which are screwed to each other. This facilitates the fabrication and assembly of the unit. It consists of five parts:

- Base: a steel plate to provide a flat surface for the stand components to be attached to.
- Stand: steel angles to provide support for the cantilevered plates.
- Wings: double-cantilevered uniform thick steel plates providing the required stiffness for the device.
- Steel Weights: steel plates providing the required mass for the MTMD.
- Dampers: these are standard dampers to provide the required damping for the device.

4 TUNING TESTS

Tests were conducted at the Virginia Tech Vibration Testing Laboratory. The facility houses a two-story full-scale test structure that was used for the studies conducted here. Figure 2 shows the MTMD on the test floor used for this study, which consists of a 30ft by 30ft concrete floor supported by steel beams and columns (at the corners).

Figure 2. MTMD on the test structure at the Vibration Testing Laboratory.

Table 1. Measured natural frequencies and damping ratio of the test floor.

Mode Number	***f_s (Hz)***	***ξ_s (%)***
1	5.225	0.51
2	9.225	0.36
3	10.65	0.42

First, the natural frequencies *(f_s)* and damping ratios *(ξ_s)* of the test floor with the MTMD inactive (locked) were measured as shown in Table 1. The MTMD wings were tuned to the measured floor natural frequencies, and subsequently placed on the test floor. The location of the MTMD placement was selected such that it can simultaneously provide vibration reduction to the first three modes of the floor. An electrodynamic shaker (APS 113) and a number of accelerometer (PCB 393C) connected to an OROS signal analyzer were used for this purpose. The total steel weights used on each wing to tune the device to the first three modes were as follows: W_1 = 54 lb W_2 =40.5 lb, and W_3 =27 lb.

The optimum MTMD frequency ratio and damping ratio based on the mass ratios and measured floor damping ratios were computed analytically. The mass ratios for the first three modes are: 0.0018, 0.00325, and 0.00167, respectively. Using the equation (1), the predicted reduction in the floor's first three modes vibration due to the application of the MTMD were 74%, 85%, and 78%, respectively. Figure 3 shows the frequency response functions (FRF) of the floor comparing the response of the floor with the MTMD inactive (locked wings) and active (unlocked wings). A burst chirp excitation was used to obtain the FRFs. The measured average reduction in the response of the first three vibration modes of the floor were 67% (Range: 58 – 73%), 50% (Range: 39– 70%), 48% (Range: 39-54%), respectively.

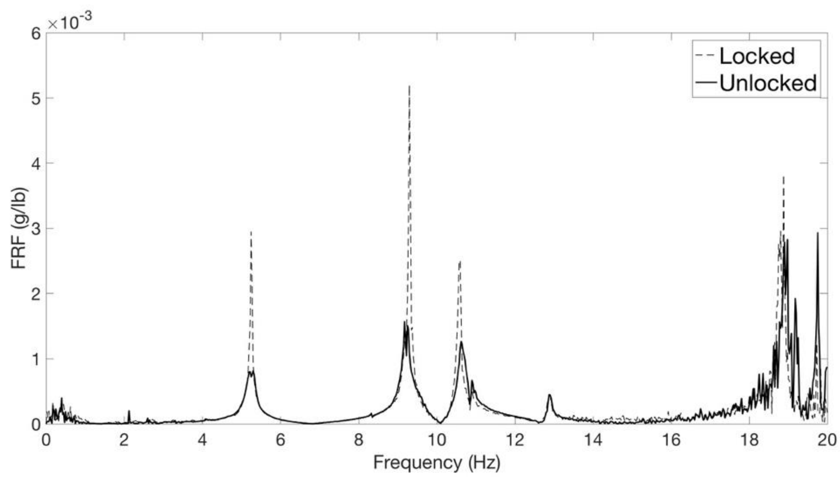

Figure 3. Comparison of the floor frequency response function with the MTMD active and inactive.

5 WALK TESTS

To evaluate the performance of the MTMD to reduce floor vibrations due to walking, 30 human subjects were asked to walk individually on the floor at the corresponding sub-harmonics of the first and second mode natural frequencies of the floor. Each person walked from one end of the floor along the center line to the opposite end and waited until the measurement was complete. The selected step frequency for walking at the first mode sub-harmonic was 158 spm (steps per minute) and for the second mode was 139 spm. Figure 4 shows a sample of the measured floor response in the time and frequency domains when the subject walked at the first mode sub-harmonic (158 spm). The average reduction in the vibration for the first mode was 59% and for the second mode was 42%. It has to be noted that typically walking excitations cannot cause resonance at the frequencies above 10 Hz since the dynamic load factor at such high frequencies is normally insignificant, which was the case for the third mode of vibration of the floor.

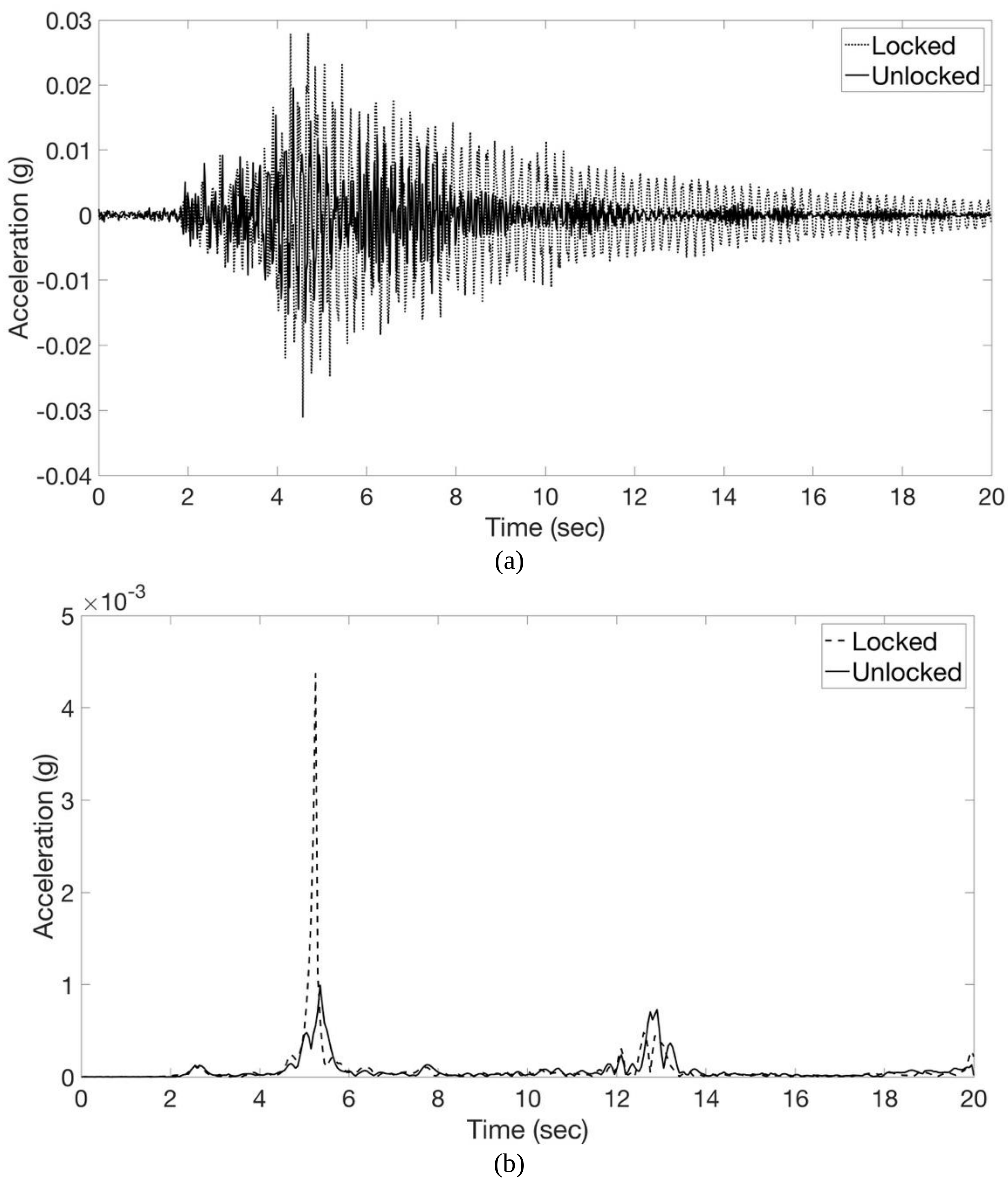

Figure 4. Comparison of the floor response when the subject walked at the first mode Subharmonic (158 spm): (a) time-history, (b) frequency spectrum.

6 CONSISTENCY TESTS

To check the consistency in the performance of the device due to the variations in the environmental conditions and the possibility of presence of non-linearities in the floor and device, the FRF curves were measured on different days while the ambient temperature was also recorded. The results showed that the performance remained consistent, which was demonstrated by the small coefficients of variation for the vibration reduction for the FRFs to be 7%, 15%, and 9% for the first three modes, respectively. The total number of measurements for each case was 40.

7 CONCLUSION

This paper provided the details of a multiple-tuned-mass-damper (MTMD) which is easy to fabricate, assemble, and tune to control vibrations of different modes of a floor system. The results of the tests conducted showed that even though the device had very small mass compared to the floor mass, it could provide good reduction in the floor vibration due to human walks at the subharmonics of the natural frequencies of a building floor system. It also showed that the device performs consistently and the system non-linearities are insignificant.

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