Chapter 5
Experimental Evaluation of an Elastomeric Tuned Mass Damper Device for Vibration Suppression on a Laboratory Floor

Sahabeddin Rifai, Russell A. Larsen, Robert Baxter, Joshua M. Mouras, Ronald Klemencic, and Onur Avci

Abstract This paper presents the experimental evaluation of a novel tuned mass damper (TMD) featuring an elastomeric sphere damping element on a full-bay 30 ft × 40 ft laboratory floor specimen. The TMD was designed by Magnusson Klemencic Associates (MKA) and tested at West Virginia University (WVU). The performance of the TMD was evaluated using frequency response functions (FRFs) from shaker excitations, free-vibration decay tests, and metronome-guided human walking tests. Analysis of the FRFs demonstrated successful tuning, with the TMD suppressing the peak resonant response by 87%. Furthermore, resonance and free decay tests showed the TMD increased structural damping over a range of 10% to 39%, with performance dependent on vibration amplitude. Under serviceability-level walking excitations, the TMD reduced the 90$^{\mathrm{th}}$ percentile Equivalent Sinusoidal Peak Acceleration (ESPA) by 7%. These results provide a successful experimental proof-of-concept, validating the TMD's effectiveness and establishing its potential as a practical solution for mitigating vibration serviceability issues in modern floor systems.

Keywords Tuned mass damper · Elastomeric damper · Vibration control · Floor vibrations · Steel floor systems

Introduction

The design of modern building floors increasingly favors open-plan layouts, longer spans, and lightweight materials. While this approach creates aesthetically pleasing and functional spaces, it often results in floor systems with low mass and stiffness, making them more susceptible to perceptible vibrations from human activities like walking or rhythmic exercises. A key factor governing this vibrational performance is the floor's inherent damping, its natural ability to dissipate energy. Modern construction methods, however, have led to systems with significantly lower damping values than their more traditional, heavier counterparts.

Resonant response in floors can occur when a harmonic of a rhythmic human activity (e.g., the pace of walking, the beat of an aerobics class, or synchronized crowd movement) excites a natural frequency of the floor, leading to serviceability issues that affect occupant comfort [1, 2]. Mitigating these vibrations often requires supplementary damping, typically introduced through dedicated devices. These devices are broadly categorized as active or passive. Active systems require an external power source to apply adaptive control forces to the structure. In contrast, passive systems generate control forces through their own motion in response to the structure's vibration, requiring no external power [3].

Among passive solutions, the tuned mass damper (TMD) is a particularly efficient and targeted device. A TMD consists of a mass, spring, and damper element tuned to a specific natural frequency of the structure. When that frequency is excited, the TMD resonates out of phase with the floor. This counter-motion allows the TMD to absorb energy from the floor and dissipate it, effectively suppressing the resonant response at its target mode [3, 4].

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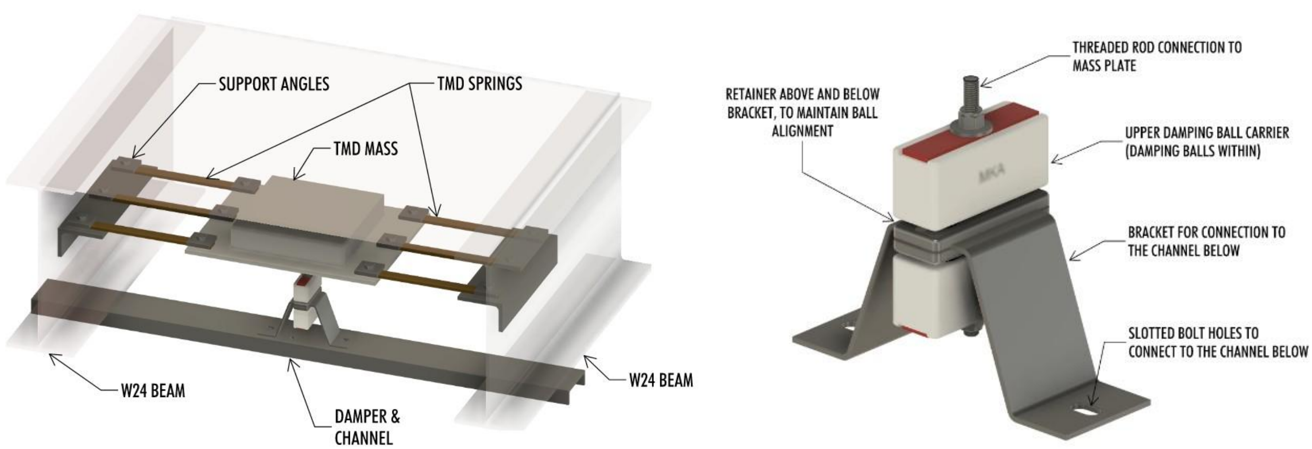
This paper presents the experimental evaluation of a novel tuned mass damper (TMD), designed by Magnusson Klemencic Associates (MKA), on a full-scale 30 ft × 40 ft laboratory floor at West Virginia University (WVU). The TMD, installed near the floor's center, comprises a steel plate mass supported by steel spring strips welded to the floor's beam webs. Its unique damping element consists of elastomeric spheres housed within a 3D-printed casing, which connects the mass to a steel channel section below. A key advantage of this design is its low-profile form factor, enabling installation within the structural depth of a typical floor system and thereby preserving architectural ceiling heights.

The objective of this study is to provide an experimental proof-of-concept for a novel TMD featuring a damper element with elastomeric spheres and to evaluate its performance in mitigating floor vibrations. The scope of this investigation involves quantifying the floor's response, both with and without the TMD, by generating frequency response functions (FRFs) from shaker excitation. Additionally, changes in system damping are determined using resonance and free-decay tests, and the TMD's effectiveness is ultimately assessed for serviceability under human-walking tests.

Elastomeric Sphere Tuned Mass Damper

The design of the MKA tuned mass damper (TMD) assembly and its damper element are illustrated in Fig. 1. Installation of the TMD begins by welding the support angles to the beam webs (Fig. 1-a) at a convenient height that leaves enough space for mass tuning plates. Second, the bottom steel channel section is aligned and connected to the bottom beam flanges by welding. Third, the bottom TMD mass plate spring strips are bolt-connected on two sides to the support angles. Finally, the damper element (Fig. 1-b) is aligned and installed between the mass plate at the top and the channel section at the bottom.



(a) (b)

Fig. 1 Schematic of the MKA TMD design. (a) View of the installed TMD with names of components. (b) A view of the MKA elastomeric sphere damper element

The TMD was installed and tested on a 30 ft × 40 ft all-steel laboratory floor structure (FastFloor) as shown in Fig. 2. The TMD was installed at the center of module 3 of the floor specimen. The plan layout of this specimen and the TMD location are shown in Fig. 3. FastFloor is a novel floor system for commercial building construction that utilizes prefabricated modules made from steel beams and plates. The modules are connected to girders on-site using field bolted connections. The tested full-bay floor specimen comprises 6 modules; 3 of which are prefabricated, and the remaining are formed by simply connecting infill plates to the prefabricated 3 modules. The steel beams are W24×68 and all the plates are $\frac{1}{2}$ in. thick. The two girders supporting the beams are W30×108. A raised access floor (RAF) was also installed on top of the steel plate.

The TMD is shown after installation in Fig. 4. During the tuning and testing process, two accelerometers were attached: one to the mass plate and one to the channel section running between the W24 bottom flanges. An additional nine accelerometers were placed on top of the floor along the centerline of Module 3 (Fig. 3). Excitations for the experimental modal analysis were provided by an electrodynamic mass shaker. The baseline FastFloor specimen without TMD includes horizontal angle blocking members connecting the beams' bottom flanges at L/3 and 2L/3 on Modules 1, 3, 5, and 6. All tests were conducted with the raised access floor (RAF) installed.

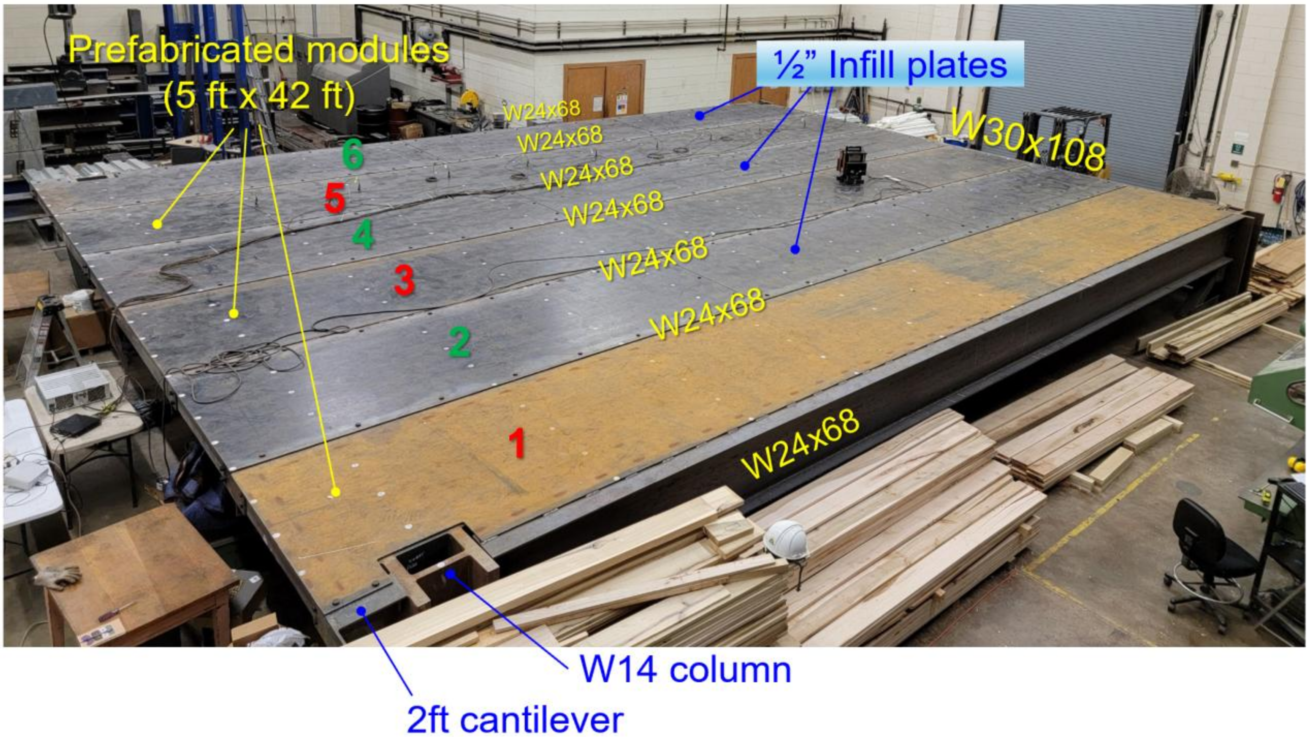


Fig. 2 View of the 30 ft × 40 ft All-Steel Floor Specimen with Components Annotated

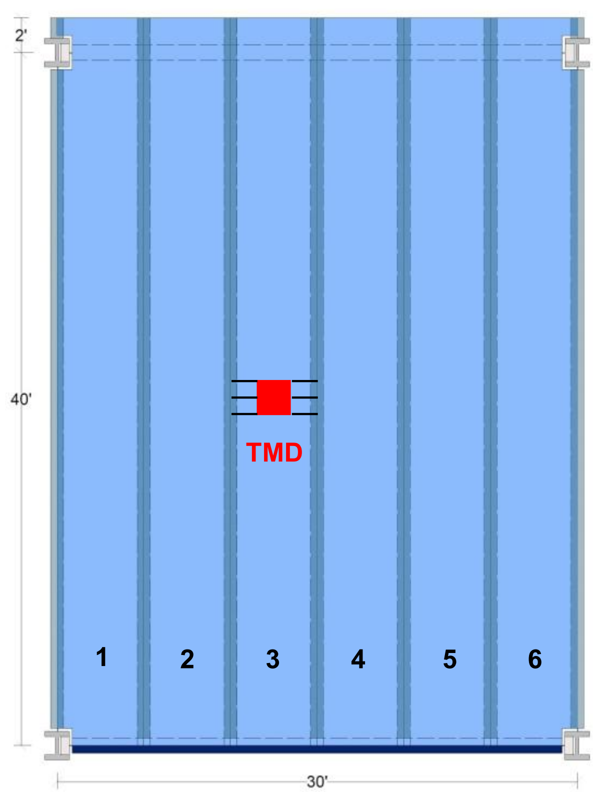


Fig. 3 Plan Layout of the FastFloor Specimen with the TMD Installed at the Center of Module 3

(a) (b)

Fig. 4 The TMD assembly after installation on Module 3: (a) view of the assembly with four spring strips; (b) the damper element, showing the elastomeric spheres separated by gaps

Tuning the Damper System

The first step was to tune the mass damper system such that its natural frequency closely matched the floor's fundamental frequency. This fundamental mode is targeted because it has the highest mass participation and is therefore the most significant contributor to the overall dynamic response. The shape of this mode, shown in Fig. 5, is characterized by the floor beams and plates bending together in a single curvature. Before the TMD was installed, the floor's baseline frequency was measured at 7.46 Hz. The tuning process was accomplished by adjusting the steel mass plates and modifying the internal gaps within the elastomeric damper element. Specifically, two different gap sizes inside the damper element were investigated to observe their effect on the system's performance.

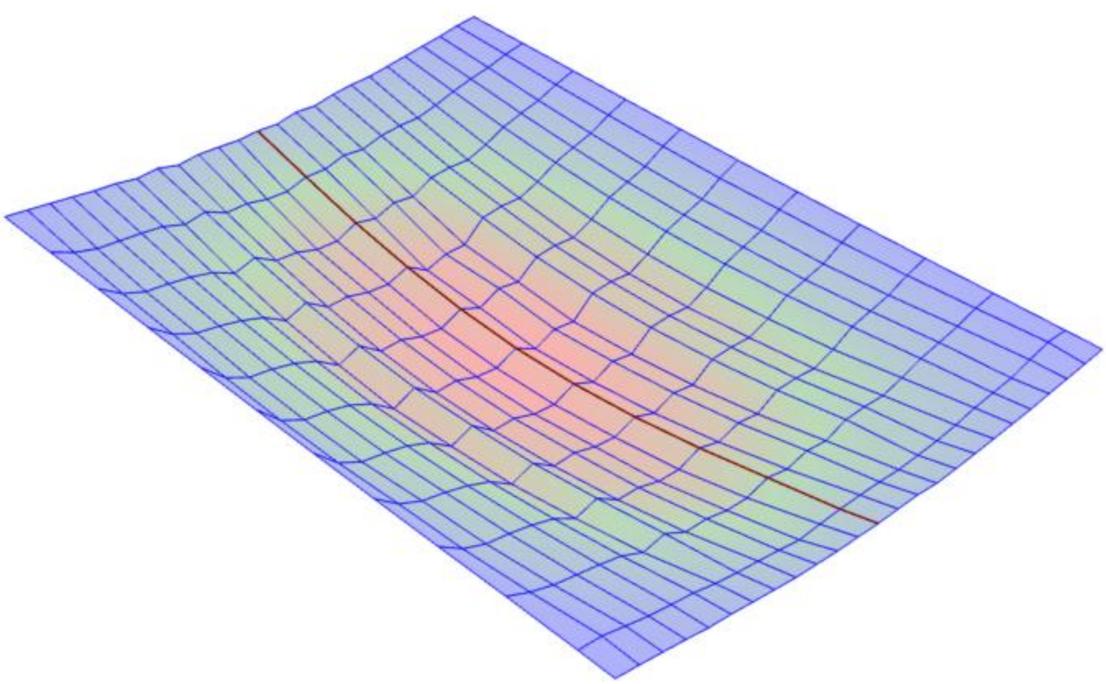


Fig. 5 Experimental Shape of the Fundamental (First) Vibration Mode of The Full-Bay Specimen

Initial Design Calculations and Experimental Optimization

Initial tuning calculations were provided by the TMD designers to guide the tuning process. Based on these calculations, a configuration of four spring strips and five standard mass tuning plates was selected to target a TMD frequency of approximately 7.4 Hz. It was anticipated that the fundamental frequency of the floor specimen would decrease slightly after the TMD was installed due to its self-weight.

To determine the optimal number of mass plates experimentally, five distinct mass configurations (0, 2, 4, 5, and 6 plates) were investigated while holding each of the two internal damper gaps at 3/16 in. The effect of each configuration was evaluated by analyzing the frequency response functions (FRFs) obtained from low-amplitude shaker excitations. The resulting FRFs, including the baseline configuration without a TMD, are presented on a logarithmic scale in Fig. 6. One observation was that the FRFs for the TMD configurations appeared rougher than the smooth baseline FRF, suggesting that the TMD introduced some nonlinear behavior into the system. The primary goal of this investigation was to identify the number of plates that produced the lowest FRF peak amplitude near the 7.4 Hz target frequency.

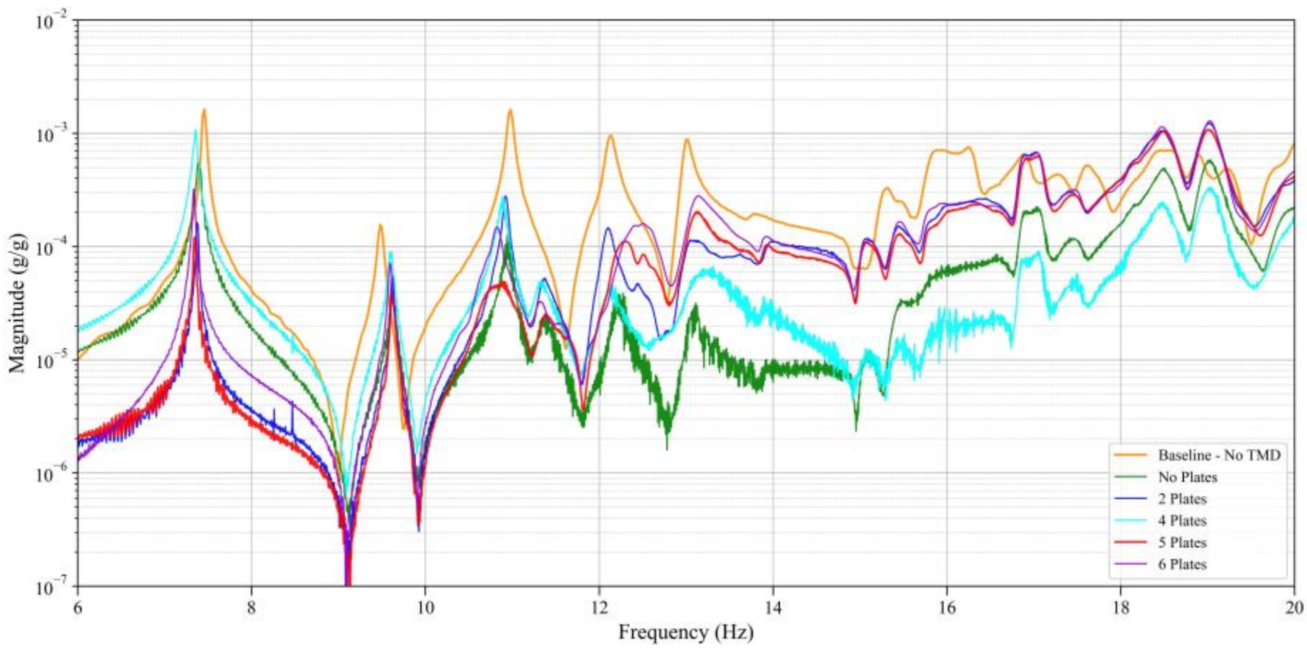


Fig. 6 Frequency response functions (FRFs) measured at the center of module 3 for different TMD mass configurations, generated using shaker excitation (logarithmic scale). The gaps in the damper are 3/16 in. each

A closer look at the 7.2–7.6 Hz frequency range, shown in Fig. 7, confirms that the five-plate configuration yielded the greatest reduction in FRF magnitude. This setup successfully reduced the peak dynamic response by approximately 87% compared to the baseline floor, a result that aligned well with the initial tuning calculations. However, an interesting phenomenon was observed: the peak FRF magnitudes for the other mass configurations did not decrease in a simple, linear trend. For instance, the configurations with four and six plates did not necessarily yield the second and third-lowest peak magnitudes. This can be attributed to the known challenges of testing large, flexible floor structures. As noted by [5], floors can exhibit nonlinear behavior, and dynamic interaction between the shaker and the structure can cause force drop-offs at resonance. These effects can lead to slight variations in measured FRF peaks, which highlights why this empirical tuning approach was valuable for identifying the configuration that most effectively reduced the peak response.

Further tuning of the TMD was conducted by adjusting the internal gap spacing within the damper element (the location of the gaps is shown in Fig. 4). Two gap sizes, 1/8 in. and 3/16 in., were investigated. It is important to note that the damper balls were always in compression with these gap widths. The amount of elastic compression varied according to the gap width. The frequency response functions (FRFs) for each configuration are compared to the baseline in Fig. 8. Based on this comparison, the 3/16 in. gap was found to provide a greater reduction in the FRF magnitude at the fundamental frequency. Gaps larger than 3/16 in. were not considered because the internal components of the damper began to show signs of looseness. Therefore, the 3/16 in. gap was identified as the optimal setting for the damper element.

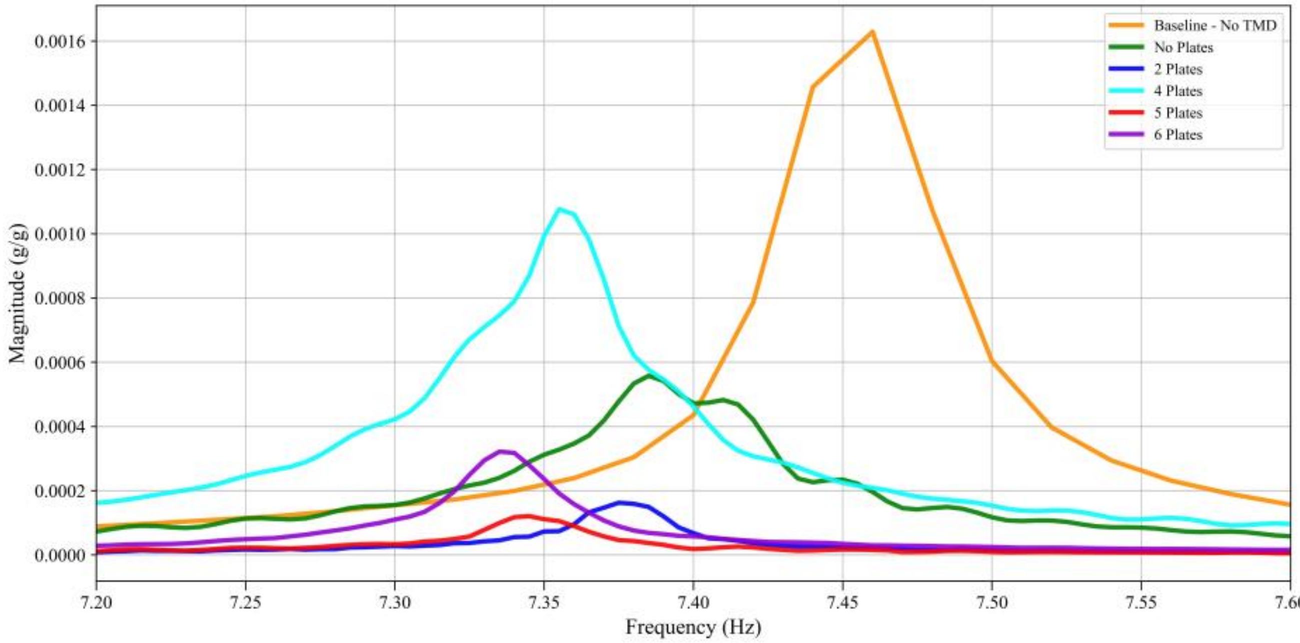


Fig. 7 Linear-scale plot of the FRFs near the fundamental mode, comparing the effect of different TMD mass configurations

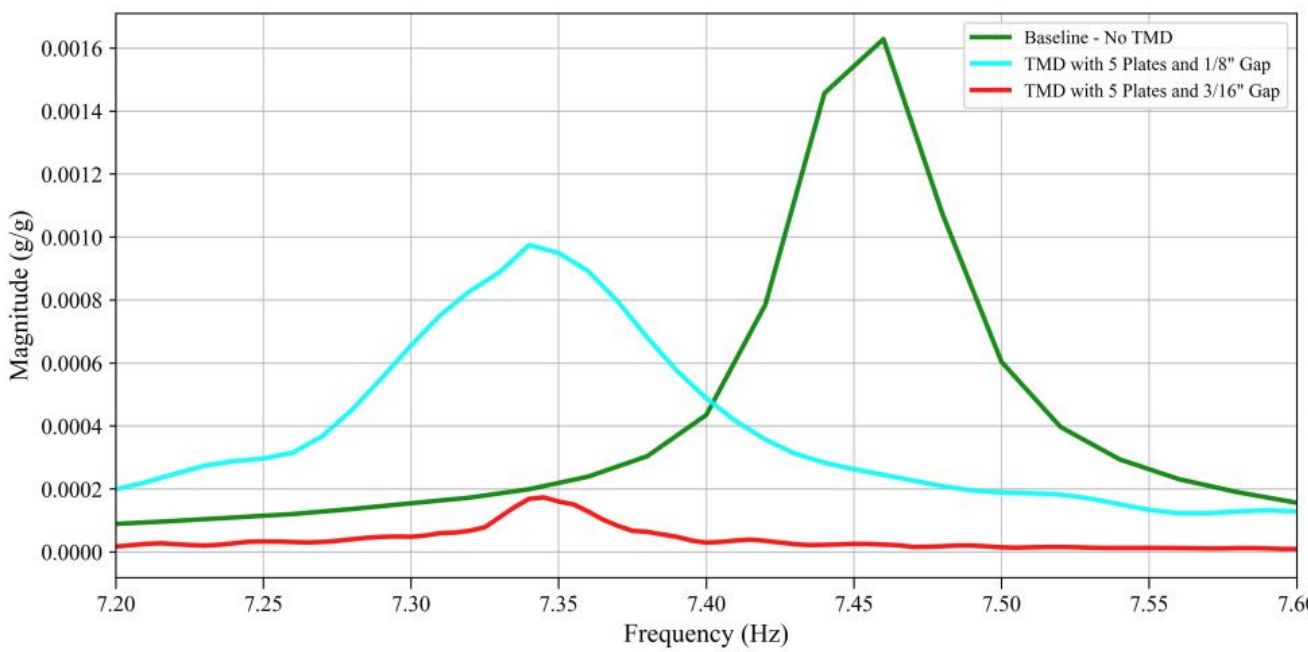


Fig. 8 Effect of the internal damper gap size on the FRFs for the TMD (linear scale)

Effect of the TMD on Structural Damping

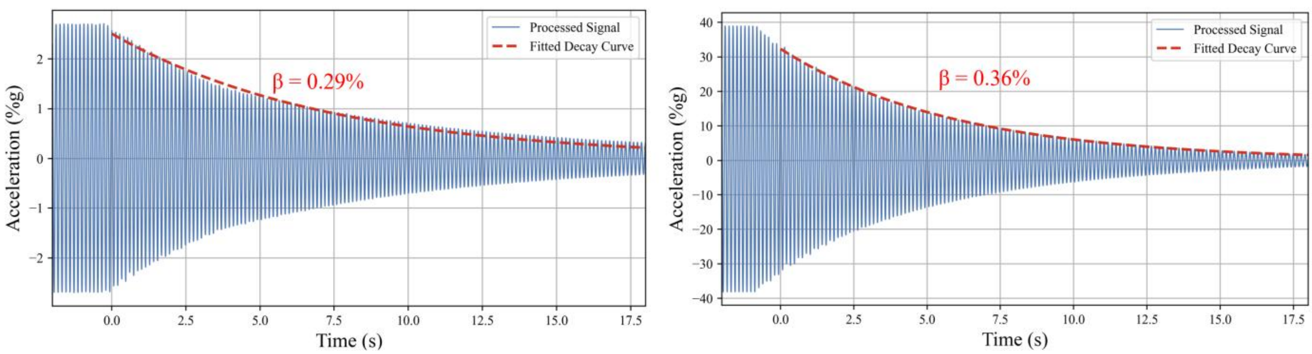
The overall structural damping of the full-bay specimen was computed using the free vibration decay method. To do this, the floor was excited to resonance at its fundamental frequency with sine waves. The shaker was then abruptly stopped, and the resulting decay of the vibration was recorded. This test was repeated at various load amplitudes to assess any potential amplitude dependence. The structural damping ratio (β) was then calculated by curve-fitting the peaks of the decay curve to the exponential decay function shown in Equation 1. To improve the accuracy of this fit, the acceleration signal was first detrended and then filtered with a Butterworth filter to remove high-frequency noise.

$$a(t) = A \cdot e^{-\beta \omega_n t} \tag{1}$$

Baseline Damping

For the baseline case (without the TMD), the damping ratio (β) was observed to increase from 0.29% to 0.36% as the excitation level was raised from low to high (Fig. 9). This amplitude-dependent trend is further detailed in Fig. 10, which shows that the damping increased rapidly at acceleration amplitudes up to approximately 4% g, after which the rate of increase in damping lessened. It is likely that system nonlinearities contributed to this higher damping at greater excitation levels.

In addition to the changes in damping, a slight decrease in the system's natural frequency (f_n) was also observed at higher excitation amplitudes. This behavior is a known form of nonlinearity, defined in the literature as the "amplitude dependence of modal parameters" [5].



(a) (b)

Fig. 9 Comparison of decay curves for the baseline specimen at (a) low and (b) high excitation amplitudes, with all signals measured at the center of module 3

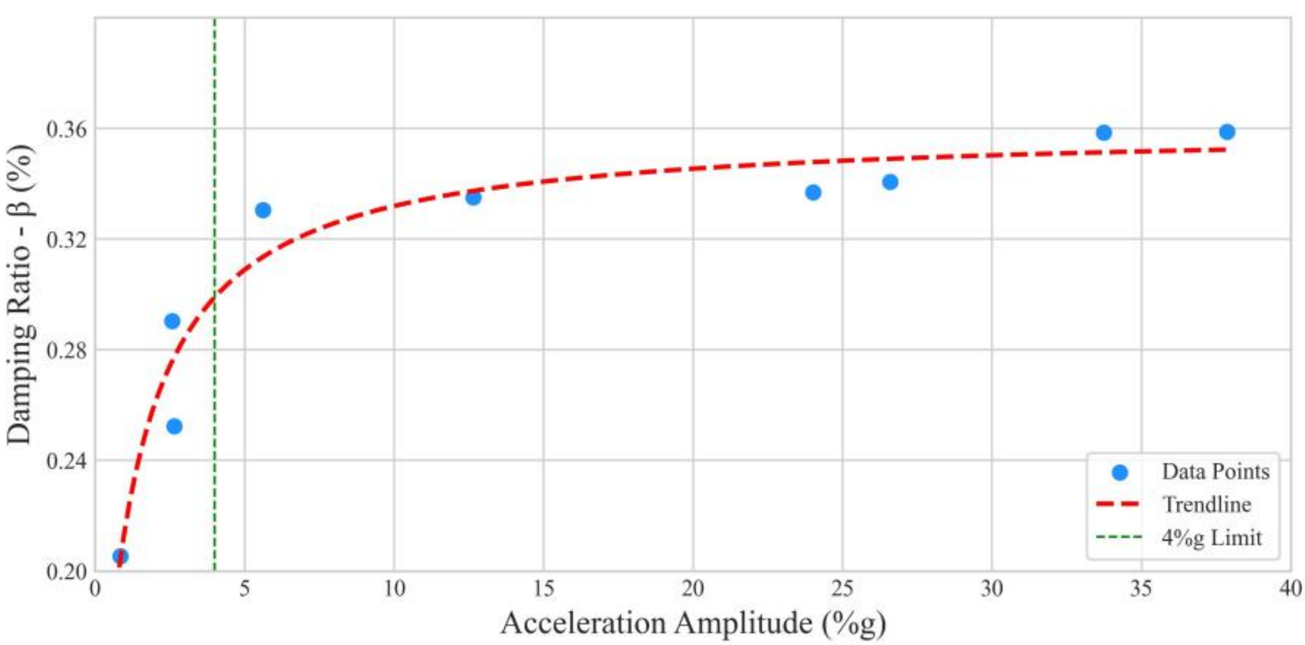


Fig. 10 Structural damping ratio as a function of excitation amplitude for the baseline case

Damping with TMD Installed

When compared to the baseline, a significant increase in the overall system damping was observed with the TMD installed. The damping ratio (β) was measured to be 0.32% at low excitation and 0.50% at high excitation (Fig. 11). These values represent an increase in damping over the baseline case of 10% and 39% for low and high amplitude excitations, respectively. Similar to the baseline case, an amplitude-dependent trend was observed, as shown in Fig. 12. The damping was found to increase rapidly with acceleration amplitudes up to approximately 4% g, after which the rate of increase slows.

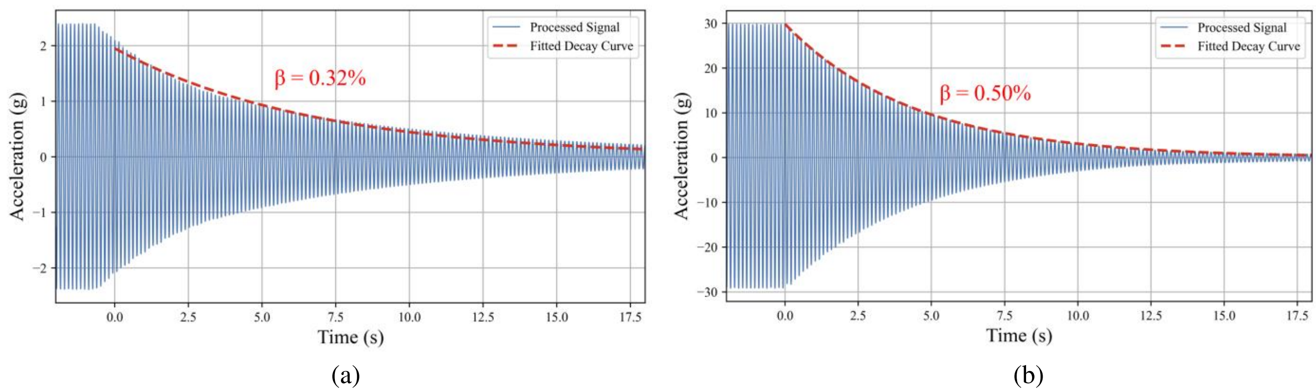


Fig. 11 Comparison of free-vibration decay curves for the TMD-equipped floor at (a) low and (b) high excitation amplitudes, with all signals measured at the center of module 3

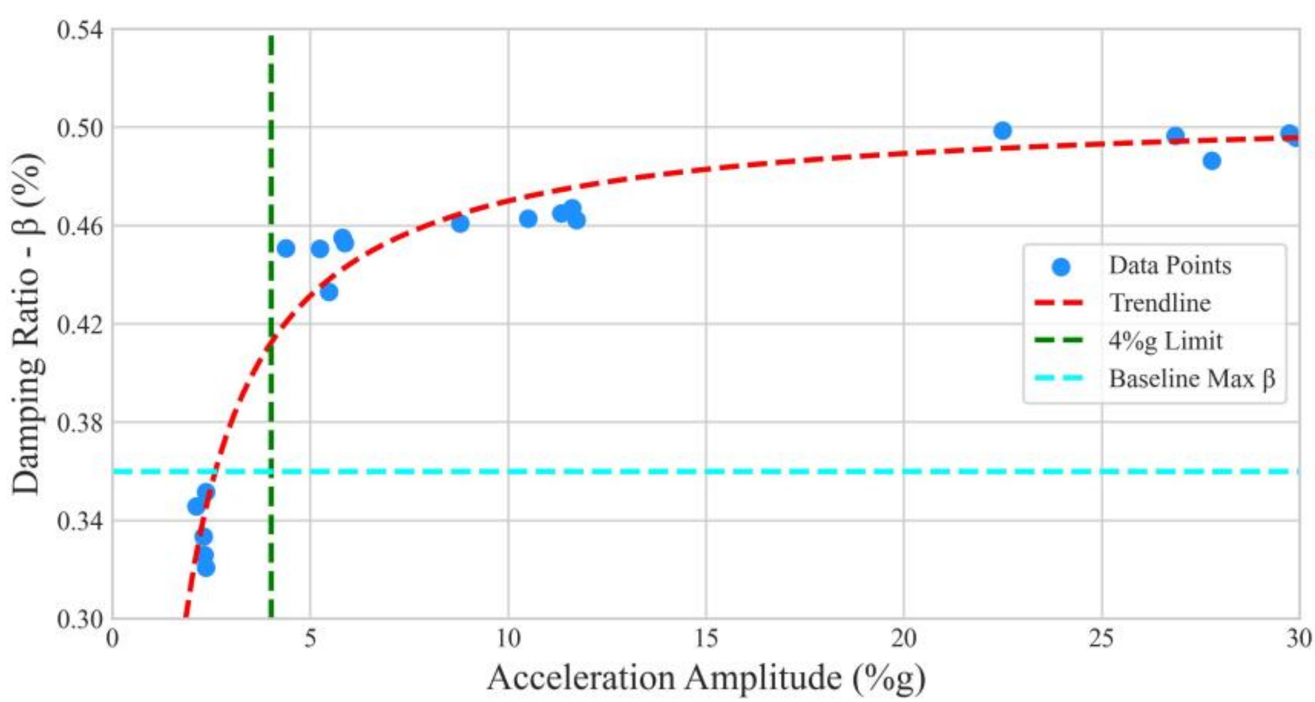
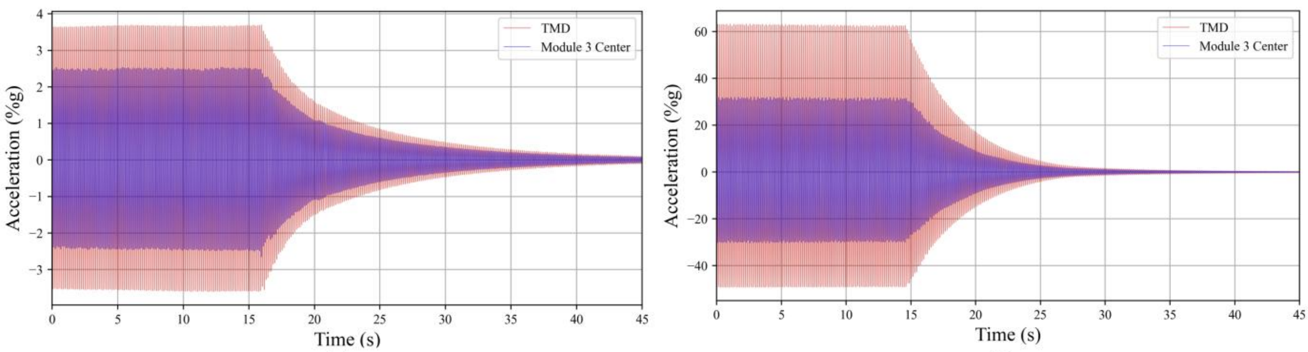


Fig. 12 Structural damping ratio as a function of excitation amplitude for the TMD-equipped floor

TMD Performance Under Resonance

The effectiveness of the TMD was evaluated by comparing its acceleration response to that of the floor's center (Module 3) during resonant sine excitation. A key indicator of effectiveness is the ratio of these two amplitudes; a higher ratio signifies that the TMD is actively moving and dissipating energy. A higher acceleration response was consistently observed at the TMD mass compared to the floor, confirming that the device was working as intended. As shown in Fig. 13, the ratio of the TMD-to-floor acceleration amplitude was found to increase from 1.5 under low excitation to 2.0 under high excitation.



(a) (b)

Fig. 13 Comparison of acceleration responses and decay curves at the TMD mass and the Module 3 center for (a) low-amplitude and (b) high-amplitude excitations

Effect of the TMD on the Floor's Response to Human Walking

We also conducted walking tests to investigate the effectiveness of the TMD design in reducing floor vibrations caused by human footfalls. To achieve resonant buildup in the floor with a normal walking pace, two individuals walked at the fourth subharmonic of the floor's fundamental frequency (f_n /4) on both baseline and TMD-equipped configurations.

Walking tests were conducted along the centerline of Module 4 at a metronome-guided pace (Fig. 14). The acceleration response was measured at the center of Module 3, and from these signals, the Equivalent Sinusoidal Peak Acceleration (ESPA) was calculated. This metric translates the complex vibration from footsteps into a standardized value that can be compared against human comfort limits.

Following the procedure in AISC Design Guide 11 and SCI P354, the ESPA was found by multiplying the maximum one-second root-mean-square (RMS) of the filtered acceleration signal by $\sqrt{2}$ [1, 6]. Since multiple trials were conducted for each configuration, the 90^{th} percentile ESPA value was used as a robust statistical measure for the final comparison.

The ESPA results from the walking tests are summarized in Table 1. To ensure a direct comparison, the baseline data presented in the table were limited to the results from the two subjects who performed tests on both configurations. With the TMD installed, a 7% decrease in the 90^{th} percentile ESPA and an 8% decrease in the maximum ESPA were observed, while the mean ESPA value remained largely unchanged. This modest level of improvement was anticipated for two main reasons. First, the previous resonance tests demonstrated that the TMD is most effective at higher vibration amplitudes, whereas normal walking is a relatively low-amplitude activity. Second, the harmonic of the walking pace that excites the floor is a higher, less impactful subharmonic of the fundamental frequency, which results in a lower input force compared to direct resonant excitation.

Table 1 Summary of ESPA results and statistics from the walking tests

			Equivalent Sinusoidal Peak Accelerations (ESPAs) [%g]				
Configuration	**# of Walking Tests**	**Step Frequency (steps/min)**	**Mean (μ)**	**Min**	**Max**	**STD (σ)**	**90^{th} Percentile (μ+1.28σ)**
Baseline	14	112	1.53	1.17	1.92	0.26	1.86
TMD	6	110	1.51	1.30	1.76	0.17	1.73

Fig. 14 Setup for the metronome-guided walking tests along the centerline of module 4, with accelerometers deployed along the centerline of module 3

Conclusions

This paper presented the experimental evaluation of a novel tuned mass damper (TMD) featuring an elastomeric sphere damping element. The study's primary objective was to provide a proof-of-concept for this design and quantify its effectiveness in mitigating vibrations on a full-scale laboratory floor specimen. The main conclusions of the study are as follows:

1. The TMD was successfully tuned to the floor's fundamental frequency by empirically optimizing its mass and gaps inside its damper element. Analysis of frequency response functions (FRFs) showed that a five-plate mass configuration with 3/16 in. damper gaps provided the highest reduction in resonant response, decreasing the peak FRF magnitude by approximately 87% compared to the baseline floor.
2. Resonance and free decay tests demonstrated that the TMD increased the floor's effective damping. The damping ratio increased by approximately 10% at low vibration amplitudes and by nearly 39% at high amplitudes.
3. Under human walking tests, the TMD achieved a modest but notable reduction in the floor's response, decreasing the 90^{th} percentile and maximum Equivalent Sinusoidal Peak Acceleration (ESPA) values by 7% and 8%, respectively. This result was consistent with the TMD's observed behavior, as its effectiveness is greatest at higher amplitudes than those typically generated by normal walking.

In summary, this study validates the performance of the novel MKA elastomeric sphere TMD. The experimental results confirm that the device effectively suppresses resonant vibrations and adds damping to the structural system, particularly at the higher vibration levels associated with serviceability issues.

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