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Analysis of the effect of passive vibration control on planar truss structures subjected to resonant loading

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Abstract. Advancements in Civil Engineering, combined with the integration of high-performance materials, have enabled the development of slender structural systems, such as planar trusses. Despite their efficiency, these structures are notably sensitive to dynamic effects, particularly under resonant excitation, which can compromise their stability and service life. In this context, a computational framework was established to analyze the performance of passive vibration-control devices in planar trusses subjected to dynamic loading, as well as to simulate multiple scenarios by varying key device parameters to achieve cost-effective implementation. The results obtained demonstrated the feasibility of employing Tuned Mass Dampers (TMDs) even in the absence of inherent damping, thereby reducing the overall cost of vibration control. These findings were validated through comparison with benchmark studies on vibration control in slender structures.

Keywords: Dynamic analysis; Structural damping; Planar trusses; Computational modeling; Civil engineering.

1 Introduction

The continuous advancement of materials and technologies in Civil Engineering has enabled the conception and realization of structures characterized by increasingly slender geometries. Within this context, planar trusses composed of slender steel members have become particularly prominent, being extensively employed in small- and medium-scale constructions subjected to diverse loading conditions. Nevertheless, such slenderness accentuates their vulnerability to long-term deterioration, a complex and multifactorial phenomenon influenced by mechanical and environmental factors.

As noted by Mazzilli et al. (2016), vibrations induced by dynamic actions significantly intensify structural fatigue owing to low inherent damping and, in certain cases, resonance with the structure's fundamental frequency. Historically, these effects were often neglected or merely incorporated through empirical impact factors, an approach now deemed analytically inadequate. In low-damping systems, excitations proximal to the natural frequency may generate considerable displacements, leading to user discomfort, material degradation, or even imminent collapse (Menezes, 2020).

In recent years, numerous investigations have introduced methodologies aimed at mitigating excessive vibrations in planar trusses, particularly in scenarios where conventional reinforcement techniques are impracticable. Among these strategies, the implementation of dynamic control devices—most notably Tuned Mass Dampers (TMDs)—has proven highly effective, as these systems are designed in accordance with the structure's natural frequency. TMDs have demonstrated notable efficiency in vibration attenuation while maintaining relatively low implementation costs (Mondin, 2025; Rossato, 2020).

This study investigates the dynamic response of a planar truss equipped with Tuned Mass Dampers (TMDs) under resonant loading. Employing a direct modeling strategy, wherein mass, stiffness, and damping are coupled at a strategically defined node, displacements with and without control devices are comparatively assessed. Variations in TMD parameters (stiffness, mass, and damping) were likewise examined to qualitatively evaluate distinct implementation scenarios.

2 Dynamic Problem

The dynamic response of a structure depends on its physical characteristics and on the properties of the applied excitation (see Mazzilli et al., 2016; Müller, 2024). Consider the damped equation of motion for structures with one or multiple degrees of freedom:

$$\mathbf{M\ddot{u} + C\dot{u} + Ku = r}, \tag{1}$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are the mass, damping, and stiffness matrices of the structure, respectively. Furthermore,the terms $\ddot{\mathbf{u}}$, $\dot{\mathbf{u}}$ and $\mathbf{u}$ are the acceleration, velocity, and displacement vectors of the structure, respectively, and $\mathbf{r}$ is the external load vector.

The dynamic response of the system of differential equations represented by Eq. (1) can be determined using different approaches. In this work, the direct application of Newmark's numerical time integration method was chosen, whose generalized formulation is given by:

$$\hat{\mathbf{K}}\mathbf{u}_{i+1} = \hat{\mathbf{r}}_{i+1}, \tag{2}$$

with

$$\hat{\mathbf{K}} = \mathbf{K} + \frac{1}{\alpha \Delta t^2}\mathbf{M} + \frac{\delta}{\alpha \Delta t^2}\mathbf{C}, \tag{3}$$

$$\hat{\mathbf{r}}_{i+1} = \mathbf{r}_{i+1} + \mathbf{M}\left[\frac{1}{\alpha \Delta t}\mathbf{u}_i + \frac{1}{\alpha \Delta t}\dot{\mathbf{u}}_i + \left(\frac{1}{2\alpha} - 1\right)\ddot{\mathbf{u}}_i\right] + \mathbf{C}\left[\frac{\delta}{\alpha \Delta t}\mathbf{u}_i + \left(\frac{\delta}{\alpha} - 1\right)\dot{\mathbf{u}}_i + \left(\frac{\delta}{\alpha} - 2\right)\frac{\Delta t}{2}\ddot{\mathbf{u}}_i\right] \tag{4}$$

the values for the index i are obtained at the instant $t = i\Delta t$, where Δt represents the constant time step adopted in the integration. The constants δ and α must satisfy the following conditions to ensure the unconditional stability of the Newmark method:

$$\delta \geq \tfrac{1}{2} \text{ and } \alpha \geq \tfrac{1}{4}\left(\tfrac{1}{2} + \delta\right)^2, \tag{5}$$

The acceleration and velocity vectors can be determined by the equations:

$$\ddot{\mathbf{u}}_{i+1} = \frac{1}{\alpha \Delta t^2}(\mathbf{u}_{i+1} - \mathbf{u}_i) - \frac{1}{\alpha \Delta t}\dot{\mathbf{u}}_i - \left(\frac{1}{2\alpha} - 1\right)\ddot{\mathbf{u}}_i \tag{6}$$

$$\dot{\mathbf{u}}_{i+1} = \dot{\mathbf{u}}_i + (1 - \delta)\Delta t\ddot{\mathbf{u}}_i + \delta \Delta t\ddot{\mathbf{u}}_{i+1} \tag{7}$$

2.1 Vibration Control

According to Soto and Adeli (2013), one way to mitigate the effects of harmonic loads on structures is to modify their original physical properties, such as increasing the moment of inertia (through section height) or directly adjusting stiffness to alter the natural frequency. These modifications aim to ensure performance under both ultimate and serviceability limit states. However, this approach is often unfeasible due to architectural, constructional, or aesthetic constraints. Therefore, the use of Tuned Mass Dampers (TMDs) represents a viable alternative.

These devices essentially consist of a mass, a spring, and, in some cases, a damper (which can be omitted to reduce costs). The TMD is tuned so that its natural frequency closely matches the natural frequency of the structure or the vibration mode to be controlled. Consequently, when subjected to excitation, the TMD generates inertial forces that act in opposition to the inertial forces induced by the external excitation on the structure, thereby contributing to the reduction of the vibratory response (Mazzilli et al., 2016; Menezes, 2020).

3 Methodology and Computational Implementation

This work developed a computational algorithm to evaluate the influence of TMDs on planar trusses using the Finite Element Method (FEM). The structural model and incident harmonic loading were defined, and an elastic dynamic analysis was performed to obtain the structure's natural vibration mode. Computational routines

were then implemented for the truss element, TMD, and discrete damper mass. Finally, variations in damper parameters were studied to assess their effects on the structure's dynamic response.

3.1 Structural Model

The chosen model consists of a planar truss tower with a height of 12 meters, simply supported on a rigid foundation (Fig. 1(a)). The TMD attached to the tower is composed of three distinct elements: a central discrete mass m, stiffness springs k and dampers c as shown in Fig. 1(b).

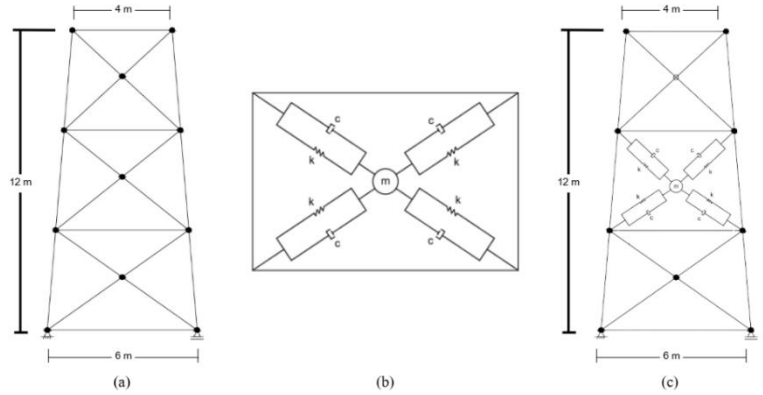


Figure 1. (a) Analyzed structure; (b) TMD model; (c) TMD coupled to the second floor of the structure.

The tower consists of 2D truss bars with one degree of freedom per node along the x and y axes. The pinned connections ensure that elements experience only axial forces. In the implemented TMD system (Fig. 1(b)), the central mass is connected to four elements, defined as the same type as the truss bars.
The local stiffness $\mathbf{K}_{\boldsymbol{l}}$ and mass $\mathbf{M}_{\boldsymbol{l}}$ matrices for the adopted truss model are:

$$\mathbf{K_l} = \frac{EA}{l}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad ; \quad \mathbf{M_l} = \frac{\rho AL}{6}\begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix} \tag{8}$$

For the respective calculations, values were defined for the modulus of elasticity E, cross-sectional area of the bar A, bar length l and specific weight of the material ρ. The transformation of the local matrices to the global system was performed using the classical 2D transformation matrix.

The structural contribution to the damping matrix is neglected to focus on the TMD's influence under resonant excitation. Omitting structural damping provides a clearer view of the device's effect on the dynamic response.

To implement the model in the calculation routine, the springs and dampers connecting two nodes were treated similarly to the bars, such that the stiffness matrix of the TMD element is identical to that presented in Eq. (8). The damping matrix $\mathbf{C}$ and additional mass matrices are shown in Eq. (9):

$$\mathbf{C} = c_a\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad ; \quad M_m = m\begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix} \tag{9}$$

where C_a is the damping value of the TMD elements coupled to the lumped mass, and m represents the value of the lumped mass associated with each degree of freedom. The form of the periodic external loading used in the calculations is:

$$\mathbf{p(t)} = p_0 \cdot \sin(\bar{\omega} t) \tag{10}$$

where p_0 is the maximum amplitude of the load and $\bar{\omega}$ is the excitation angular frequency of the load. The harmonic load was applied at the upper left node of the truss, where the largest displacements occur.

4 Results and Discussion

The structure is composed of steel profiles, and its members exhibit linear elastic behavior. The geometric

and mechanical properties are presented in Table 1.

Table 1. Properties of the adopted steel section.

Parameter	Value (unit)
E	200 (GPa)
A	78.54 (cm²)
ρ	7850 (kg/m³)

Using these values, the mass and stiffness matrices of the structure without the TMD were constructed. A modal analysis of the truss then yielded the natural frequency vector, with the first component being $\omega = 170 rad/s$. This value was adopted for the periodic loading applied to the upper-left node, oriented along the x-axis:

$$\boldsymbol{p(t)} = 100\,\sin(170t) \tag{11}$$

Time integration yielded the evolution of the structure's horizontal displacements at the upper-left node, which shows a growth trend, as illustrated in Figure 2(a).

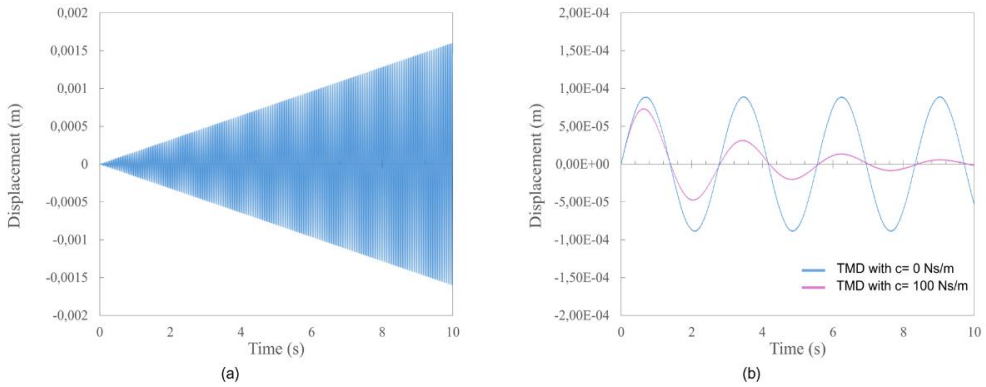


Figure 2. (a) Displacements at the top of the structure; (b) Displacements at the top of the structure with TMD.

In the structure with the TMD, to find a combination of mass, stiffness, and damping that would yield the smallest maximum displacement, tests were conducted varying these parameters to determine which had the greatest influence on the structure's dynamic response. The obtained data are shown in Figure 3.

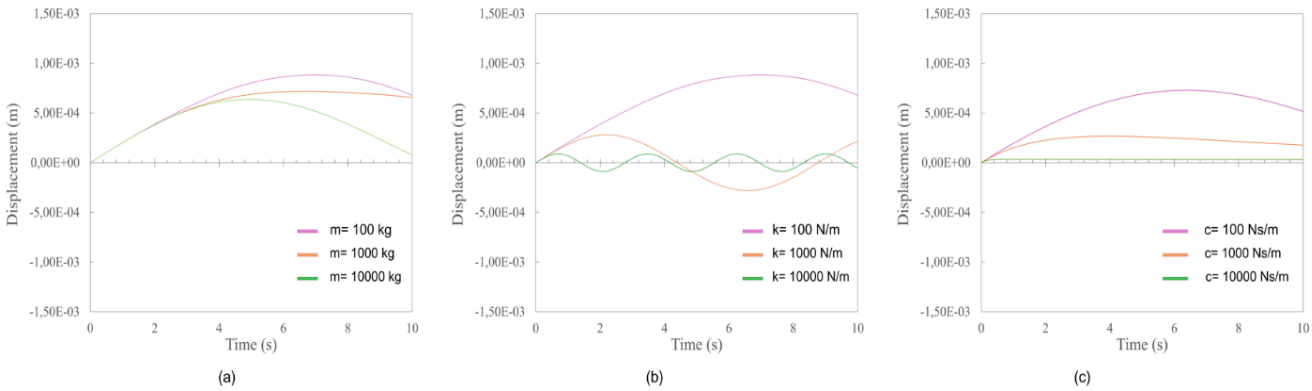


Figure 3. (a) Variation in discrete mass; (b) Variation in stiffness; (c) Variation in damping.

Variation in stiffness k (Fig. 3(b)) had a more significant impact on the response frequency and reduction of maximum amplitude, especially at higher values, which greatly decreased the structure's maximum displacements. Variation in mass m (Fig. 3(a)) only reduced amplitude for much larger masses, which may be unfeasible for certain structures. Therefore, for an undamped TMD, tuning stiffness is more effective for vibration control than increasing mass alone.

Damping c (Fig. 3(c)) significantly reduces amplitude by dissipating energy without greatly altering the response frequency, but very high values increase the TMD's cost. When considering mass, stiffness, and damping together, it is observed that higher stiffness reduces damping effectiveness because damping depends on the relative velocity between the mass and its supports, which decreases with high stiffness. Lower stiffness increases the mass movement, enhancing damping effects. Thus, stiffness tunes the system, while damping controls the intensity and speed of vibration attenuation.

With the aim of making the use of the TMD as cost-effective as possible, the values $m = 100kg$ e $k = 10kN/m$ were defined, and two analyses were performed: without and with the use of critical damping $c = 10^2Ns/m$. A preliminary investigation determined that coupling the TMD to the structure's second level (or second floor) yielded the greatest efficiency in reducing maximum displacements (Fig. 1(c)).

Analyzing the displacements in Fig. 2(b), the TMD's effect on the structural dynamic response is evident. After the transient period, the amplitude stabilizes and gradually decays, showing effective energy dissipation. Undamped displacements reach considerably higher amplitudes, while the damped response stabilizes as the device dissipates vibratory energy. Without damping, the system vibrates at a slightly off-natural frequency, which may reduce displacements but can induce long-term structural fatigue.

5 Conclusions

This study presented an analysis of the dynamic response control of planar truss structures subjected to resonant excitation, emphasizing the practical implementation of Tuned Mass Dampers (TMDs) based on their governing parameters: mass, stiffness, and damping. The results demonstrated that TMDs are effective in mitigating maximum displacements and in enhancing the equivalent structural damping. It was further observed that the undamped configuration, although not inherently dissipating energy, achieved a substantial reduction in displacements and, when combined with a reduced mass ratio, may constitute a technically viable and low-cost alternative for vibration control. The successful computational implementation confirmed that the optimal TMD configuration can be efficiently determined in accordance with the dynamic characteristics of the structure. Nevertheless, the application of undamped devices warrants careful consideration of potential fatigue effects and verification of the permissible number of vibration cycles under service conditions.

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Authorship statement. The authors declare sole responsibility for the authorship of this work and confirm that all material contained herein is either original or incorporated with the explicit permission of the respective copyright holders.References

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