# Research and application of Tuned mass dampers - type vibration absorber devices for buildings

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**Abstract.** Tuned mass dampers (TMD) is one of the most common devices for the passive control of structures subjected to wind, earthquakes, etc, the structure of these dampers consists of three main parameters: mass, damping, and spring. The article considers the use of TMD in building to reduce the acceleration and amplitude of vibrations on the upper floors. In this research, some current applications of TMD were proposed to study the response of 3-story height buildings with periodic load using time history analysis with and without the TMD. The study indicates that the response of structures such as displacements and acceleration of the upper floors can be dramatically reduced. The goal of the study is to identify an economically attractive solution that allows the fullest use of the potential of building structures in high-rise construction, abandoning the need to build massive frames leading to over-consumption of materials.

**Keywords:** Vibration absorber; TMD; Damped structure; High rise building.

## 1. Introduction

Urbanization, coupled with modern design and construction technologies, has resulted in taller and lighter structures. As an example, the world's tallest man-made structure the Burj Kalifa tower stands a remarkable 828 m from its base with an estimated weight of over 110,000 tonnes (Baker et al. 2007). One of the trade-offs of building to larger heights is the susceptibility to vibration due to the inherent flexibility of the structure. When excited by environmental dynamic loads, such as wind, this could result in large amplitude motion at the top of the structure.

There are two significant negative effects of structural vibrations on building structures (Sain et al. 2007). The first effect is the long-term fatigue of structures due to the periodic dynamic loading. It is well established that the leading cause of material failure in building structures is due to fatigue.

The second effect is the human perception of the induced motion. Humans are very perceptive to even minor vibrations. Sensitive people can perceive accelerations as low as 0.05g (Kareem et al. 2007). Between 0.1g and 0.25g, structural motions may affect an individual's ability to work, and over the long term, it may lead to motion sickness (Kareem et al. 2007).

The desire then is to mitigate structural vibrations in building structures. The control of structural vibrations can be achieved by various methods (Mendis et al. 2007). The amplitude and frequency of structural vibrations can be manipulated by modifying the structural mass, stiffness, shape, and damping. In the case of wind-induced vibrations, changing the geometry of the structure can reduce the aerodynamic forces; hence, lowering the amplitude of structural vibrations. Adding additional bracing will also stiffen the structure and reduce building sway (Mendis et al. 2007). Alternatively, the addition of passive or active stabilizing forces on the structure from an external dampening device can also be implemented to mitigate the effect of structural vibrations (Gerges and Vickery 2005). One such example is the tuned mass dampener (TMD). TMDs operate by providing additional dampening to the building structure. They are advantageous over conventional design methods-especially for taller lighter construction since they are economical and can be implemented as an add-on to existing or new structures. Real structures may employ a combination of vibration suppression methods. An example of such a structure is Taipei 101, the second tallest man-made structure in the world. The skyscraper, which is shown in Figure 1, stands 508 m above ground level in a region which experiences strong winds, ground vibrations, and typhoons (Tamboli et al. 2008). Design elements of the structure include three TMDs, one of which is a pendulum TMD and the largest TMD in the world at 660 tonnes (Tamboli et al. 2008).

A TMD, or harmonic absorber, is a passive system (although variants include active elements) that can be modeled with a mass, a spring, and a damper. The TMD is tuned to a ratio of the structure's natural frequency (or another modal frequency). When the structure is excited at the tuned frequency, the damper resonates out-of-phase with the point of connection to the structure (Setareh et al. 2006). Vibration energy is dissipated from the structure via dissipative elements (dampers) that are a part of the TMD system. As a result, it reduces the vibration of the building. Although there are several different implementations of the TMD design, the two most common types of TMDs used are translational TMDs, and pendulum TMDs (PTMD). The concept of the TMD is not by any measure novel, and its performance is well documented (e.g., Gerges 2003, Chang 2010, and Mendis 2007). It was first developed by Herman Frahm in 1909 to reduce the vibration in the hull of ships (Conner 2003). Today it is commonly used in buildings, automobiles, and virtually any system where vibration suppression is desired.

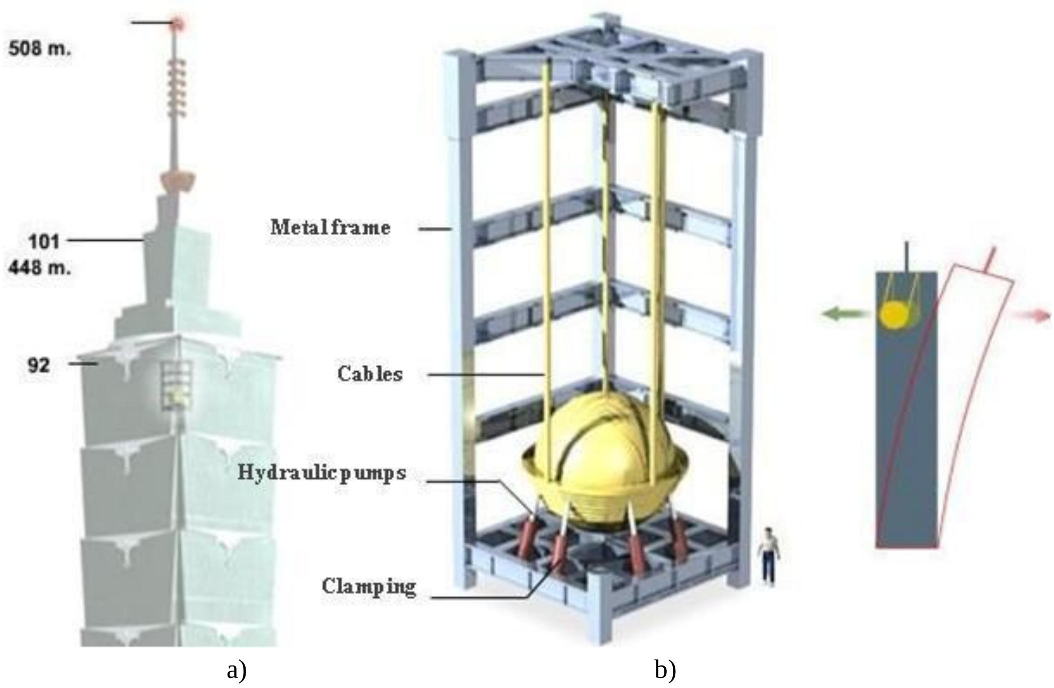


**Figure 1:** (a) Taipei 101 (Powell 2007) (b) Taipei 101 PTMD

The advantages of this system are external power is not needed, provides a large damping force, and can be installed on the existing structure. Some of its limitations are a narrow frequency, sensitivity to mistuning, and the need for a dedicated area to house the system.

The main purpose of this article is the following:

- Present a fundamental of tuned mass dampers (TMDs).
- Application of TMD on a simple frame.
- Comparison of the effectiveness of TMD with some other methods in reducing vibration of 2D frames 3- story.

# 2 Type of TMD

## 2.1 Translational TMD Systems

Translational TMD can be either unidirectional or bidirectional systems (Conner 2003). In unidirectional systems, the motion of the TMD mass is restricted to a single direction, often by placing the mass on a set of rails or roller bearings, as depicted in Figure 2. In bidirectional systems, the mass can move along both coordinate axes. In either topology, a set of springs and dampers are placed between the TMD mass and the supporting structure which is fixed to the structure.

Translational TMD systems have been implemented in large-scale structures for over 40 years (Kareem et al. 2007). Examples of structures containing translation TMD systems include the Washington National Airport Tower, the John Hancock Tower, and the Chiba Port Tower.

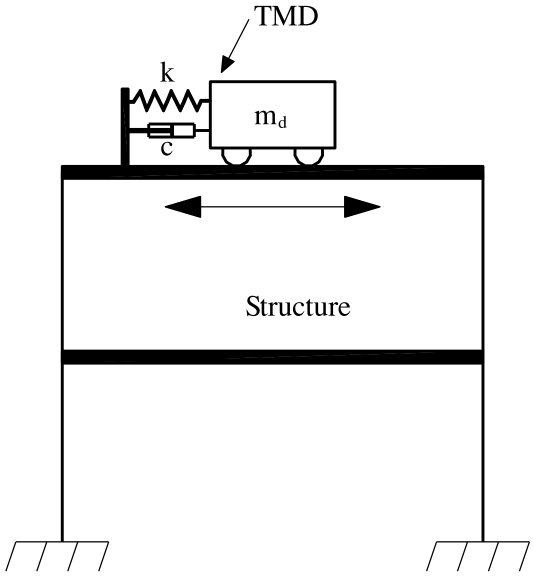


**Figure 2.** Schematic of a unidirectional translational TMD

## 2.2 PTMD Systems

PTMDs replace the translational spring and damper system with a pendulum, which consists of a mass supported by a cable that pivots about a point, as illustrated in Figure 3. They are commonly modeled as a simple pendulum. For small angular oscillations, they will behave similarly to a translational TMD and can be modeled identically with an equivalent stiffness and equivalent damping ratio. Hence, the design methodology for both the translational TMD system and PTMD systems are identical (Conner 2003). A major motivating factor for using a PTMD system over an equivalent translational TMD system is the absence of any bearings to support the TMD mass (Conner et al. 2003). The bearing support structure used in the translational TMD assembly is expensive and susceptible to wear over the lifespan of the TMD system. As a result, PTMD designs can be less expensive to 9 manufacture and last longer. Nearly 50% of structures in Japan that use TMD systems utilize PTMD systems (Kareem et al. 2007). Examples include Crystal Tower in Osaka, Higashimyama Sky Tower in Nagoya, and Taipei 101 in Taipei (Conner 2003). Studies on the use of PTMD systems generally focus on the optimization of PTMD design parameters to reduce excessive lateral deflections in structures. Gerges and Vickery (2003) utilized a non-linear wire rope spring PTMD system in an experimental case study, concluding that their performance approaches optimal linear TMD designs while providing smaller relative displacements

for lower mass ratios. Setareh et al. (2006) presented optimization algorithms for a PTMD system induced by pedestrian loading.
With this pendulum system to increase the efficiency of reducing vibrations, It often installs an additional damper for the mass as shown in Figure 3b.

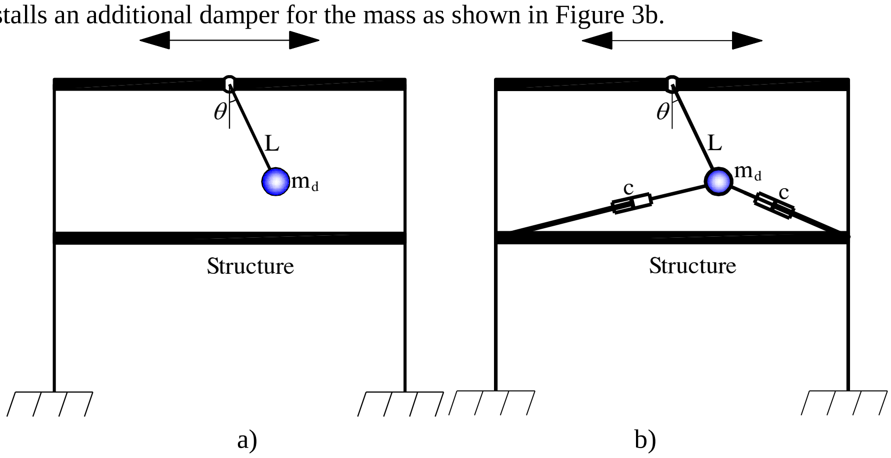


**Figure 3:** Schematic of a PTMD

## 3 Determining the parameters of TMD

The parameter of TMD will be based on Den Hartog optimization criteria assuming that the structure fits the condition for the optimization criteria.

### 3.1 Den Hartog's Optimization Criteria

TMD efficiency in reducing structural response can be gained by following the basic development of Den Hartog for the simple case where the structural system is considered undamped (c=0) and is subject to a sinusoidal excitation with frequency ω $(f(t) = P_0 \sin \omega t)$(Soong & Dargush, 1997). This procedure compares the dynamic effect of a TMD with the static deflection produced by the maximum force applied statically to the structure. The dynamic amplification factor for an undamped structural system, R, is

$$R = \frac{y_{\max}}{y_{st}} = \sqrt{\frac{(\alpha^2 - \beta^2)^2 + (2\zeta_d \alpha\beta)^2}{[(\alpha^2 - \beta^2)(1-\beta^2) - \alpha^2\beta^2\mu]^2 + (2\zeta_d \alpha\beta)^2 (1-\beta^2-\beta^2\mu)^2}} \quad (1)$$

Where $\beta = \omega_f/\omega$ External force excitation frequency ratio
$\alpha = \omega_d/\omega$ TMD frequency ratio
$\mu = m_d/m$ TMD mass ratio
$\omega^2_d = k_d/m_d$ Squared natural frequency of TMD
$\omega^2 = k/m$ Squared natural frequency of structural system
$\zeta_d = c/c_c = c/2m\omega_d$ Damping ratio of TMD

Figure 4, shows a plot of R as a function of the frequency ratio $\beta$ for $\alpha = 1$ (tuned case), $\mu$=0.05, and for various values of TMD damping ratio $\zeta_d$

Without structural damping, the response amplitude is infinite at two resonant frequencies of the combined structure/TMD system. When the TMD damping becomes infinite, the two masses are virtually fused and the result is an SDOF system with a mass of 1.05m so that the amplitude at resonant frequency becomes infinite again. Therefore, somewhere between these extremes, there must be a value of $\zeta_d$ for which the peak becomes a minimum.

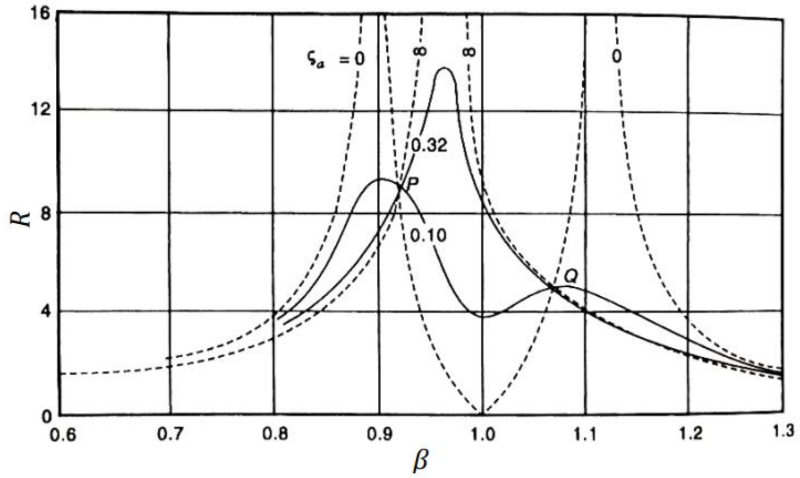


**Figure 4.** Amplification Factor as a function of β (Soong & Dargush, 1997)

There are two points (P and Q) in Figure 4 which is independent of damping ratio $\zeta_d$ and the minimum peak amplitude can be obtained by first properly choosing $\alpha$ to adjust these fixed points to reach equal heights. The optimum frequency ratio $\alpha$ following this procedure is determined as

$$\alpha_{opt} = \frac{1}{1+\mu} \tag{2}$$

Which gives the amplitude at P or Q

$$R = \sqrt{1+\frac{2}{\mu}} \tag{3}$$

A good estimate for $\zeta_{opt}$ can be determined as the average of two values which make the fixed points P and Q maxima in figure 4 gives

$$\zeta_{opt} = \sqrt{\frac{3\mu}{8(1+\mu)}} \tag{4}$$

The maximum amplification factor and optimum absorber parameters are summarized in Table 1 for a variety of excitations and response quantities density is assumed.

**Table 1.** Optimum Absorber Parameters attached to undamped SDOF Structure (Warburton, 1982)

| Case | Excitation | | Optimized absorber parameter | |
|---|---|---|---|---|
| | Type | Applied to | $\alpha_{opt}$ | $\zeta_{opt}$ |
| 1 | Periodic Force | Structure | $\frac{1}{1+\mu}$ | $\sqrt{\frac{3\mu}{8(1+\mu)}}$ |
| 2 | Acceleration | Base | $\frac{\sqrt{1-\mu/2}}{1+\mu}$ | $\sqrt{\frac{3\mu}{8(1+\mu)(1-\mu/2)}}$ |
| 3 | Random Force | Structure | $\frac{\sqrt{1+\mu/2}}{1+\mu}$ | $\sqrt{\frac{\mu(1+3\mu/4)}{4(1+\mu)(1+\mu/2)}}$ |
| 4 | Random Acceleration | Base | $\frac{\sqrt{1-\mu/2}}{1+\mu}$ | $\sqrt{\frac{\mu(1-\mu/4)}{4(1+\mu)(1-\mu/2)}}$ |

## 3.2 Mass of TMD

The mass ratio $\mu$ of the TMD mass to the kinetic equivalent structural mass has to be sufficient. For small ratios ($\mu \leq 0.025$ ) big vibration amplitudes of the TMD mass relative to the structure are resulting. This can create a space problem for proper integration of the TMD in the available structural gap, but also the TMD gets usually much more expensive due to more and bigger springs.
In addition, a small mass ratio is decreasing the effective range of the TMD. The TMD mass movements are significantly smaller for bigger ratios ($\mu \geq 0.025$) and the effective range for a 100% TMD efficiency around the resonance frequency is greater.

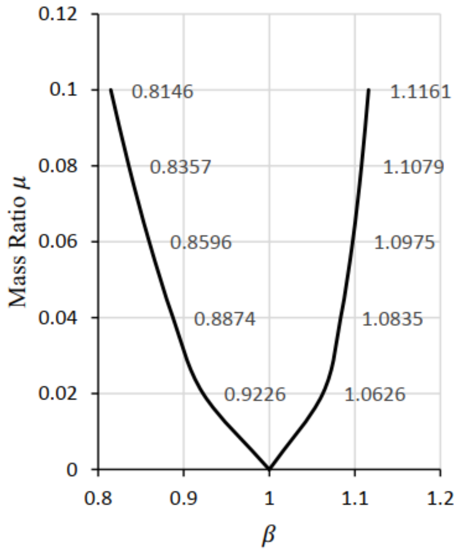


**Figure 5.** Frequency range with respect to μ

# 4 Applied example

One example will be used in this paper. 2D frame three stories will be used for a preliminary TMD application study using FEA software as shown in figure 6a.

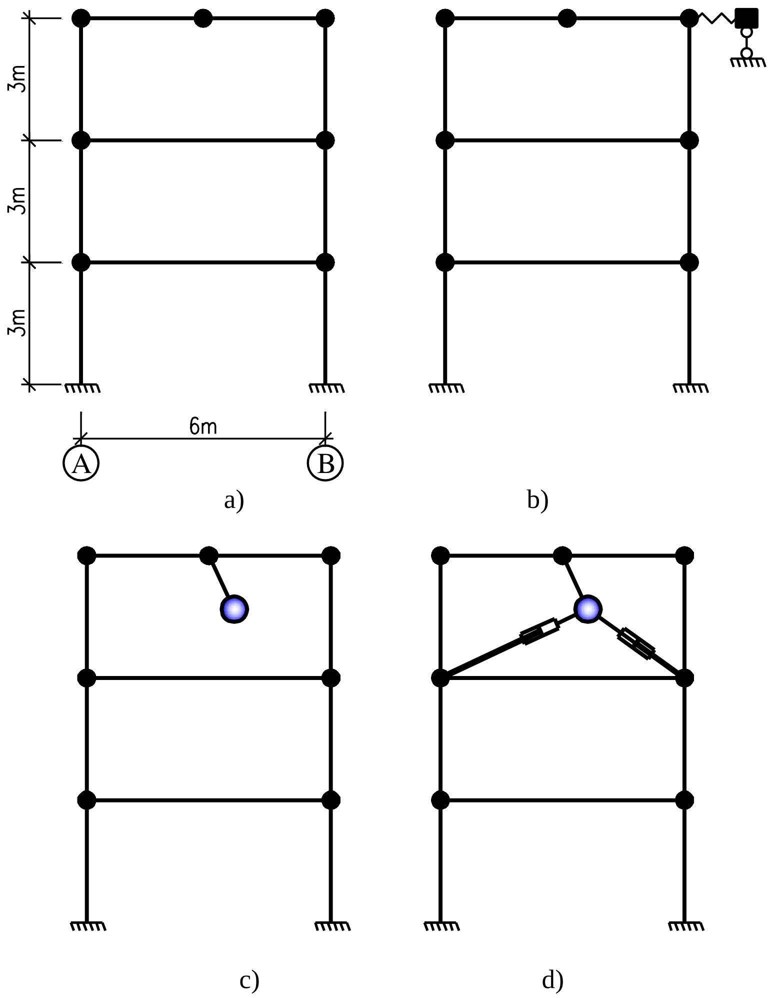


**Figure 6**. a) Frame model; b) Frame with TMD
c) Frame mounting a pendulum without damper; d) Frame mounting a pendulum with damper

**Table 2:** Model parameters

| | Elastic modulus(MPa) | Section size | Mass |
|---|---|---|---|
| Beams | $27\times10^3$ | 20cm×60cm | 0 |
| Column | $27\times10^3$ | 20cm×30cm | 0 |

A tuned mass damper is attached to the structure on the top floor, in three cases as shown in figure 6. Model parameters are shown in Table 2. The mass at each node is 25 tons. A sinusoidal force with amplitude 5000N was applied to the Frame with period T=1.13s.

The parameters of TMD will be based on Den Hartog optimization criteria. The mass ratio for the TMD will be 0.04 for this paper. Depending on the available space, a larger mass can be used where it will increase the efficiency range. Then optimum frequency ratio $\alpha_{opt} = 1/(1+\mu)$ can be calculated. The frequency ratio gives up the frequency of the TMD which can be used to calculate the spring constant $k$ of the system. The damping ratio of the TMD will be calculated by using the mass ratio $\zeta_{opt}$ in(4). The damping coefficient can be found by using the damping ratio $c = 2\zeta_{opt} m\omega$.

The parameter of TMD will be calculated for 3 cases as shown in Figure 6

***Case Figure 6b***

Mass ratio chosen to be $\mu$=0.04 so mass of TMD $m_d$ =10.8 tons

Optimum frequency ratio $\alpha_{opt} = 0.962$

Frequency of damper $f_d$ = 0.882 Hz. $\omega_d$ = 5.54rad/s

Spring stiffness constant $k_d$ = 33.15 ton/m

Optimum damping ratio $\zeta_{opt} = 0.1155$

Damping coefficient $c = 1.382$ *ton.s/m*

***Case Figure 6c***

Install a pendulum with a mass of 10.8 tons and a length of 0.3m. With these parameters, the natural frequency of the pendulum is also approximately the structure's natural frequency.

***Case Figure 6d***

Install the pendulum similar to case 2 but with additional dampers. The damper is linear and has a damping coefficient of 1500 Ns/m.

***Result***

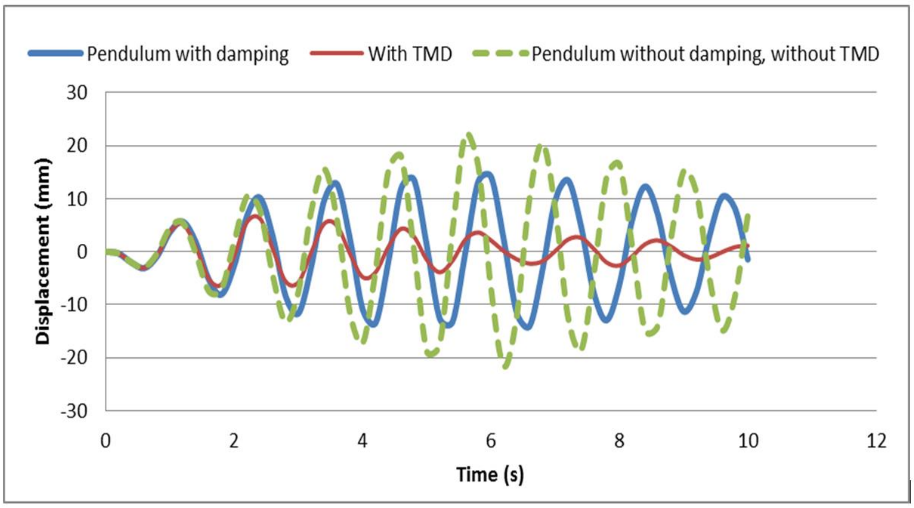


**Figure 7**. Displacement over Time of Frame at the top floor.

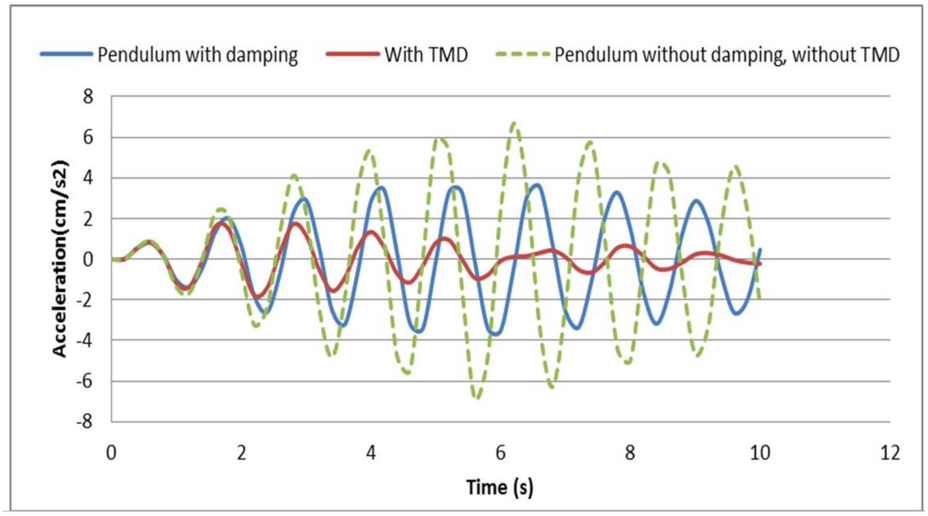


**Figure 8.** Acceleration over Time of Frame on Top floor

**Remark**

In general, both TMD and PTMD can reduce the vibration of the structure. However, the TMD case still has a better effect on reducing vibration than the PTMD case. Theoretically, undamped devices (as in case 2) can only suppress oscillations in a very narrow frequency domain. In this example, given a sufficiently wide frequency domain of the load, the case pendulum without damping does not effectively reduce vibration.

## 5 Conclusions

This paper introduced the most popular device for passive control of structures, TMD. Practice calculating the optimal parameters of TMD by Den Hartog's optimization criteria and applying TMD to the three-story frame structure for three cases. The results of numerical modeling using Sap2000 software have proved to be effective in reducing vibrations for the structure when TMD is installed. Of the three cases of TMD attachment, the TMD case gave the best vibration reduction results, the case of PTMD without damped did not have a significant effect. Therefore, to increase the effectiveness of PTMD in practice, it is necessary to add a damper to this device.

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