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DAMPING ANALYSIS OF A METALLIC STRUCTURE PROTOTYPE WITH A PENDULUM TUNED MASS DAMPER

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Abstract. *High-rise buildings have been becoming higher than never in the last years. Together with this condition came the dissemination of the Pendulum Tuned Mass Dampers (PTMD) as one of the main devices to mitigate induced vibration in this type of structures. Many studies approach variants of control methods, such as passive control, semi-active and others as well as methods for optimized definition of the variables that governs the dynamic behavior of the structure. Despite the technology applied in this type of damping system, its working principle obeys the most fundamental concepts of mechanical vibration. This paper evaluates in a simple manner the dynamic response of a slender metallic frame structure prototype under unidirectional free vibration. The prototype built, followed the design guidelines simulated by FEM (Finite Elements Method). The comparison of the time history acceleration, between the structure without the PTMD and post-implementation of it, shows a significant improvement in structural damping.*

Keywords: *Pendulum, Damper, Vibration, Structure, TMD*

1. INTRODUCTION

The pendulum tuned mass dampers have gained attention within the civil engineering by its application to mitigate vibration in high-rise buildings, especially those highly susceptible to vibration induced by wind loading and seismic events. The intrinsic flexibility of such slender structures can cause problems that range from early fatigue of the structural elements to discomfort of its occupants, this last one being directly correlated to the acceleration resultant from the structure motion that usually is of higher intensity on the top floors (Gunawardena et al., 2017).

The PTMD is a secondary oscillator coupled to a primary oscillator (structure). Its working principle is based on an inertial force generated in opposite direction of the structure displacement. This force is achieved through a heavy mass that is usually positioned at the top of the building and suspended by steel cables, thus making a simple pendulum (Constantinou et al., 1998). The Figure **1** shows the equivalent lumped mass models and a real application example.

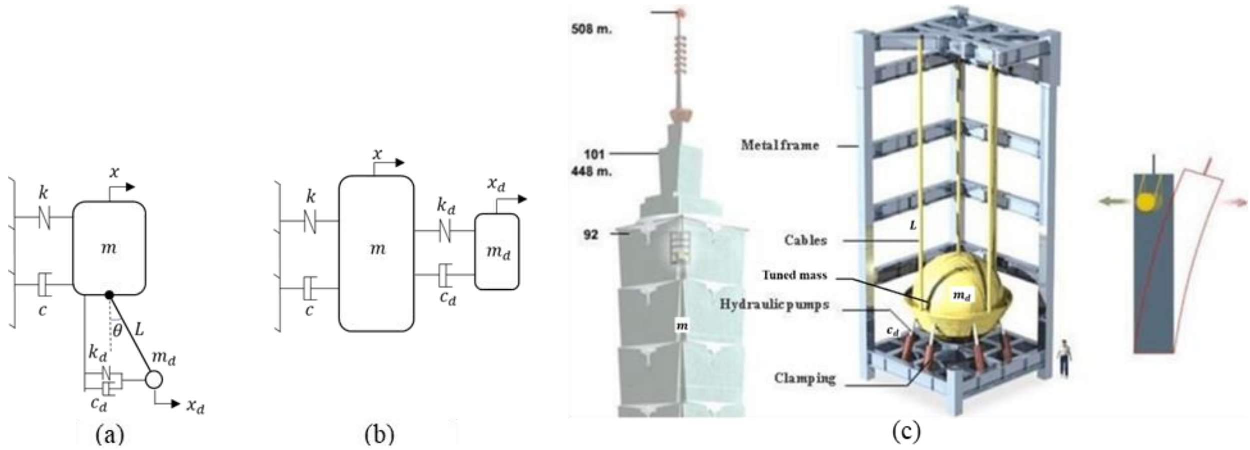


Figure 1**.** Lumped mass model of a structure with a PTMD: (a) Pendulum detailed, (b) Simplified system. (c) PTMD applied to Taipei 101 building. Adapted from (Teplyshev et al., 2018). Licensed under

An analysis of this type of damping system can be performed in several ways, for example, imposing harmonic forces upon the structure as demonstrated in the study of (Lourenco, 2011) for an adaptative PTMD. Another way could be to subject the structure to a saw-tooth wave excitation as investigated by (Carlot, 2012) on the dynamic response under action of wind loading. Within this context of methods of analysis, a numeric approach through FEM (Finite Element Method), provided by the CAE (Computer Aided Engineering) software, comes as a powerful tool for a direct study of this kind of structures with PTMD in order to achieve the best and more efficient design.

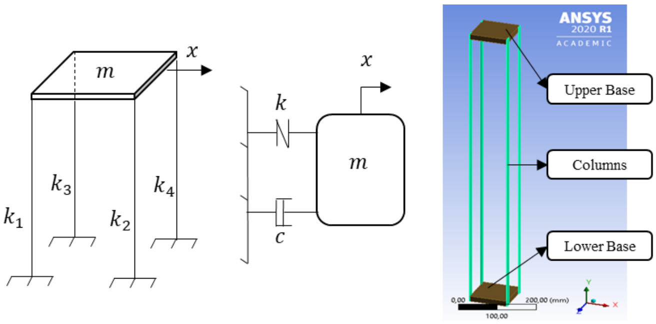
This work takes a simplified approach to demonstrate the fundamental working principle of a PTMD. It proposes an experimental unidirectional free vibration test on a slender metallic frame structure prototype to evaluate its dynamic behavior before and after implementation of the PTMD. The building of the prototype was achieved by having the design guidelines as reference that were analyzed through simulations with numerical models. The simulations performed uses the classical FEA (Finite Elements Method) approach to solve the equation of motion Eq. (1) of a dynamic system modelled as a 3D geometry and discretized in finite elements as detailed by (Filho, 2005). This mathematical model is fully implemented in the ANSYS® Student Version software which was used in this work.

$$[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{F(t)\} \tag{1}$$

The results of the simulations revealed the desired dynamic behavior to be analyzed with the structural members chosen for design, where the structure should be a stable system whilst allowing for enough bending to be applied during the tests on the prototype. Thus, calculation of the vibration frequency and damping rate parameters could be done for comparison through application of the fundamental equations for simple harmonic motion with damping, detailed by (Rao, 2017).

2. BASIC 3D GEOMETRY AND DESIGN CONSTRAINS

The numeric-experimental study began from the design of a slender metallic tridimensional frame structure as a basic 3D geometry. Amongst the possible frame layouts, one with a block suspended by four columns has been chosen, refer to Figure 2c. It represents the frame and lumped mass models in Figure 2a and Figure 2b that also shows the variables related to the structure stiffness, damping, mass and displacement (*k*, *c*, *m* and *x*, respectively). Design constraints were adopted for the basic geometry in order to limit its size to the maximum dimensions of 150 x 150 x 1000 mm in volume and to have the first mode natural frequency (bending) between 0.9 and 1.10 Hz. The modelling of the basic 3D geometry was done in DesignModeler® where it enabled to parameterize the main geometry dimensions to stablish the design points (input parameters) (ANSYS, 2019a); the list of defined parameters and its initial values is shown in Figure 2d.



(a) (b) (c)

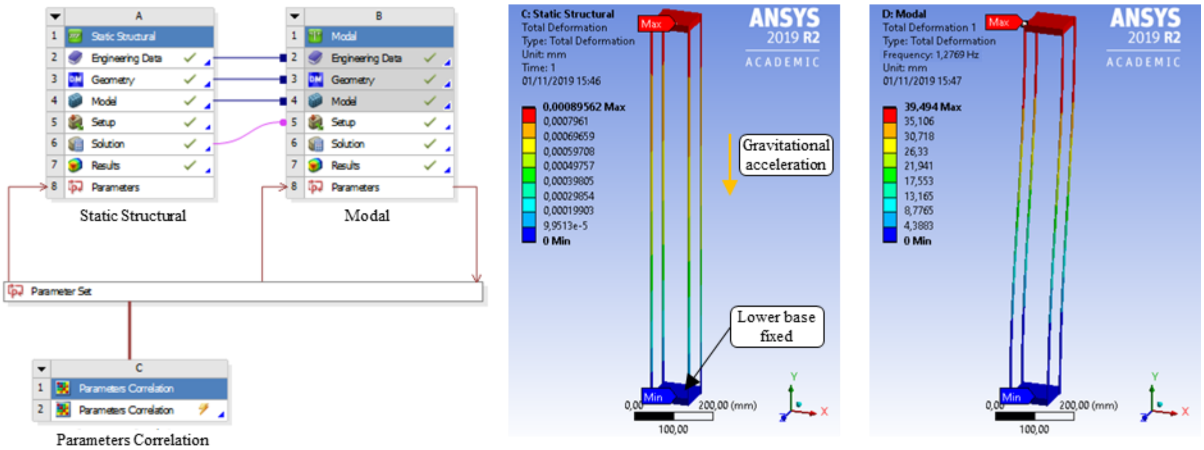
Name	Value	Type
Base_Lenght_X	100 mm	Length
Base_Width_Z	100 mm	Length
LowerBase_Thickness	19 mm	Length
UpperBase_Thickness	6,3 mm	Length
Column_Thickness	2,1 mm	Length
Column_Width	12,7 mm	Length
Structure_Total_Height	1000 mm	Length

(d)

Figure 2. (a) System model, (b) lumped mass model, (c) basic 3D geometry, (d) design points

3. MODAL ANALYSIS SIMULATION

To determine the vibration modes of the structure that was initially modeled as the basic geometry (Figure 2c), a modal analysis simulation was applied, as per Ansys Workbench schematic shown in Figure 3a. In the pre-processing definitions for the simulation it was neglected any effects of energy dissipation (undamped system), but it was considered the stresses (pre-stress) due to the action of the gravitational acceleration. These stresses results were imported from the solution of a static analysis simulation (block Static Structural shown in Figure 3a). Regarding the materials applied, it was set aluminium alloy from the standard Ansys material library. The simulation was also defined with a boundary condition by applying a fixed support on the lower base, refer to Figure 3b. In this analysis it was of interest only the first natural mode and its frequency, in which case the solution computed $f_1 = 1.2769\ Hz$ (bending mode in the x-direction), refer to Figure 3c.



(a) (b) (c)

Figure 3. (a) Workbench schematic for modal analysis, (b) boundary conditions and result in static analysis, (c) result for the first mode of vibration

4. PARAMETERS CORRELATION ANALYSIS

To install the PTMD into the structure employing the dashpots of Figure 11 and with the system layout shown in Figure 5a, a design modification was required on the geometry basic dimensions. From the solution of the modal analysis simulation, describe in the previous section, it enabled to set the first mode frequency as a design point (output parameter) that together with the input parameters defined in the basic geometry modelling (Figure 2d), makes a set of dependent and independent variables. Through the use of the Ansys Design Exploration tool, the parameter set of design points was submitted to a correlation analysis computed in the Parameters Correlation module, as per schematic shown in Figure 3a. This resource permitted to verify the most suitable dimension (input parameter) to be modified whilst keeping the frequency within the constraint. In this manner the analysis was executed with the default module configuration for a Spearman correlation (ANSYS, 2016b, 2016c). Amongst the most sensible parameters found, that is, those in which had more influence over the vibration frequency when changed, it was only feasible to modify the *Base_Length_X* parameter. The choice considers the design constrains as well as to not compromise the columns stiffness and the structural stability.

Following with an analysis of the type what if (ANSYS, 2016a), the modification was set to the value *Base_Length_X = 150 mm*, detail shown in Figure 4, resulting on a new first mode frequency $f_1 = 1.149\,Hz$.

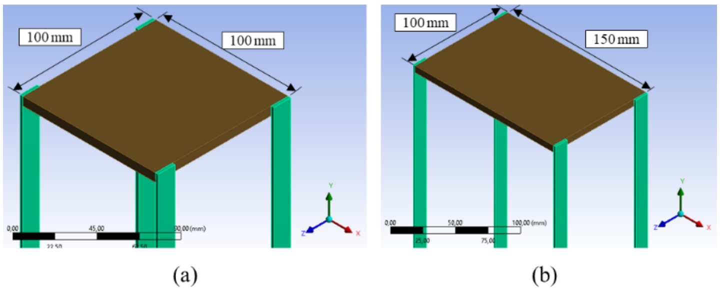


Figure 4. Basic dimensions. (a) Initially adopted, (b) Modified

5. GEOMETRY AND MODEL WITH THE PTMD

To implement the PTMD illustrated in Figure 5a, fitting components had to be included in the basic geometry from Figure 4b. The PTMD system should be installed bellow the top of the structure, therefore two hanging supports attached onto the upper base were modelled for anchoring and positioning of the dampers, refer to Figure 5b. Both supports provide adjustment of its horizontal and vertical position in a manner to cover enough space to accommodate the dampers and the required pendulum length Figure 5a. In this manner, the simple pendulum period equation, Eq. (2), could be applied to determine the initial pendulum length.

$$T = 2\pi\sqrt{\frac{L}{g}} \Rightarrow L = \frac{\left(\frac{1}{f_1}\right)^2 g}{4\pi^2} \Rightarrow L = \frac{1}{{f_1}^2} \cdot \frac{g}{4\pi^2} \tag{2}$$

Therefore, in order to tune the pendulum oscillating frequency to the structure vibration frequency of Figure 4b, a length $L = 188\,mm$ would be required, in this way, the designed supports had to cover this minimum length. However, recalculation of this dimension to consider the increase in mass because of the added supports that also changing the vibration frequency, is described in the next section. With this modification the geometry model for the structure without the PTMD is complete, Figure 5c.

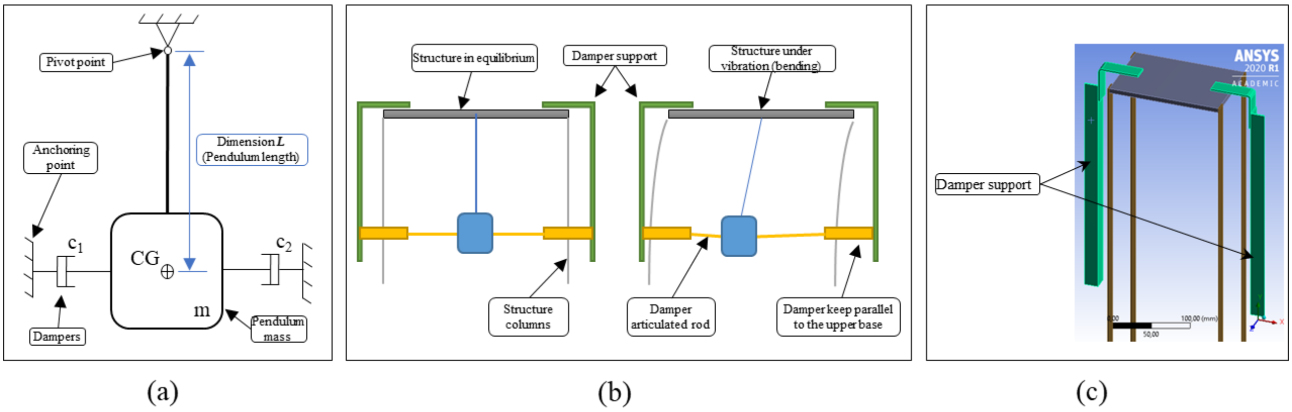
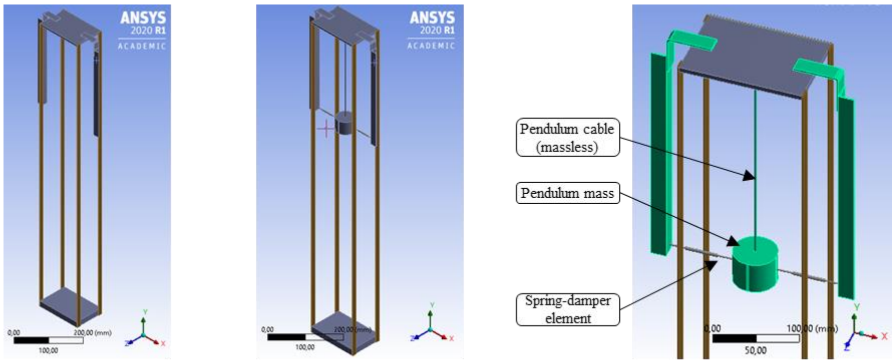


Figure 5. (a) PTMD system, (b) dampers and supports functionality, (c) structure design without the PTMD

From the geometry without the PTMD (Figure 6a), a new one was copied in order to simulate the structure with the PTMD implemented (Figure 6b). The PTMD system was modelled by two spring-damper elements and two solids, one to represent the pendulum mass (counterweight) and the other to represent the cable that holds it, refer to Figure 6c. Both solids were attributed with material properties of Structural Steel (ANSYS material library), but for ease of design control over them, a modification in the material density had to be imposed. By setting the material density to $\sigma = 1 \cdot 10^{-19}\ kg/m^3$, a direct control of their total mass was possible, independently of the solid volume (Siqueira, 2014). In this manner, the pendulum had a $m_d = 0.350\ kg$ applied as distributed mass on all its surfaces and the pendulum cable kept as a solid of neglected mass.



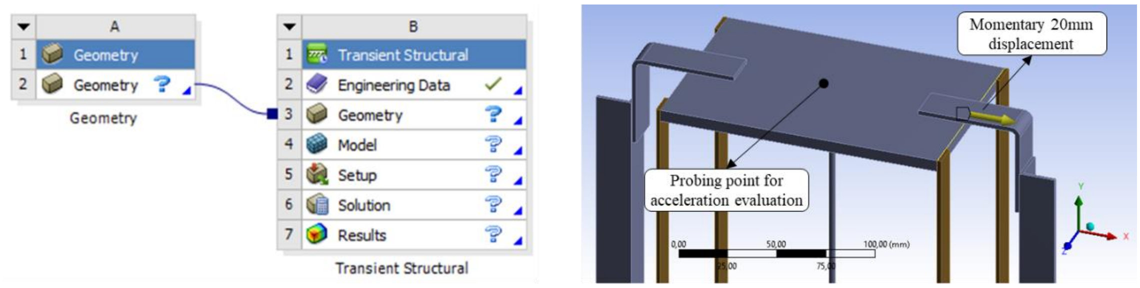
(a) (b) (c)

Figure 6. (a) Geometry without the PTMD, (b) geometry with the PTMD, (c) PTMD model implemented

6. FREE VIBRATION SIMULATION

Both geometry designs (Figure 6a and Figure 6b) were subjected to a free vibration simulation using the Transient Structural analysis, the Ansys Workbench schematic is shown in Figure 7a. Regarding the masses of structure members, it all had to be redefined in accordance with the real measurements retrieved during the building of the prototype, see Table 1 for the values found.

For this simulation, the same boundary conditions shown in Figure 3b were considered in the pre-processing definitions. The only difference lies in the initial condition adopted in the analysis, where the structure is put to vibrate from an initial and momentary 20 mm displacement applied on the upper base and in the x-direction (Figure 7b) (Siqueira, 2014). Finally, the mesh applied had to be more refined in the columns (Figure 8) since they are the members subjected to bending and direct define the structural stiffness, thus it is of higher influence over the dynamic response of the structure.



(a) (b)

Figure 7. (a) Workbench schematic for transient analysis, (b) Initial condition for free vibration simulation and location for evaluation of the acceleration

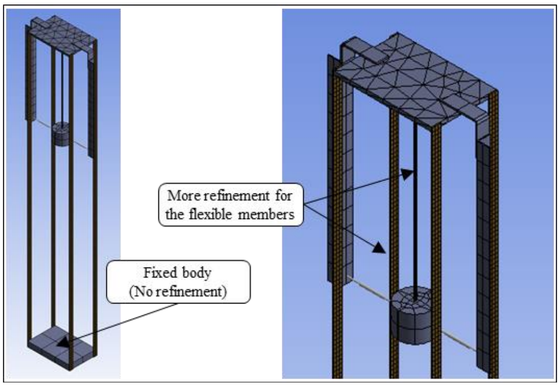


Figure 8**.** Meshed model for free vibration simulation

In order to evaluate the acceleration at the top of the structure, it was made use of the probing resources from the ANSYS Mechanical (ANSYS, 2019b). This feature enabled the data generation and plotting of the time history acceleration in relation to the very top of the structure (Figure 7b). Thus, from the first simulation executed with the structure model without the PTMD and the results that were obtained, shown in Figure 9a, the following calculations were applied for determining of vibration properties. By beginning with the average oscillation period:

$$\bar{T} = \frac{\sum_{i=1}^{n} \Delta t_i}{n} \quad (3)$$

The calculated average frequency is determined by:

$$\bar{f} = \frac{1}{\bar{T}} \quad (4)$$

Also, the logarithmic decay can be expressed as:

$$\delta = \frac{1}{n} \ln \frac{x_1}{x_{n+1}} \quad (5)$$

Thus, the damping rate is:

$$\zeta = \frac{\delta}{\sqrt{\delta^2 + 4\pi^2}} \quad (6)$$

Therefore, the calculated average frequency for the structure without the PTMD was $\bar{f}_1 = 1.024\ Hz$, and the damping rate for that mode was $\zeta = 0.0003$. Based on these values determined, the required pendulum length to tune it to the structure frequency is $L = 237\ mm$, as per Eq. (2). By attributing the pendulum length to the structure model with the PTMD, the same simulation was performed, but now considering the PTMD elements. In this sense, the spring-damper proprieties were set to a longitudinal stiffness of $k_d = 1 \cdot 10^{-6}\ N/mm$ and damping constant $c_d = 0.001\ Ns/mm$. The simulation result for the structure with the PTMD implemented is presented in Figure 9b.

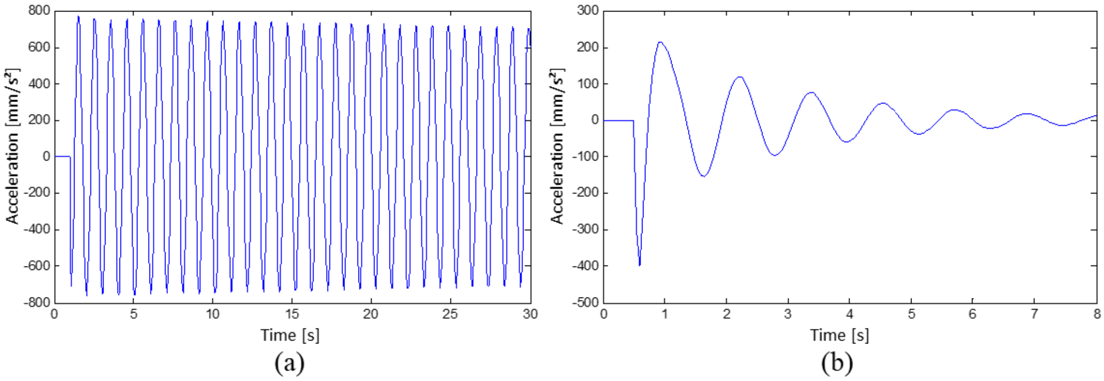


Figure 9. Simulation results. (a) Model without the PTMD, (b) model implemented with the PTMD

7. PROTOTYPE AND EXPERIMENTAL TEST

Based on the structure model, shown in Figure 6a, the prototype was built employing aluminium alloy 6063 as the material to all structural members and the assembly was done with bolts, nuts and washers, Figure 10a shows the prototype without the PTMD. The PTMD system is composed by a hollow brass cylinder calibrated as the pendulum mass ($m_d = 0{,}350\ kg$) and a segment of common string to serve as a cable, Figure 10b. The system also includes two dashpot (air dampers) with adjustable damping constant from 0 to 0,088 Ns/mm, refer to Figure 11. For design control, a verification of the masses of each part had to done and the measured values are presented in Table 1.

Table 1. Masses of the prototype without the PTMD

Item	Mass [kg]	Includes
Column (each)	0,070	-
Upper assembly	0,504	Upper base, supports, sensor and accessories
Pendulum mass	0,350	-

The acquisition of the acceleration data was achieved through an apparatus composed by a motion processor unit (model MPU-6065) and an Arduino board. The sensor was mounted on top of the structure to capture the acceleration in the same location as analyzed in the free vibration simulations (Figure 7b). To set the initial condition (upper base displaced), a segment of common thread was used to hold the structure in the deformed position guided by a regular ruler as shown in Figure 10c. By cutting of the thread, the structure could freely vibrate.

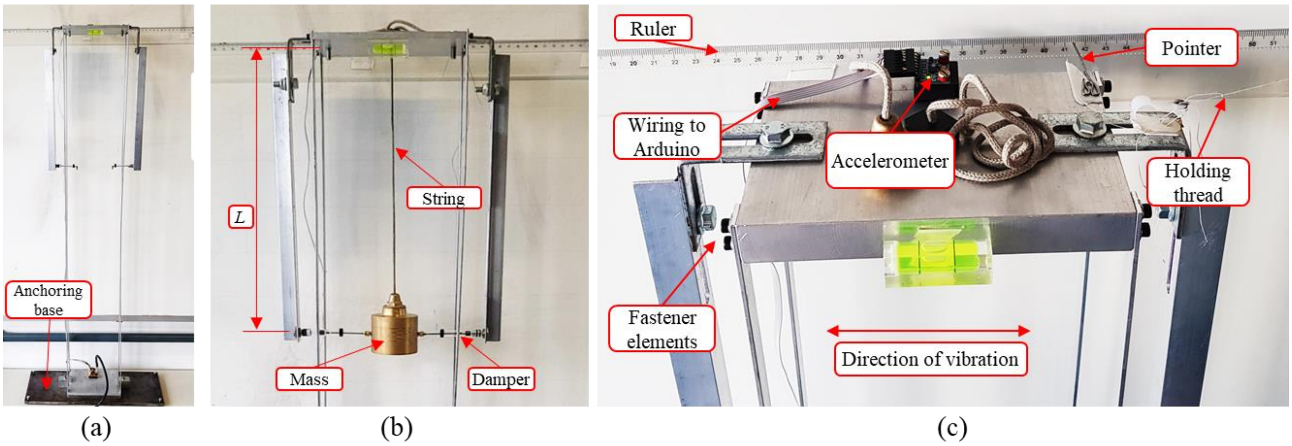


Figure 10. (a) Prototype without the PTMD, (b) prototype with the PTMD, (c) experimental setup

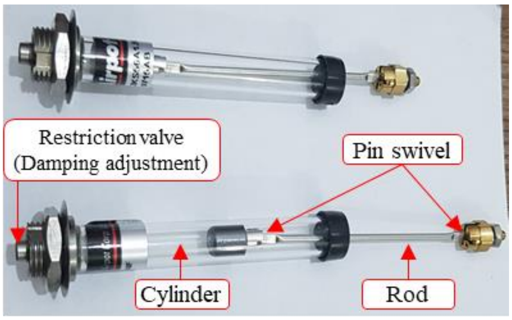


Figure 11. Dashpot (Air Damper) – Model Airpot 2KS56

The result of the first test with the prototype without the PTMD is presented in Figure 12a. Subsequently, using Eq. (3)(4)(5)(6), the calculated frequency was $\bar{f}_1 = 0.901\ Hz$ and the damping rate was $\zeta = 0.0014$. Based on that, the required pendulum length, as per Eq. (2), was $L = 306\ mm$. Once with the PTMD fitted into the structure, the dampers were adjusted through their restriction valve by closing it in 1% (approximately $c_d = 0{,}001\ Ns/mm$). The test was then repeated and the result regarding the prototype implemented with the PTMD is shown in Figure 12b.

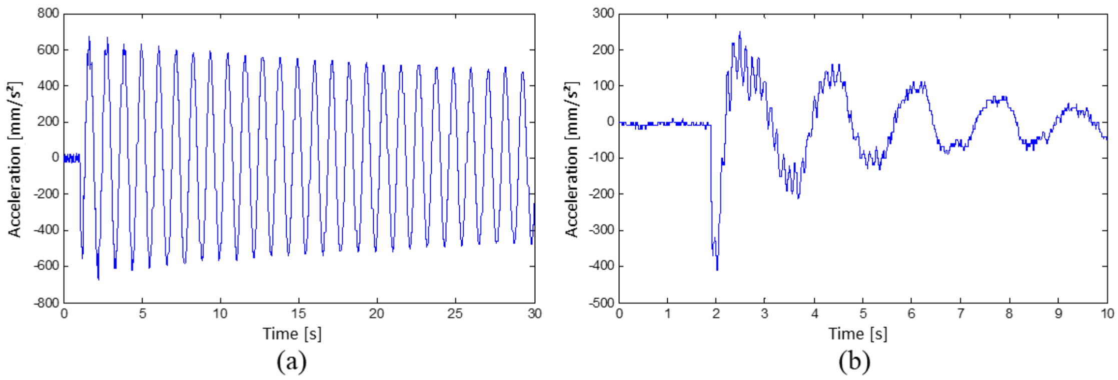


Figure 12. Experimental test results. (a) Prototype without the PTMD, (b) prototype implemented with the PTMD

8. RESULTS AND DISCUSSION

The Table 2, shows a summary of the vibration parameters calculated based on the numeric simulation and the experimental test results. The first important point to be noticed is regarding the percentage error of the vibration frequency calculated from the experimental test in relation to the numerical simulation. For both structure configuration, a more refined finite element mesh in the numerical model should be considered to yield a lower error, that is because the ANSYS® Student Version which was used limits the number of elements and nodes for meshing. Despite that, the results are still valid to analyze the influence of the PTMD.

Regarding the dynamic behavior of the prototype without the PTMD, it can be observed that there is structural (hysteretic) damping, however it corresponds to a damping rate smaller than 1%. Such rate is solely responsible for the time taken by the structure to achieve at least 50% reduction of the initial amplitude (more than 30 seconds). These values enable for this structure design an approach of undamped system in the numerical model, but limited to the maximum displacement tested (20 mm). However, more severe conditions like higher frequencies or bigger displacements should be take into account the frictional effects on all contact faces between the structural members as well as non-linear solution method, all defined in the pro-processing configuration. These assumptions combined with a more refined finite element mesh should bring the simulation result (Figure 9a) to a value closer to that of the prototype (Figure 12a).

With the implementation of the PTMD, a significant increase in damping was achieved, not only by the shorter time to dissipate the kinetic energy, but also the decrease of the maximum amplitude. The level of damping observed can be directly correlated to the mass ratio of the structure-PTMD system, as this design employed a pendulum mass equivalent to 69% to that of the structure mass. Such ratio does not reflect what is practiced for real constructions, in which it does not exceed 5% (Connor & Laflamme, 2014). Also, the signal noise on the data, shown in Figure 12b, should not be of great importance over the result as its source may be related to an internal signal disturbance of the motion processor unit itself.

Finally, looking at the data for the structure configuration with the PTMD (Table 2), it is noticed that the percentage error is also bigger than that of the configuration without the PTMD. Such value must be mainly due to the fact that the dashpots did not had a precise adjustment of its damping constant. Therefore, the level of damping for the pendulum mass must be a critical parameter for proper working of the PTMD and a deeper analysis of it should be considered.

Table 2. Summary of the calculated parameters

Structure configuration	Numerical model				Experimental prototype				Frequency [% Error]
	$\bar{T}_1[s]$	$\bar{f}_1[Hz]$	δ	ζ	$\bar{T}_1[s]$	$\bar{f}_1[Hz]$	δ	ζ	
Without PTMD	0.98	1.024	0.0020	0.0003	1.11	0.901	0.0089	0.0014	12.0
With PTMD	1.16	0.862	0.4893	0.0776	1.43	0.701	0.2356	0.0375	18.7

9. CONCLUSION

The study of the dynamic response under unidirectional free vibration for the structure model proposed, presented the expected result after implementation of the PTMD, that is, a significant increase in structural damping was achieved. It shows that the PTMDs are in fact a possible solution to mitigate vibration in slender structures. The simulation by finite elements techniques utilized in the ANSYS® Student software, provided a direct mean to evaluate the dynamic behavior of the system prior to the building of the prototype.

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