

A tuned mass damper with lightweight plate configuration for vibration suppression at targeted natural frequency

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ABSTRACT

Previous research shows that using a plate as the stiffness of a tuned mass damper (TMD) and no added mass to a beam-like TMD are feasible as a damper. This research combines those two concepts by using a plate model as a new TMD (Plate TMD). The absence of auxiliary mass makes the model lighter and simpler than conventional models. This study also uses a new method to adjust the TMD natural frequency by conducting a parametric design study of the plate's outer diameter and hole spoke width with a circular plate shape. The study has shown that the target natural frequency of the plate can be obtained with a lighter mass in the minimum spoke width selection. This model was applied to a beam-like cable model for simulations and to an all-aluminum alloy cable for experiments. The study used finite element method (FEM) simulations and experiments with an electrodynamic shaker. The results show that the Plate TMD model is feasible and can attenuate the vibration at the target natural frequency. In this study, the response amplitude ratio decreased by more than 93 percent.

Keywords: design study, natural frequency, plate model, tuned mass damper.

INTRODUCTION

Tuned mass dampers (TMD) play a crucial role in controlling structural vibrations [1,2]. Structures can experience excessive vibration if resonance occurs [3]. One such structure is a cable structure. Vibrations in cables are generally caused by the excitation force caused by the wind hitting them. These wind-induced vibrations are called aeolian vibrations. Aeolian vibrations typically range from 5 to 200 Hz [3,4]. Cables have many natural frequencies. Resonance will occur at certain disturbing frequency ranges. Deformation due to resonance can cause fatigue failure in cables [3,5]. In addition, cable vibrations are also caused by earthquakes, as an excitation of the base motion [6]. This can be detrimental to societies and industries that use cable structures as their infrastructure [7]. The application of cables/beam models as supporting or retaining structures in bridges, or cable-stayed bridges [8], as well as

supports for unbalanced rotating masses [9,10]. Cables are one of the main vibration systems with multiple resonances, in addition to cantilever bars [11], plates [12], and truss structures [13].

TMDs have long been used as passive vibration control devices designed to reduce resonant vibration response by adding a mass-spring-damping system tuned to the natural frequency of the primary structure [14,15]. Physical evidence shows that structures with TMDs experience a significant reduction in vibration response. This reduction occurs because the TMD adds a general degree of freedom to the vibration system, resulting in a lower vibration response at the undamped primary structure's natural frequency. TMD settings can be adjusted depending on the application, such as placement at the point where maximum vibration occurs or distributed for multiple vibration modes. Proper tuning and placement are essential to maximize the effectiveness of TMD applications [16]. TMDs can also be

applied to vibration systems with multiple resonant frequencies.

Application of simple TMD to multimodal vibration systems has demonstrated a reduction in vibration response across multiple modes [17,18]. This happens because dampers decrease the amplitude of vibrations for certain resonance frequencies. However, when you use a basic TMD in a vibration system, it comes with its own problems and is particularly ineffective in dealing with more than one resonant frequency [19,20]. Furthermore, basic TMDs tend to be ineffective in multimodal systems and may cause insignificant reductions in overall vibration control. This creates the need for more advanced TMDs. In the case of multimodal vibration systems, different types of TMDs are needed to deal with different aspects of the complexity.

Several models of TMDs have focused on vibration reduction in multimode systems, including plates [21], shell [22], cantilever beam [11], cable [23], inerter [24], and frictional mass [25]. These models have been successfully implemented to reduce the effects of vibration responses and have been designed to interact with multiple resonance frequencies using different mechanical configurations. The mechanics of these models are based on adjusting the parameters of dampers to engage with certain modes of vibration for a more efficient reduction of the response. Some models have many degrees of freedom [4] or adjustable parameters [26,27] for better adaptation to a wide range of vibration system conditions. This versatility is capable of improving performance for complex vibration systems. The use of TMDs with a plate or cantilever bar model combined with additional mass is an example of this advanced design concept.

Tuned mass dampers that combine plate [28,29] models with additional mass have proven effective in reducing the vibration response in multimode systems. Other TMD models with additional mass use cantilever rods [12,30]. These dampers use a cantilevered plate or bar as a spring element, which interacts with the mass to create a tuned system. This configuration can be precisely tuned for specific vibration modes. Adding mass can increase the inertial effect, thereby improving the damper's ability to resist vibration at the target frequency. A well-known practical application is the Stockbridge damper, widely used on overhead transmission lines, which uses a mass-spring system with additional weights attached

to flexible messenger [31,32]. The Stockbridge damper effectively reduces wind vibration, thus demonstrating the true effectiveness of cantilever-type tuned mass dampers with added mass. Its design principles have inspired numerous adaptations in structural vibration control, emphasizing the importance of mass placement in damper performance [33,34]. Furthermore, the design flexibility of these models allows for customization to specific structural requirements. Tuned mass dampers with plates and cantilever rods, even without added mass, have also been explored for their vibration mitigation capabilities.

TMDs with plate [21,35] and without additional mass have also been able to reduce vibration response. Other TMD models without additional mass also use cantilever rods [36,37]. In these models, the plate and cantilever rod act as both spring stiffness elements and mass elements. Adding a load to the main vibration system increases its deflection. This change in natural frequency can be exploited to better match the damper tuning to the system's vibration characteristics. These models simplify damper design by eliminating the need for separate mass components, potentially reducing cost and complexity. The natural frequency can be adjusted by modifying the dimensions of the plate and cantilever rod.

The dimensions of a TMD with a cantilever plate or bar and no added mass are important for determining its natural frequency [38,39]. Variations in length directly affect the natural frequency and can be tuned for the desired vibration mode. Changes in width and thickness also affect the mass and stiffness, further influencing the tuning process. These geometric parameters provide a design with optimized performance for a specific application.

The application of holes in a plate can effectively reduce its natural frequency. The size and space of these holes affect the plate, both its stiffness and mass, leading to a decrease in the natural frequency [40,41]. This method of modifying plate properties is a practical approach for damper tuning. The addition of these holes offers a lightweight model solution for adjusting the plate's natural frequency. Therefore, the application of holes in the plate is a method that can be used to adjust the natural frequency of TMDs with a plate model.

A perforated plate model using numerous small holes has been shown to reduce the natural frequency and influence the plate's vibration mode [42,43]. The use of numerous holes places

significant emphasis on the spacing between the holes, their shape, and their size. This is due to the limited plate size. This hole model results in a limited range of changes in the plate's natural frequency. Therefore, this study proposes the use of holes with spokes. This allows for a wider range of changes in the outer diameter and spoke width, thus broadening the natural frequency range. In addition, the use of many small holes to obtain the same natural frequency requires a longer plate manufacturing process.

Previous studies have not implemented a circular plate model without additional mass and with the addition of holes. The objectives of this research are to combine the concept of the plate as stiffness and the concept of TMD without additional mass, and to apply a parametric design study to obtain a lightweight TMD model with plate model for a certain target natural frequency. This TMD model is hereinafter referred to as the Plate Tuned Mass Damper (Plate TMD). The application of the hole shape on the circular plate model allows a larger range of hole sizes. In this study, the hole shape uses spokes so that its size changes according to the plate diameter and the spoke width. Without mass and the addition of holes in this new model makes the system mass lighter in general compared to models with added mass and without holes, thereby reducing the load on the main vibration system. In addition, this research presents a new parametric design study on the TMD with this new model, which focuses on targeting specific natural frequencies. Adjusting the plate diameter and spoke width will change the hole and plate sizes in general and also regulate the natural frequency of the Plate TMD with circular shape. The use of this parametric design study can increase the potential for using circular plates which are rarely used as TMD compared to other models, and have never been applied in previous TMD studies. The proposed design is validated experimentally through prototyping and vibration testing, which verifies the simulation results and demonstrates practical feasibility. Finite element simulations are performed to determine the capabilities of this new TMD model on cable models.

METHODS

The cable model is considered the main vibration system in this study using a 16 mm² All Aluminum Alloy Cable (AAAC). This small-diameter

cable was selected as the test was done on a lab scale with limited space in which the maximum response values (resonance at low frequencies) could be measured with the available sensors. Nevertheless, this AAAC represents the main vibration system in which resonance can also be studied and which is in the condition and dimensions of the cable. This representation of the cable enables a more focused study of the vibration of a multimode system. The AAAC has a set of unique mechanical properties that help in establishing a realistic assessment of the effectiveness of the tuned mass damper. Therefore, by using the model of the cable, this study helps in assessing the effectiveness of the damper in dealing with the problem of the inherent natural frequencies of the cable structure. The verification of the material properties includes the measurement of density using digital scales for an accurate calculation of the mass. The modulus of elasticity of the material is determined by some mechanical test and is obtained by a Universal Testing Machine which provides adequate information on the elastic behavior of the material. Generally, Poisson's ratio is considered to be 0.33 for aluminum and its alloys [44]. Table 1 shows dimension and material characteristics of cable models for simulation.

In vibration analysis, the model of a circular cross-section beam with both ends fixed serves as a simplified representation of a cable. These fixed end boundary conditions provide a better estimation of the cable's behavior in a structural system where the cable is firmly fixed and will not allow the ends of the cable to move or rotate. The fixed-fixed beam model is useful in determining the natural frequencies and the mode shapes of the system.

The beam model is analyzed in ANSYS using the Harmonic Response analysis and a gravitational load is used as a pre-stress. The gravitational force was applied in the Y-direction to all the elements. The experimental study provides data about the cable's vibration. The first, last, and interval frequencies are kept the same for both the simulation and the experiments. For both the simulation and the experiments, the amplitude ratio of the response with the peak value happens at the resonance condition of the cable. This approach helps in the precise modeling of the dynamic response of the beam. The main objective of the simulation is to determine the natural frequency of the beam of the single-loop model. Table 2 shows the simulation parameters of cable model. The used component of the tuned mass damper

Table 1. Dimension parameters and material characteristics of cable models for simulation

Parameters and properties	Value
Length	0.425 m
Diameter	0.00494 m
Young's Modulus	2781.5 MPa
Poisson's ratio	0.33
Density	2797 kg/m ³

is an aluminum plate (Al1100), which possesses a thickness of 0.001 m, and constructs a circular mass damper without any additional mass. The mechanical and physical characteristics of the material are derived from Al1100 testing through a Universal Testing Machine and digital scales. Table 3 shows the plate model parameters.

This study addresses the influence of plate dimensions on the natural frequency of a system and the effect of such design on the mass of a TMD using parametric design study. This method allows precise adjustment of TMD natural frequency by modifying geometric parameters, such as diameter, and hole size. The study uses simulation and analysis to identify the potential design configurations that effectively reduce vibrations and the load on the structure. The flow-chart (Figure 1) shows the process of the study and a sequence of design, simulation, and evaluation iterations to optimize the TMD in a cable vibration system. The models of the plates, with and without holes, that were used in the study are shown in Figure 2.

In the parametric design study of the Plate TMD, setting the outer diameter of the plate and the width of the spokes requires parametric optimization. This optimization aims to achieve a natural frequency that is as close as possible to the target frequency and at the same time minimizes the mass. In the case of the models with and without holes, the outer diameter specification is the same,

while the width of the spokes is determined only for the model with holes. The preliminary phase of the parametric design study commences with the model that lacks holes to define the outer diameter. Iterate the simulation with a larger outer-diameter interval until the natural frequency ($f_n(D_i=D_a)$) in the iteration sequence falls below the target natural frequency ($f_{n-target}$), as stated in Equation 1.

$$f_n(D_i = D_a) < f_{n-target} < f_n(D_i = D_{a-1}) \quad (1)$$

where: $D_{i+1} = D_i + 0.010$ m; $i=1,2,3,\dots,a$ (iteration i).

Iterate the simulation with smaller outer-diameter intervals until the natural frequency ($f_n(D_j=D_b)$) in the iteration sequence exceeds the target natural frequency, as stated in Equation 2.

$$f_n(D_j = D_{b-1}) < f_{n-target} < f_n(D_j = D_b) \quad (2)$$

where: $D_{j+1} = D_j - 0.001$ m; $j=1,2,3,\dots,b$ (iteration j).

The outer diameter (D) is determined based on the smallest difference between the two natural frequencies from the last two iterations and the target natural frequency, as in Equation 3.

$$D \left\{ \min \left[\begin{array}{l} |f_n(D_j = D_{b-1}) - f_{n-target}| \\ |f_n(D_j = D_b) - f_{n-target}| \end{array} \right] \right\} \quad (3)$$

If this outer diameter exceeds the specified maximum allowable dimension, the study proceeds using the model with holes, in which both the outer diameter and the spoke width are varied.

Table 3. Parameters of the plate model for simulation purposes

Parameters	Value or detail
Thickness	0.001 m
Young's Modulus	13610 MPa
Density	2661 kg/m ³
Poisson's Ratio	0.33

Table 2. Simulation parameters associated with the cable model

Parameters	Value or detail
Length	0.425 m
Mesh method	MultiZone
Mesh element size	1.5e-003 m
Excitation type	Acceleration; Y component; base excitation; one cable model end
Excitation acceleration	9.81 m/s ²
Frequency response scope	1 node; at the center of the cable model
Boundary conditions	Fixed-fixed, 2 faces, all model ends

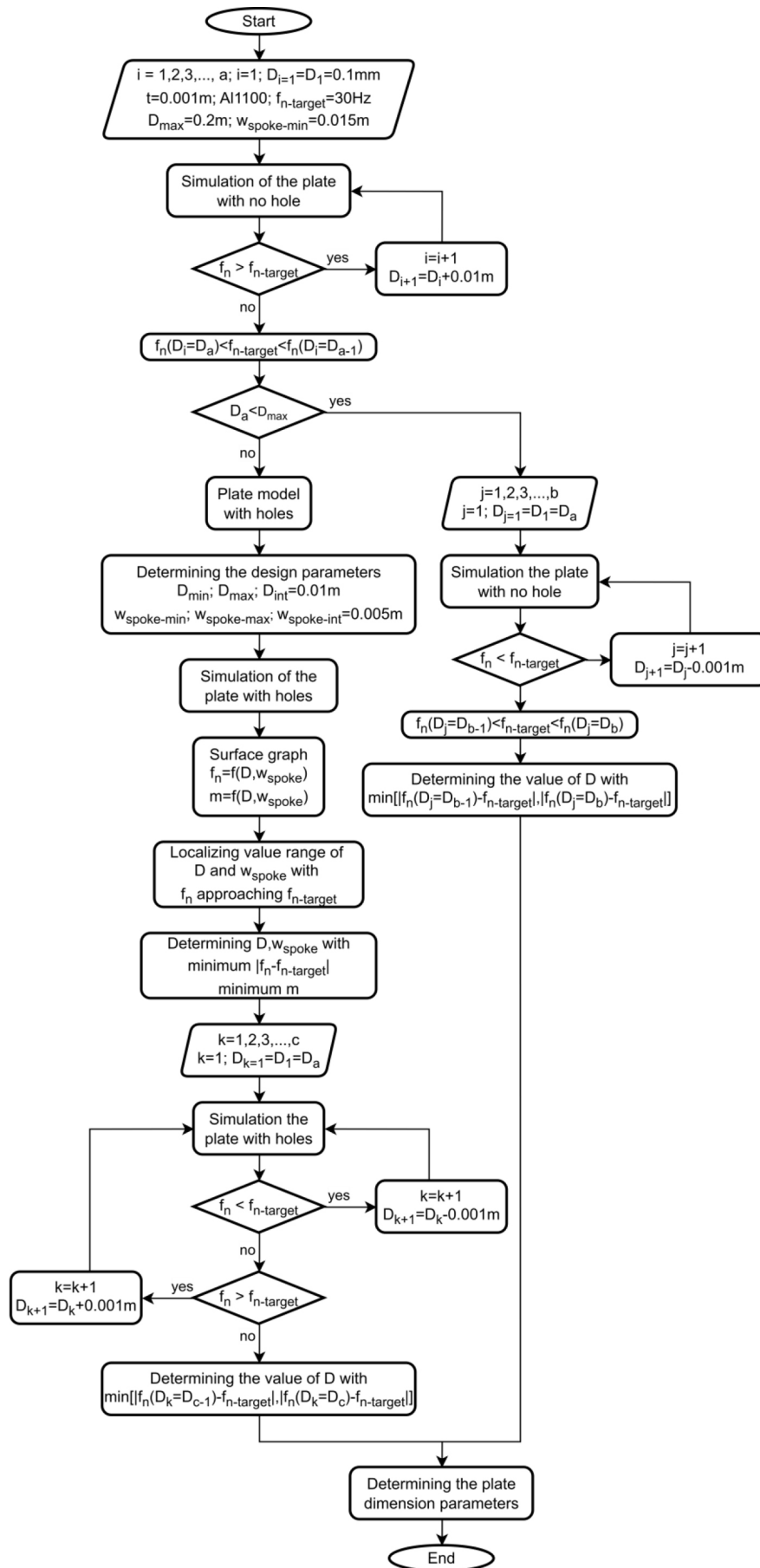


Figure 1. Flowchart illustrating the parametric design study framework

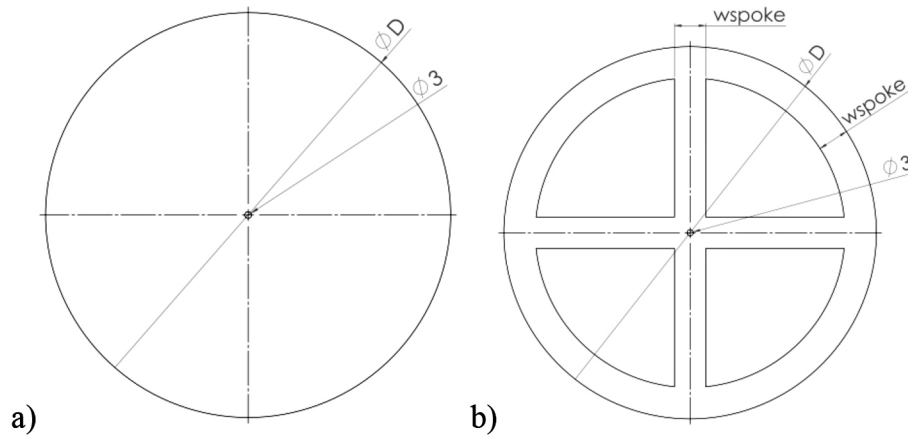


Figure 2. The plate illustrations with a) no holes, and b) holes

The outer diameter and spoke width are selected based on the smallest difference between the natural frequency (f_n) and the target natural frequency ($f_{n-target}$), and the smallest mass value (m), as in Equation 4.

$$D, w_{spoke} \left\{ \begin{matrix} \min(|f_n - f_{n-target}|) \\ \min(m) \end{matrix} \right\} \quad (4)$$

The spoke width uses the minimum value. The outer diameter (D) is determined based on the same principle as before, as in Equation 5.

$$D \left\{ \min \left[\begin{matrix} |f_n(D_k = D_{c-1}) - f_{n-target}| \\ |f_n(D_k = D_c) - f_{n-target}| \end{matrix} \right] \right\}_{w_{spoke-minimum}} \quad (5)$$

where: $D_{k+1} = D_k + 0.001 \text{ m}$ if $f_n > f_{n-target}$; $D_{k+1} = D_k - 0.001 \text{ m}$ if $f_n < f_{n-target}$; $k=1,2,3,\dots,c$ (iteration k).

The simulation for the parametric design study of the plate is conducted using ANSYS software with the Modal analysis system, incorporating gravity load as a pre-stress condition. The simulation outputs include the plate's natural frequency

Table 4. Simulation parameters of the plate model

Parameters	Value or detail
Geometry	Midsurface
Mesh method	Triangles
Mesh element size	1e-003 m
Boundary conditions	Fixed support; inner circle
Connections	Bonded

and directional deformation. The selection of the natural frequency is predicated upon the directional deformation or the mode shape that corresponds to vertical vibration, thereby ensuring its relevance to the objectives of vibration attenuation. This conceptual framework of mode shape has also been employed in prior studies to obtain the desired natural frequency of the cable model [4,45]. Table 4 shows the simulation parameters of plate model. The determination of this mode shape is essential for adjusting the plate to correlate with the vibrational dynamics of the cable, as shown in Figure 3.

The parametric design study is used to select the outer diameter and spoke width of the plate. A

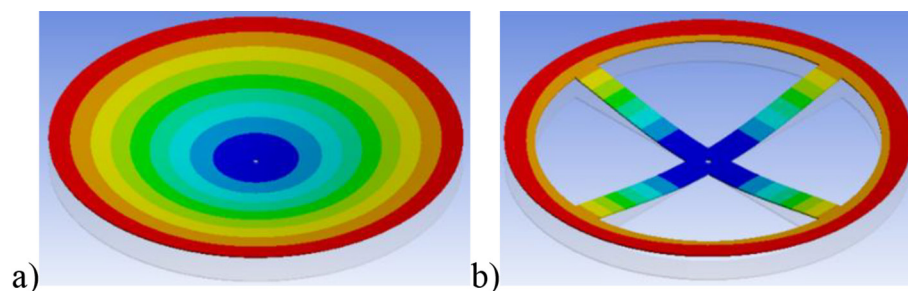


Figure 3. The typical mode shape of the plate with a) no holes, and b) holes, corresponds to vertical vibration

detailed drawing is prepared for the design of the model and is then fabricated for the experiment. The 3D model is also assembled with a beam-like cable model for the purpose of FEM simulation.

The simulation for the cable model with and without the Plate TMD is carried out on ANSYS software and the Harmonic Response Analysis is performed. Parameters of the cable model, dimensions of the TMD plate, and configurations of the simulations are incorporated to maintain coherence. The cable-TMD connection is assumed to be brass, which is defined by Young's Modulus of 1×10^{11} Pa and a Poisson's Ratio of 0.33 to demonstrate the brass's mechanical properties. The connections are all bonded. The simulation focuses on the cable's frequency response, measuring acceleration in the Y axis at the central node of the cable model mesh. The frequency range is set from 10 Hz to 50 Hz to capture the

relevant behavioral vibrations for assessment of the TMD's response to vibration reduction.

The empirical investigation of cable oscillation, both in the presence and absence of a Plate TMD, incorporates the cable's type, dimensions, and material properties, alongside the dimensions and material characteristics of the plate TMD, as determined through the parametric design study. The cable system is fixed at both ends to simulate realistic boundary conditions. The excitation is applied as a base motion acceleration at one end of the cable, produced by an electrodynamic shaker, specifically the Ai Si Li ES-3 model. Vibration response measurements are captured using an Omega ACC-103 accelerometer positioned at the center of the cable to monitor acceleration signals. These signals are transmitted to a PicoScope 2000 series oscilloscope for data acquisition. The setup of the testing equipment is in Figure 4.

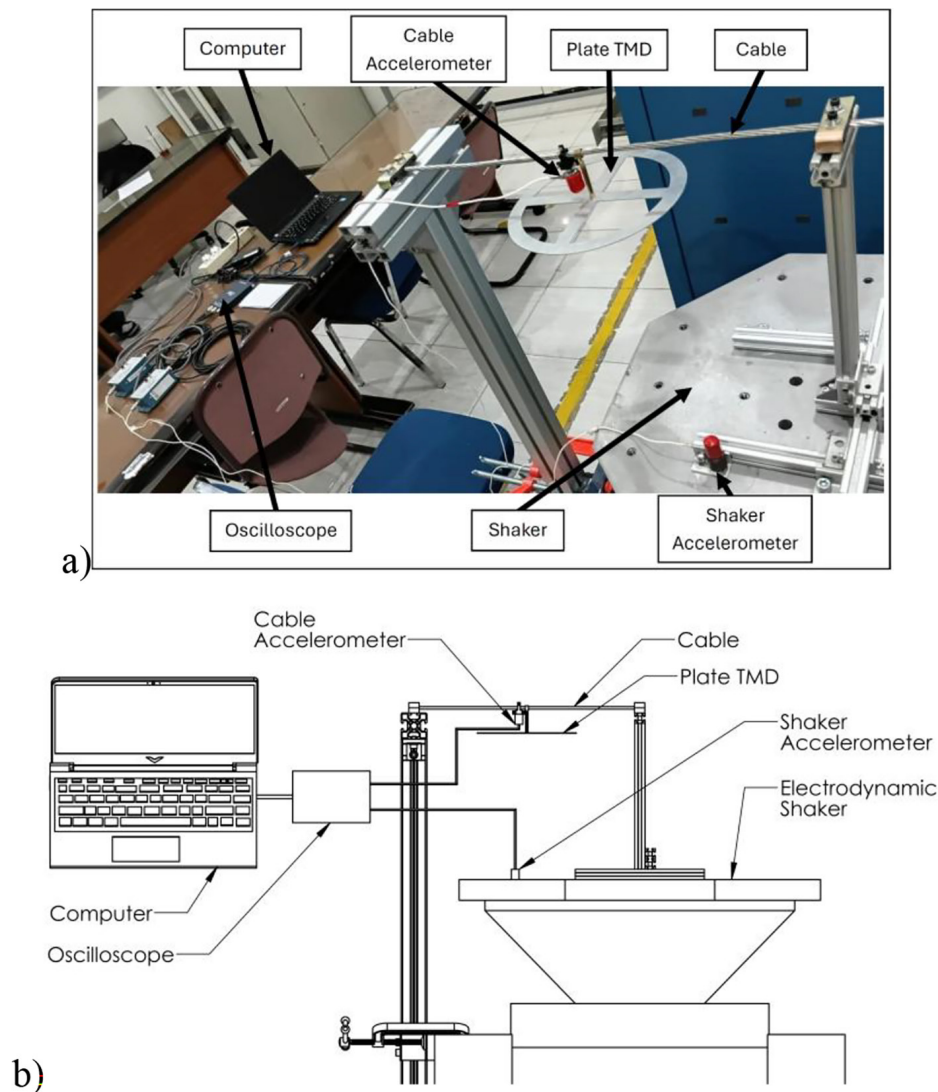


Figure 4. Photograph a) and schematic diagram b) of the experimental setup

Acceleration response amplitudes were obtained from the simulation and experimental acceleration data, and their ratios to the excitation acceleration amplitude were calculated. Four conditions were analyzed: simulation without Plate TMD, simulation with Plate TMD, experiment without Plate TMD, and experiment with Plate TMD. The amplitude ratios in the system condition, with and without the Plate TMD, are compared and analyzed to determine whether the Plate TMD can attenuate vibration.

RESULTS AND DISCUSSION

Parametric design study

From the harmonic response analysis simulation and experiment of the undamped beam-like cable model, the resulting natural frequency of 25 Hz serves as the target frequency for tuning the damper to reduce cable model vibration

effectively. The model is in single-loop vibration mode, as illustrated in Figure 5. This frequency forms the basis for evaluating the tuned mass damper's performance in mitigating resonant vibrations within the cable structure.

The first parametric design study of the plate for the tuned mass damper aims to attenuate vibration at the target cable frequency ($f_{n-target}$) of 25 Hz, with the maximum outer diameter ($D=D_{max}$) constrained to 0.2 m. In this phase, the plate is simulated without any holes, starting with an initial outer diameter ($i=1$; $D_i=D_1$) of 0.1 m. The design iteratively increases the outer diameter by 0.01 m increments ($i=i+1$) as long as the plate's natural frequency (f_n) remains higher than the target frequency ($f_{n-target}$). The iteration continues until the natural frequency at the final diameter iteration ($i=a$; $f_n(D_i=D_a)$) falls below the target frequency. The study results are presented in Figure 6.

In the plate model without holes, increasing the outer diameter decreases the natural frequency nonlinearly. This is because increasing

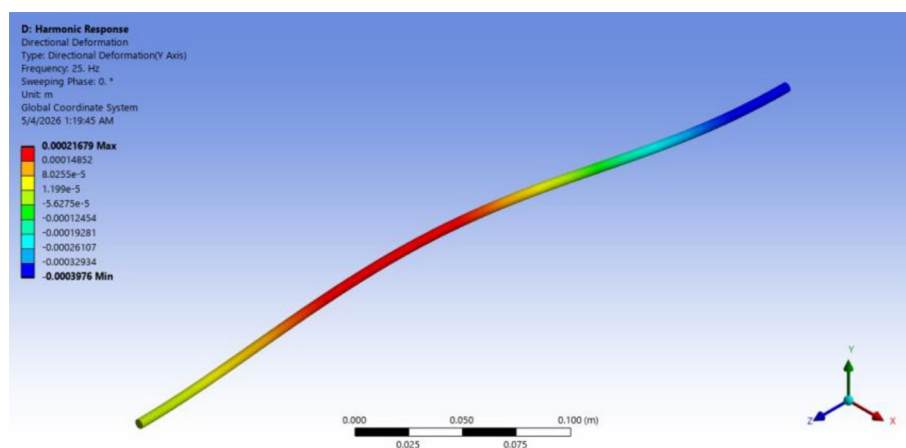


Figure 5. The undamped cable model in single loop mode

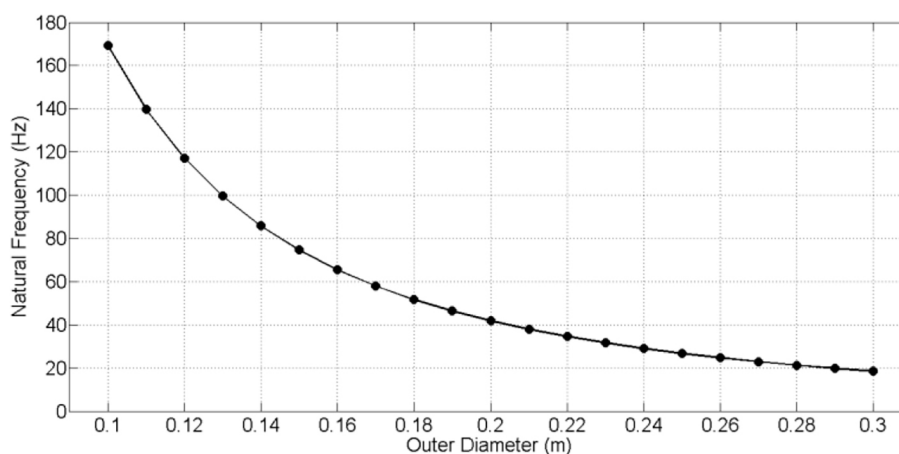


Figure 6. Plate's outer diameter vs natural frequency

the outer diameter at a small initial diameter will increase the plate mass less than at a large initial diameter. The small initial diameter condition has a smaller outer circumference than the large initial diameter condition. Furthermore, the smaller the outer diameter of the plate, the greater the plate stiffness.

The second parametric design study of the plate begins if the last-iteration outer diameter ($i=a$; $D_i=D_a$) from the previous phase is less than the maximum allowable outer diameter ($D_{\max}$) of 0.2 m. In this phase, the final outer diameter from the first study ($D_i=D_a$) is set as the initial outer diameter value ($j=1$; $D_j=D_1$). The study then proceeds by decrementally reducing the outer diameter by 0.001 m at each iteration ($j=j+1$). The iteration continues until the plate's natural frequency is closest to the target frequency ($f_{n\text{-target}}$), at which point the outer diameter is selected, and the study phase ends. In this case, a natural frequency of 25 Hz is obtained when the plate has an outer diameter between 0.25 and 0.26 m. The simulation results are shown in Equation 6, Equation 7, and Equation 8. Therefore, the study shifted to a plate design with holes.

$$f_n(D_{a-1} = D_{15} = 0.25 \text{ m}) = 26.6936 \text{ Hz} \quad (6)$$

$$f_n(D_a = D_{16} = 0.26 \text{ m}) = 24.6713 \text{ Hz} \quad (7)$$

$$f_n(D_{a-1} = D_{15} = 0.25 \text{ m}) < 25 \text{ Hz} < f_n(D_a = D_{16} = 0.26 \text{ m}) \quad (8)$$

where: $D_a > 0.2 \text{ m}$.

In the other scenario of the second parametric design study, where the last iteration's outer diameter ($i=a$; $D_i=D_a$) exceeds the maximum allowable outer diameter ($D_{\max}$), two design parameters are

treated as independent variables: the plate's outer diameter (D) and the spoke width (w_{spoke}). The study constrains the outer diameter to a minimum value ($D_{\min}$) of 0.1 m, a maximum value ($D_{\max}$) of 0.2 m, and increments (D_{int}) of 0.01 m. Simultaneously, the spoke width varies between a minimum ($w_{\text{spoke-min}}$) of 0.015 m and a maximum ($w_{\text{spoke-max}}$) of 0.025 m, adjusted in intervals ($w_{\text{spoke-int}}$) of 0.001 m. Surface graphs are generated to visualize the relationships between the dependent variables (natural frequency and mass) and the independent variables, as illustrated in Figure 7 and Figure 8. Figure 9 shows the relationship between outer diameter and spoke width to effective modal mass. Figure 10 shows the relationship between outer diameter and spoke width to equivalent stiffness considering the maximum deformation due to a plate circumference load of 10 N.

The surface graph interpretation of the parametric design study shows that increasing the spoke width has a more pronounced effect on raising the plate's natural frequency at smaller outer diameters compared to larger ones. As the plate's outer diameter increases, its natural frequency decreases. Conversely, at greater outer diameters, increasing the spoke width significantly contributes to an increase in the plate's mass more than it does at smaller diameters. Additionally, the plate's mass increases with increasing outer diameter. With the same plate material and thickness, the distributed mass value per unit area is 2.6738 kg/m². The next step of the study involves establishing the combination of outer diameter and spoke width that yields a natural frequency in accordance with the target frequency, achieving equilibrium between both natural frequency and mass.

The equivalent model of the plate vibration system can be seen from the trend of the influence

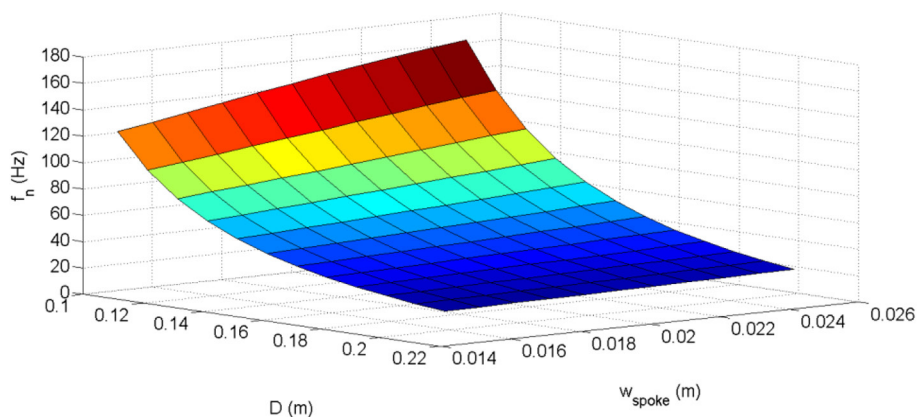


Figure 7. Surface graph of plate's outer diameter and spoke width vs natural frequency

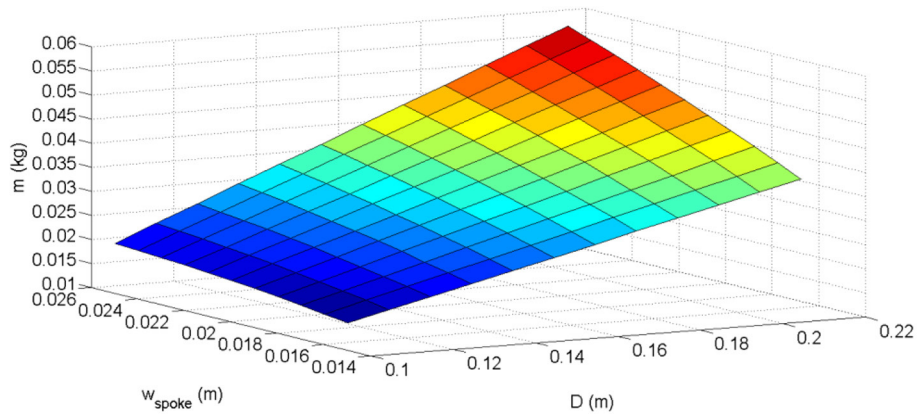


Figure 8. Surface graph of plate's outer diameter and spoke width vs total distributed mass

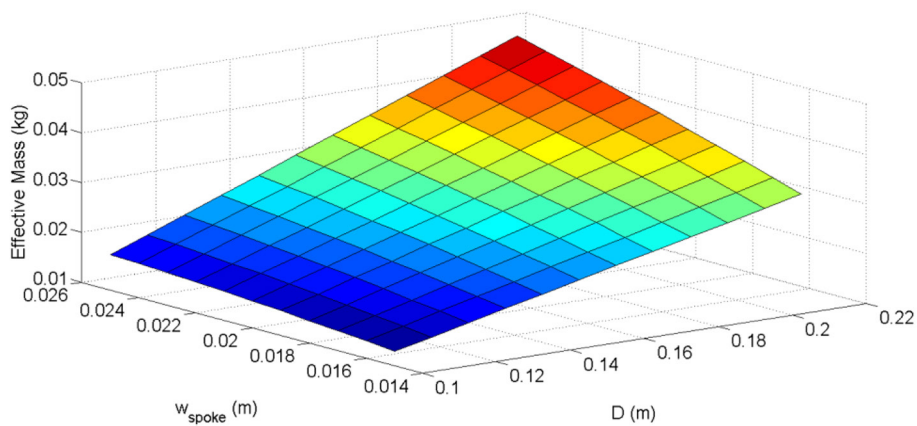


Figure 9. Outer diameter and spoke width vs effective mass

of changes in the outer diameter and spoke width on the effective mass and equivalent stiffness. Effective mass shows the amount of mass that contributes to movement during the 3-plate vibration mode condition which is a review of the vibration mode that affects the reduction of vibration at the target natural frequency (Figure 9). An increase in the outer diameter and spoke width values indicates an increase in the effective modal mass because an increase in the outer diameter and spoke width values causes an increase in the total distributed mass. Meanwhile, equivalent stiffness considers the application of a load on the circumference of the plate of 10 N and the maximum value of the resulting deformation (Figure 10). An increase in the outer diameter value causes a decrease in equivalent stiffness due to an increase in mass and a decrease in stiffness from the lengthening of the spoke with a fixed width. Meanwhile, an increase in spoke width causes an increase in equivalent stiffness. This shows that the effect of increasing stiffness due to increasing the spoke width is more significant than the

effect of increasing mass, resulting in an increase in natural frequency.

In a plate model with the holes and the same spoke width, the increase in outer diameter to the decrease in natural frequency is not linear. The physical phenomenon is the same as the condition in the plate model without holes. This radially elongated spoke acts like a cantilever bar with the plate mass at the circumference similar to the mass of the cantilever tip. The increase in outer diameter in the condition of a small initial diameter will increase the plate mass at the circumference less than in the condition of a large initial diameter because the condition of a small initial diameter has a smaller outer circumference than the condition of a large initial diameter. In addition, the smaller the outer diameter of the plate causes the spoke to be shorter so that the plate stiffness is greater.

In plate models with the holes and the same outer diameter, increasing spoke width tends to increase natural frequency linearly. Increasing spoke width not only increases stiffness but also

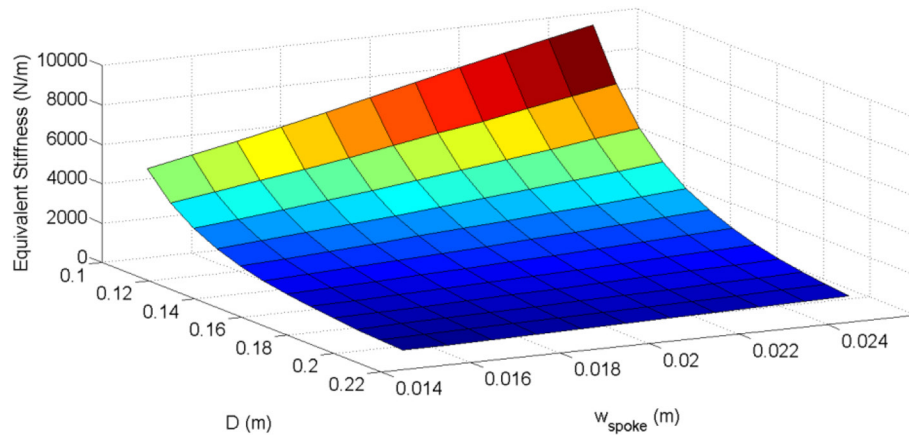


Figure 10. Outer diameter and spoke width vs equivalent stiffness

increases plate mass. Changes in natural frequency are more significant at smaller diameters. Increasing spoke width, which causes an increase in natural frequency, indicates that the increase in stiffness is more significant than increasing plate mass.

The combination of outer diameter and spoke width that yields a natural frequency equal to the target frequency of 25 Hz varies within specific ranges, as illustrated in Figure 11. At the minimum spoke width of 0.015 m, the outer diameter required to achieve the target natural frequency

lies between 0.19 m and 0.2 m. Conversely, at the spoke width of 0.025 m, the outer diameter shifts higher, falling between 0.2 m and 0.21 m. The mass with the smaller outer diameter and minimum spoke width is smaller than that with the smaller outer diameter and maximum spoke width, so that the minimum spoke width condition is chosen.

The iteration of the parametric design study for obtaining the outer diameter with the minimum spoke width focuses on the range between

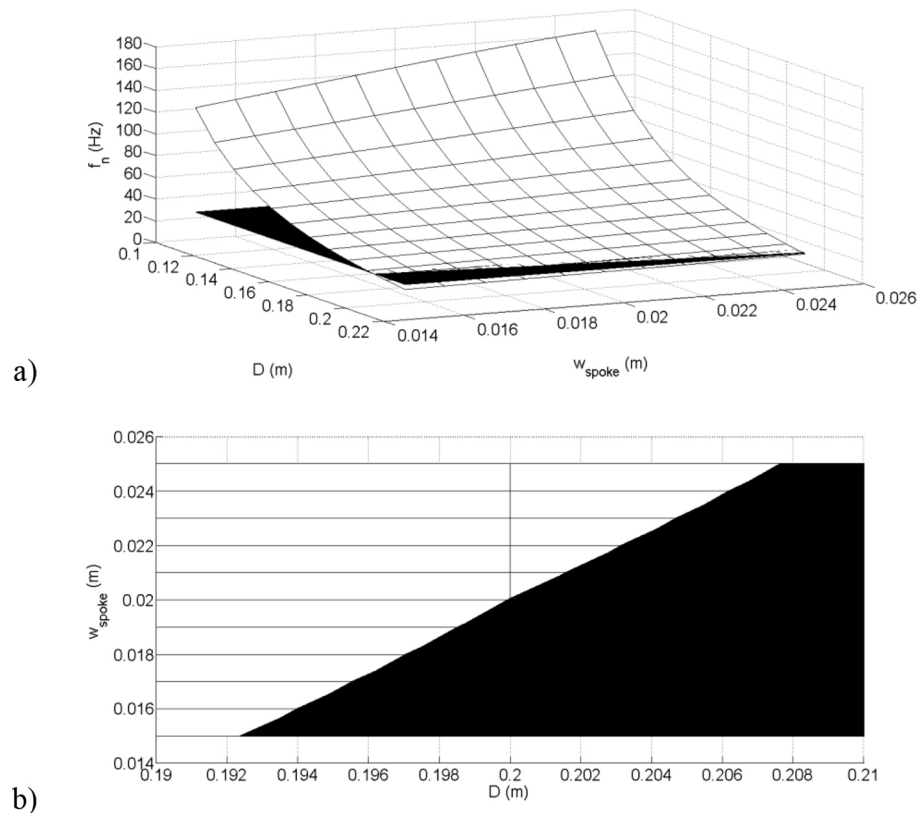


Figure 11. Surface graph a) of plate's outer diameter and spoke width vs natural frequency (white) that is crossed with the target natural frequency 25 Hz surface graph (black), and its top view b)

0.19 m and 0.2 m, as illustrated in Figure 12 and in Equation 9. During this phase, the outer diameter is varied incrementally within this interval to identify the plate dimension that yields natural frequencies closest to the target frequency of 25 Hz.

$$D, w_{spoke} \left\{ \min(|f_n - f_{n-target}|) \right\} = \min(m) \quad (9)$$

$$= D\{\min(|f_n - f_{n-target}|)\}, 0.015 \text{ m}$$

where: $0.19 \text{ m} < D < 0.2 \text{ m}$.

The graph interpretation shows two natural frequencies near the target frequency of 25 Hz, corresponding to outer diameters of 0.192 m and 0.193 m. The increase in outer diameter and the decrease in natural frequency appear linear. This does not contradict the physical phenomena described previously. This occurs because the range of changes in outer diameter is small and varies under conditions of large outer diameter plates in general. Consistent with the previous explanation, increasing outer diameter does not significantly decrease the natural frequency. The natural frequency at 0.192 m is 25.0711 Hz (Equation 10), resulting in a smaller frequency gap of 0.0711 Hz (Equation 11) from the target. At 0.193 m, the natural frequency is 24.7747 Hz (Equation 12), with a frequency gap of 0.2253 Hz (Equation 13). Given the closer proximity to the target frequency, the outer diameter of 0.192 m is selected as the plate dimension. The mass is 0.0345 kg (Equation 14). The selection of plate dimensions is stated in Equation 15. Figure 13 shows the three initial mode shapes of the plate. The third mode shape is selected according to the typical vibration mode that has been previously stated. Figure 14 shows the final detailed drawing and the fabrication result.

$$f_n(D = 0.192 \text{ m}, w_{spoke} = 0.015 \text{ m}) = 25.0711 \text{ Hz} \quad (10)$$

$$|f_n(D = 0.192 \text{ m}, w_{spoke} = 0.015 \text{ m}) - f_{n-target}| = |25.0711 - 25| \text{ Hz} = 0.0711 \text{ Hz} \quad (11)$$

$$f_n(D = 0.193 \text{ m}, w_{spoke} = 0.015 \text{ m}) = 24.7747 \text{ Hz} \quad (12)$$

$$|f_n(D = 0.193 \text{ m}, w_{spoke} = 0.015 \text{ m}) - f_{n-target}| = |24.7747 - 25| \text{ Hz} = 0.2253 \text{ Hz} \quad (13)$$

$$m(D = 0.192 \text{ m}, w_{spoke} = 0.015 \text{ m}) = 0.0345 \text{ kg} \quad (14)$$

$$D \left\{ \min \left[\begin{matrix} 0.0711 \text{ Hz} \\ 0.2253 \text{ Hz} \end{matrix} \right] \right\}, w_{spoke-minimum} = D\{0.0711 \text{ Hz}\}, 0.015 \text{ m} = 0.192 \text{ m}, 0.015 \text{ m} \quad (15)$$

Performance study

The results from both the FEM simulation and the experimental study demonstrate the effectiveness of the Plate TMD in reducing cable vibrations. In both simulation and experimental results, the amplitude ratio at the target frequency of 25 Hz is higher in the condition without TMD compared to the condition with TMD. This reduction in acceleration amplitude ratio confirms that the Plate TMD effectively attenuates vibration in the cable model. This is shown in Figure 15 and Table 5.

The simulation and experimental results demonstrate that the application of the Plate TMD effectively reduces the vibration response compared to the system without a TMD, as shown in

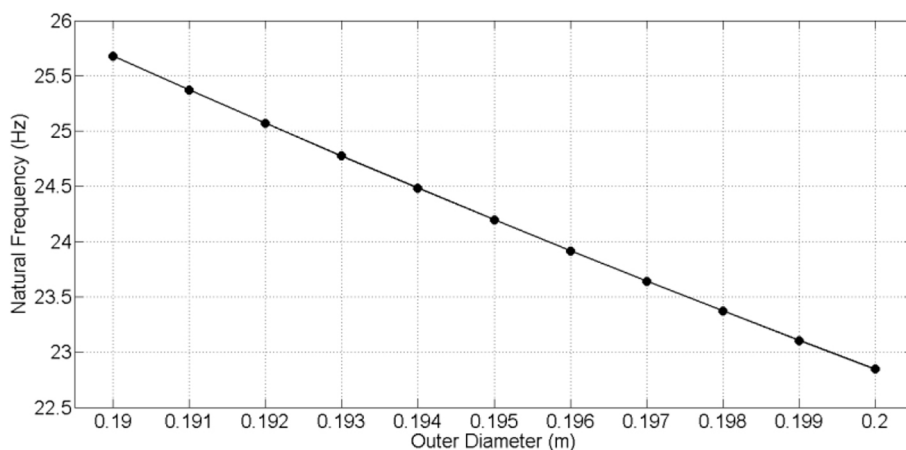


Figure 12. Plate's outer diameter vs natural frequency with minimum spoke width

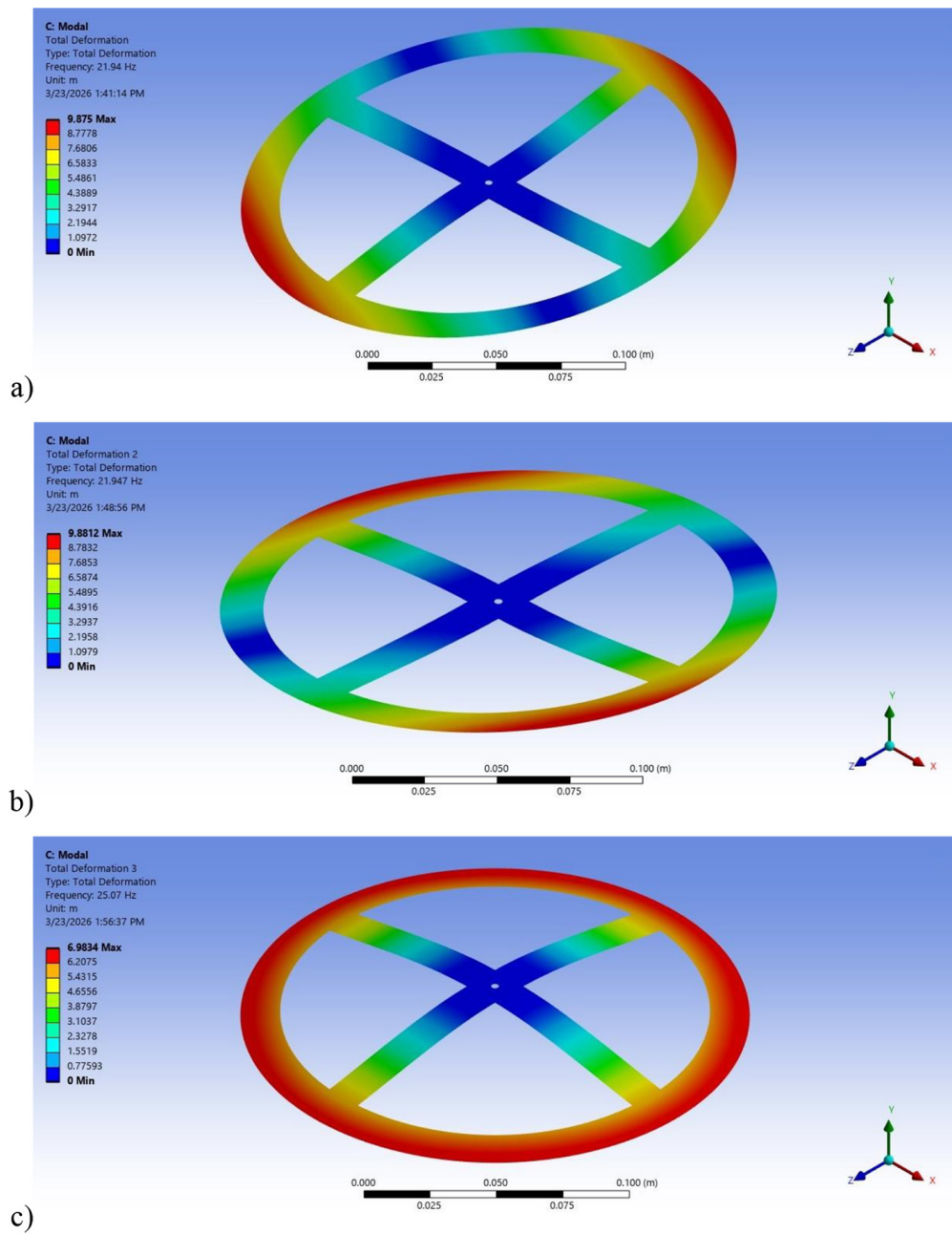


Figure 13. 1st a), 2nd b), and 3rd c) mode shapes of Plate TMD

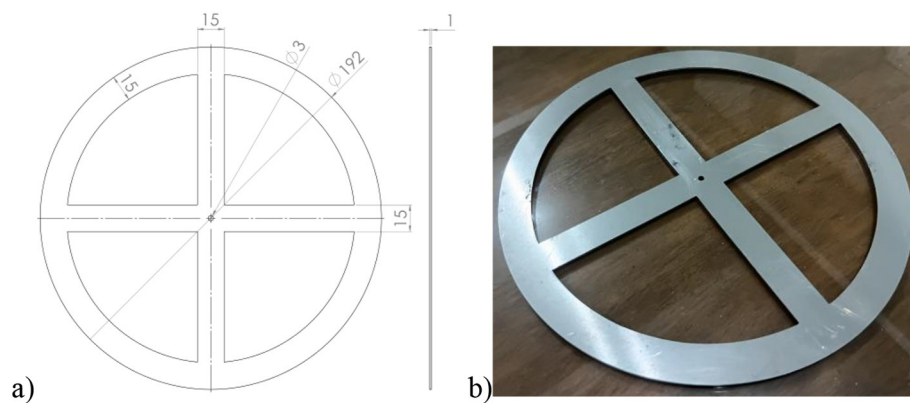


Figure 14. Final design drawing a) and fabrication result b) of the plate

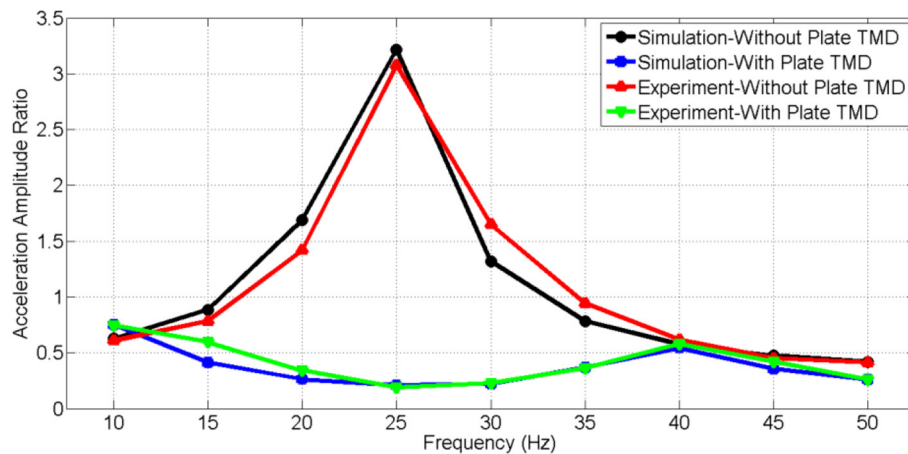


Figure 15. Simulation and experimental results for conditions without and with Plate TMD

Table 5. Comparison of simulation and experimental results at the target frequency

Conditions	Amplitude ratio at 25 Hz		
	Simulation	Experiment	Error (%)
Without plate TMD	3.2131	3.0757	4.4683
With plate TMD	0.2076	0.1888	9.9554
Reduction (%)	93.54	93.86	

Figure 15 and Table 5. Both results consistently show lower acceleration amplitudes when the Plate TMD is employed, confirming its capability to attenuate vibrations at the targeted natural frequency. These findings align with typical

outcomes observed in TMD applications on primary vibration systems.

The differences and discrepancies between simulation and experiment values occur for several reasons. This can be caused by the simplification of

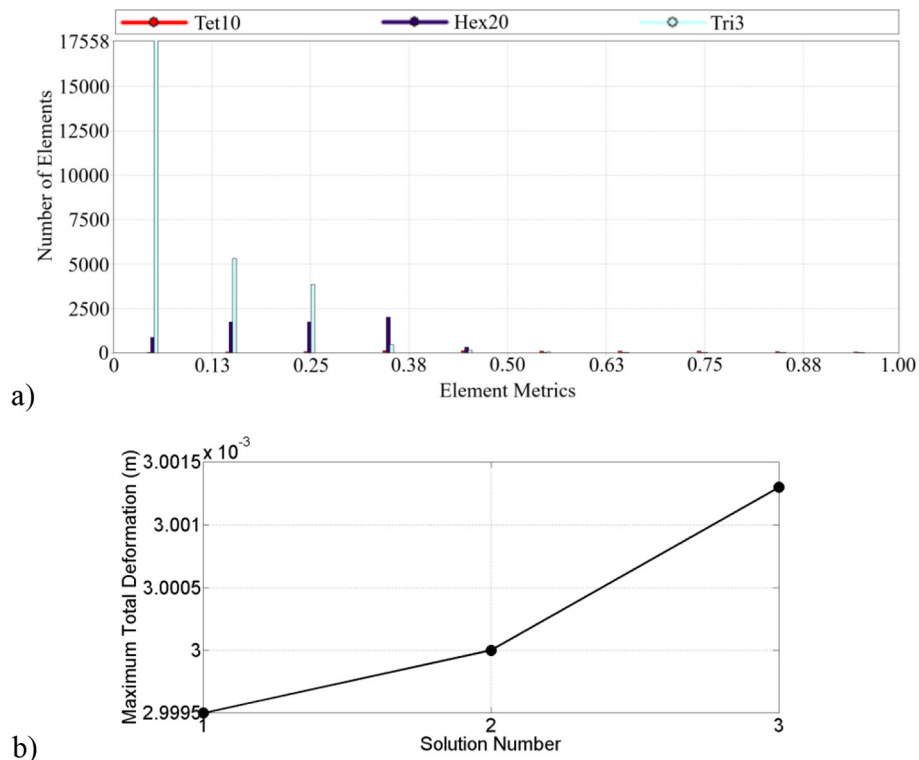


Figure 16. Mesh quality a) and convergence b) study

the simulation model compared to the actual cable shape, imperfections in the Plate TMD manufacturing, and inaccuracies in obtaining the material properties. However, the graphical trends between the simulation and experiment results are similar. Similarly, the natural frequency values, along with the peak amplitude ratio values, between the simulation and experiment results are similar. This indicates that the simulation is relevant and valid compared to the experiment.

Mesh quality can be seen from the mesh metric graph of skewness in Figure 16a. The mesh metric value of skewness has an average of 0.13016, which means the average mesh quality is excellent. The mesh convergence study can be seen in Figure 16b and Table 6. This shows the results of the simulation converged with the mesh conditions used. The largest percentage change was 0.003%, below 10%.

The harmonic response simulation results show deformation in the plate TMD model at the target frequency, as shown in Figure 17. This indicates the plate dynamics in reducing the vibration of the cable model at that frequency. The deformation pattern on the plate tends to move uniformly in the plate circumference mass. This deformation pattern corresponds to the plate vibration mode considered as a reference for the Plate TMD's natural frequency.

This Plate TMD model demonstrates the ability to reduce vibration at the target frequency. This

model has a light mass with the mass selection in the previous parametric design study. In addition, the light mass is also obtained by eliminating additional masses that are usually applied to TMD models. The plate is considered as an entity that has stiffness and also acts as a mass distributed throughout the plate body. This is very different from other TMD models, where the addition of mass means acting as a lumped mass while the stiffness element is separate. Therefore, this Plate TMD model is simpler than other TMD models.

The comparison of damper performance applied to the cable model shows that the Plate TMD achieves a greater reduction compared to other damper types, as shown in Table 7. The application between damper models uses the same boundary conditions with fixed-fixed ends, and harmonic excitation. Traditional TMD and TMIS performances were the ratio of the response amplitude to the static deformation. The asymmetrical Stockbridge damper performance was defined as the maximum value of the peak-to-peak amplitude ratio. Traditional TMD, consisting of a simple mass, spring, and damper system, and the asymmetrical Stockbridge damper, which uses different mass sizes and messenger lengths, shows relatively low reduction. Their parameters are not being specifically adjusted to the cable's dynamic characteristics. The tuned mass inerter system (TMIS) performs better in cables with multiple sections but is less

Table 6. Mesh convergence study

Solution number	Element size (m)	Nodes	Elements	Maximum total deformation (m)	Change (%)
1	0.0015	47594	34380	2.9995e-03	
2	0.0010	75448	40028	3.0000e-03	2.9995e-03
3	0.0005	357701	102648	3.0013e-03	3.0000e-03

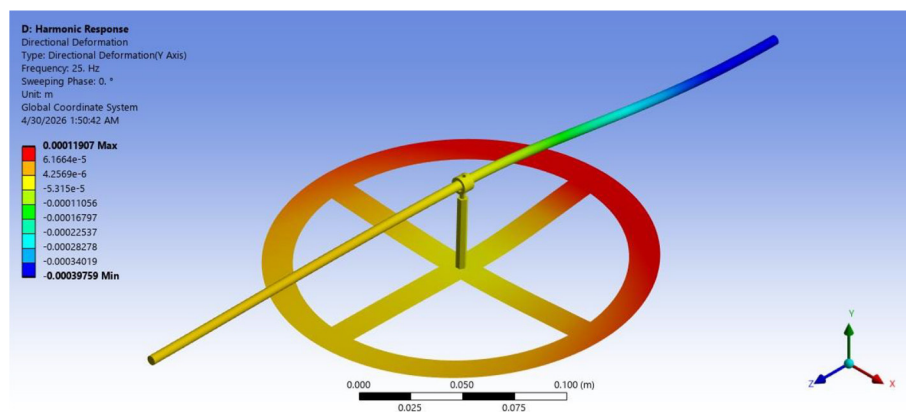


Figure 17. Harmonic response of the damped cable at 25 Hz

Table 7. Evaluation of performance relative to the other damper models

Damper models	Reduction (%)
Traditional TMD [24]	44.77
Tuned Mass Inerter System (TMIS) [24]	49.25
Asymmetrical Stockbridge damper [46]	42.73
Plate TMD	93.86

effective for single-section cables. Distinctly, other damper models necessitate supplementary mass for effective vibration mitigation, in contrast to the Plate TMD, which functions devoid of additional mass, thereby making it lighter, simpler in production, and easier to install, while providing practical benefits along with its vibration-damping capabilities.

Further investigation concerning the Plate TMD ought to prioritize the enhancement of its design through optimization methodologies to improve its efficacy in vibration attenuation. Although the present study illustrates the feasibility and efficiency of the Plate TMD in mitigating cable vibrations, there exist further studies to refine its parameters to achieve improved efficiency. Moreover, the exploration of the Plate TMD's application in alternative primary vibration systems beyond cable infrastructures could expand its practical relevance and versatility. Such advancements would yield a more adaptable and optimized vibration-control mechanism, thereby facilitating a wider range of structural applications. In addition, further studies can be carried out on the main vibration system with real or industrial scale for experiments, such as on the type of cable vibration system, including various sizes and types.

CONCLUSIONS

The model of the circular plate integrates the dual principles of plate stiffness and the absence of additional mass from previous TMD studies, and makes the TMD lighter, simpler to manufacture, and easier to install. However, the plate can be considered to have stiffness and as a mass distributed throughout its body. The Plate TMD in this research is a new TMD model that employs a model-simplification concept and a natural frequency adjustment method. The model uses no additional mass and uses additional holes for the natural frequency adjustment.

The circular plate is used in this TMD concept, with a parametric design study of the outer diameter and hole dimensions. The simple circular plate model without a hole is the prioritized model for the first phase of the study, subject to an outer dimension limitation. In the case of this study, the outer diameter without holes is more than 0.2 m or exceeds the limit with a target frequency of 25 Hz, A11100 plate material, and a thickness of 0.001 m. Adding holes is for the next phase of the study if the target natural frequency is not achieved. Increasing the outer diameter decreases the natural frequency non-linearly. Increasing the outer diameter at a small initial diameter will increase the plate mass less than at a large initial diameter. The smaller the outer diameter of the plate, the greater the plate stiffness. Increasing spoke width tends to increase natural frequency linearly. Increasing spoke width, which causes an increase in natural frequency, indicates that the increase in stiffness is more significant than increasing plate mass. The study has shown that the target natural frequency of the plate can be obtained with a lighter mass in the smallest spoke width. In the case of this study, the outer diameter was chosen as 0.192 m and the spoke width was 0.015 m with the frequency gap of 0.0711 Hz, and the mass of 0.0345 kg.

The design result of the plate TMD model is used for attenuating the vibration of the beam-like cable model. The vibration studies use both FEM simulation and experimental methods. The study results show that the plate TMD model is feasible and capable of attenuating the vibration at the target natural frequency. In this study, the response amplitude ratio decreased by more than 93 percent. The vibration suppression is aligned with typical results in TMD applications.

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