Bursa Uludağ University Journal of The Faculty of Engineering, Vol. 26, No. 3, 2021 RESEARCH

DOI: 10.17482/uumfd.878114

OPTIMIZATION OF NON-TRADITIONAL TUNED MASS DAMPER FOR DAMPED STRUCTURES UNDER HARMONIC EXCITATION

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Received: 10.02.2021; revised: 17.09.2021; accepted: 23.09.2021

Abstract: Tuned mass dampers (TMDs) are used to reduce dynamic vibrations of structures under environmental loads such as wind or seismic excitation. In this paper, the optimum design of non-traditional tuned mass dampers (NT-TMD) attached to a damped main structure under harmonic excitation was investigated. Unlike the traditional TMD, the damping element in NT-TMD is directly connected to the ground. In this study, the effectiveness of NT-TMD on the attenuation of vibrations on the damped main system under harmonic load is investigated. The optimum parameters of the NT-TMD are obtained by using the hybrid pattern search (HPS) technique. According to numerical results, it is seen that non-traditional TMD is more effective than traditional TMD in reducing vibration.

Keywords: Harmonic excitation, Optimum design, Performance, Tuned mass damper

Harmonik Etki Altındaki Sönümlü Yapılar için Geleneksel Olmayan Ayarlı Kütle Sönümleyicinin Optimizasyonu

Öz: Ayarlı kütle sönümleyiciler (AKS) rüzgâr ve sismik etkiler gibi çevresel yükler altındaki yapıların titreşimlerinin azaltılmasında kullanılmaktadır. Bu yazıda, harmonik etki altındaki sönümlü bir ana yapıya eklenen geleneksel olmayan ayarlanmış kütle sönümleyicilerin optimum tasarımı araştırılmıştır. Geleneksel AKS'den farklı olarak, sönüm elamanı direk olarak yere bağlanmaktadır. Geleneksel olmayan AKS'nin optimum parametreleri hibrit model arama tekniği kullanılarak elde edilmiştir. Numerik sonuçlar titreşim azaltılmasında geleneksel olmayana AKS'nin geleneksel AKS'ye göre daha etkili olduğu görülmektedir.

Anahtar Kelimeler: Harmonik etki, Optimum tasarım, Performans, Ayarlı kütle sönümleyici

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1. INTRODUCTION

Recently, various control systems have been used frequently to reduce vibrations on real structures (Esen and Koc, 2015; Koc, 2020 a,b; Araz and Kahya, 2021). Traditional and non-traditional TMDs have an easy-to-maintain and low cost unlike other types of system. Thus, there is a potential for TMDs to be used effectively in reducing different dynamic responses on the real structures.

The first TMD invented by Frahm (1909) without damping element had an effect in a narrow frequency range. Later, Ormondroyd and Den Hartog (1928) studied the classical TMD which has a mass system, a damping element and a spring element. Then, Den Hartog (1956) derived the optimum tuning parameters for TMD using the fixed-points theory. After these pioneering works, different methods for the optimum design of the TMD have been presented to minimize the dynamic vibrations of structures under different external excitations (Warburton, 1982; Fujino and Abe, 1993; Tsai, 1995; Jangid, 1999; Asami et al., 2002; Li, 2002; Bakre and Jangid, 2004; Marano et al., 2007; Marano and Greco, 2011; Nigdeli and Bekdas, 2014; Matta, 2015; Yazdi et al., 2016; Mate et al., 2017; Kahya and Araz, 2017; Dell'Elce et al., 2018; Mrabet et al., 2018; Araz and Kahya, 2018; Yucel et al., 2019; Kahya and Araz, 2019; Araz, 2020; Araz and Kahya, 2020; Ruge and Wagner, 2020).

NT-TMD devices are firstly studied by Ren (2001) and Liu and Liu (2005). In these studies, the dynamic behavior of the main structure subjected to harmonic load effect is reduced by using NT-TMD. Wong and Cheung (2008) investigated the control performance of a NT-TMD attached to main system under ground motion. The effectiveness of a NT-TMD in reducing the peak vibration velocity of the structure due to harmonic load is investigated by Cheung and Wong (2009). They also presented mathematical formulation for obtaining optimal parameters of a NT-TMD to reduce the resonant vibration amplitude and the total vibration energy of the structure system (Cheung and Wong, 2011a, b). Yuan et al. (2018) developed an experimental system to obtain the control performance of the NT-TMD.

When damping is existing at the main structure, there are too little research on the non-traditional tuned mass dampers. Liu and Coppola (2010) obtained the performance of NT-TMD by using sequential simplex method and Chebyshev's equi-oscillation theorem. Anh and Nguyen (2014) investigated the effectiveness of the NT-TMD by using approximate analytical solution for damped structure under external excitation. Anh et al. (2016) presented new method for obtaining the tuning parameters of a NT-TMD.

Unlike previous studies, the optimum parameters of NT-TMD are obtained for a damped main system instead of an undamped main system subjected to harmonic excitation. The tuning parameters of a NT-TMD are obtained by using the hybrid pattern search (HPS) technique which depends on the pattern search (PS) and genetic algorithm (GA). Although PS is widely used in the solution of different engineering problems, it is rarely used in the optimization of the tuned mass dampers. Therefore, it is preferred in the solution of the optimization problem in this study. However, no previous works have investigated the relationship between the robustness and the design parameters of a NT-TMD and TMD systems. The above state of the art leads to the motivation for further investigating the NT-TMD in this paper. In this respect, NT-TMD has been applied to a damped single degree of freedom (sdof) system due to harmonic excitation. Then, we compared the results obtained by HPS technique with those obtained by other methods and evaluated the control performance of these two available devices (i.e. NT-TMD and TMD). Therefore, this work is organized as follows. section 2 provides a detailed formulation of the problems being addressed, and section 3 discusses the optimization problem and optimal parameters in detail. Then, section 4 provides some numerical examples, followed by discussions and conclusions in section 5.

2. MATHEMATICAL MODELING

2.1. Traditional TMD

A sdof damped main system attached with a TMD or NT-TMD is presented in Fig. 1 (a). With an excitation force $f_0 \sin(\omega t)$ exerted on the main structure, the dynamics of the two dof system in Fig. 1 (a) can be described by the equations of motion:

$$m_s \ddot{x}_s + c_s \dot{x}_s + c_t(\dot{x}_s - \dot{x}_t) + k_s \dot{x}_s + k_t(\dot{x}_s - \dot{x}_t) = f_0 \sin(\omega t) \tag{1}$$

$$m_t \ddot{x}_t + c_t(\dot{x}_t - \dot{x}_s) + k_t(\dot{x}_t - \dot{x}_s) = 0 \tag{2}$$

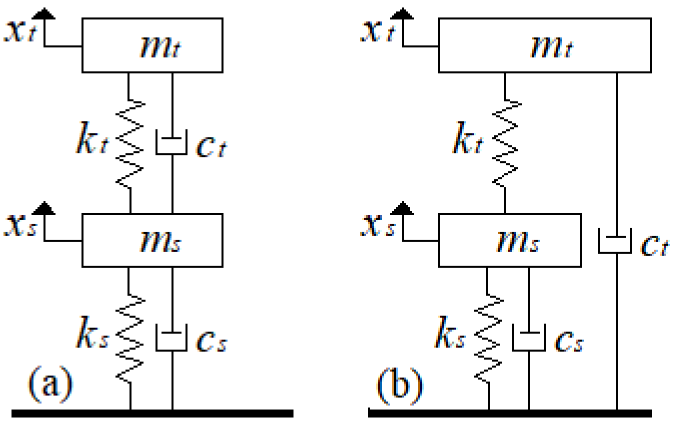

Figure 1:
TMD and NT-TMD attached to main structure, respectively

where the subscripts *t* and *s* are the TMD and main system, respectively. The mass, damping and stiffness coefficients are indicated by *m*, *c* and *k*, respectively. ω and f_0 are the frequency and amplitude of the external force respectively.

The dynamic magnification factor (DMF) for the displacement of the structure is given as follows (Asami et al., 2002)

$$DMF = \sqrt{\frac{(2\alpha\beta\xi_t)^2 + (\alpha^2 - \beta^2)^2}{a^2 + b^2}} \tag{3}$$

where

$$\begin{aligned} &\beta = \frac{\omega}{\omega_s}, \quad \alpha = \frac{\omega_t}{\omega_s}, \quad \xi_s = \frac{c_s}{2m_s\omega_s}, \quad \xi_t = \frac{c_t}{2m_t\omega_t}, \quad \mu = \frac{m_t}{m_s} \\ &a = [2\beta\xi_s(\alpha^2 - \beta^2) + 2\alpha\beta\xi_t(1 - \beta^2 - \mu\beta^2)] \\ &b = [\alpha^2 - (1 + 4\alpha\xi_s\xi_t + \alpha^2 + \mu\alpha^2)\beta^2 + \beta^4] \end{aligned} \tag{4}$$

where the tuning ratio, the mass ratio, the damping ratio and the natural frequency of the TMD are indicated by α, μ, ξ_t and ω_t, respectively. The natural frequency of the main system and the forced frequency ratio are indicated by ω_s and β, respectively.

For TMD device, the optimum tuning parameters for undamped main system are determined from Eq. (5) (Den Hartog, 1956)

$$\xi_t = \sqrt{\frac{3\mu}{8(1+\mu)}} \quad \alpha = \sqrt{\frac{1}{1+\mu}} \tag{5}$$

2.2 Non-Traditional TMD

A sdof damped structure attached with a NT-TMD is presented in Fig. 1 (b). With an excitation force $f_0 \sin(\omega t)$ exerted on the main structure, the dynamics of the two dof system in Fig. 1 (b) can be described by the equations of motion:

$$m_s \ddot{x}_s + c_s \dot{x}_s + k_s(\dot{x}_s + \dot{x}_t) - k_t \dot{x}_t = f_0 \sin(\omega t) \tag{6}$$

$$m_t \ddot{x}_t + c_t \dot{x}_t + k_t(\dot{x}_t - \dot{x}_s) = 0 \tag{7}$$

where the mass, damping and stiffness coefficients are indicated by m, c and k respectively. The subscripts t and s are the NT-TMD and main structure, respectively. ω and f_0 denote the frequency and amplitude of the external force, respectively.

DMF of the structure for the displacement is given as follows (Anh and Nguyen, 2014)

$$DMF = \sqrt{\frac{(2\alpha\beta\xi_t)^2 + (\alpha^2 - \beta^2)^2}{a^2 + b^2}} \tag{8}$$

where

$$\begin{aligned} &\beta = \frac{\omega}{\omega_s}, \quad \alpha = \frac{\omega_t}{\omega_s}, \quad \xi_s = \frac{c_s}{2m_s\omega_s}, \quad \xi_t = \frac{c_t}{2m_t\omega_t}, \quad \mu = \frac{m_t}{m_s} \\ &a = [2\beta\alpha\xi_t(1-\beta^2+\mu\alpha^2) + 2\xi_s\beta(\alpha^2-\beta^2)] \\ &b = [\alpha^2 - (1+\alpha^2+\mu\alpha^2+4\alpha\xi_s\xi_t)\beta^2 + \beta^4] \end{aligned} \tag{9}$$

The given parameters in Eq. (9) are previously defined in Eq. (4).

The optimum tuning ratio of a NT-TMD are given by in references (Liu and Coppola, 2010; Anh and Nguyen, 2014), respectively.

$$\alpha = \sqrt{\frac{1-4\xi_s^2}{1-\mu}} \tag{10}$$

$$\alpha = \frac{1}{\sqrt{1-\mu}\left(\sqrt{1+\frac{\pi^2}{(\pi^2-2)^2}\xi_s^2} + \frac{\pi}{\pi^2-2}\xi_s\right)} \tag{11}$$

Although they obtained the optimum values of the tuning ratio, used the formula $\xi_t = \left(3\mu/(8-4\mu)\right)^{0.5}$ suggested by Ren (2001) for the damping ratio of NT-TMD. The values obtained from this formula is not optimal as it is recommended for the undamped main structure.

3. OPTIMIZATION OF NT-TMD

General formulation of the optimization problem can be expresses as follow

$$\min DMF(p) \quad p^L \le p \le p^U \tag{12}$$

where $p = [\xi^{opt} \ \alpha^{opt}]$ is termed the design vector and $DMF(p)$ denotes the objective function. $p^U = [0.5 1.2]$ and $p^L = [0 0.8]$ are the upper and lower bound vectors of the design variables, respectively. These bounds are chosen based on the literature (Anh and Nguyen, 2014).

It was assumed that the mass ratio μ of the control devices is known. Therefore, two design variables (i.e., ξ^{opt} and α^{opt}) are needed for the optimization of both devices. The following ranges for these parameters for are given by

$$\alpha^{opt} = \frac{\omega_t}{\omega_s}, \quad \xi^{opt} = \frac{c_t}{2m_t\omega_t}, \quad \mu = \frac{m_t}{m_s} \tag{13}$$

where ξ^{opt} and α^{opt} denote the damping ratio and the optimum tuning ratio for considered control devices.

The optimization problem addressed in this study is solved by using the hybrid pattern search (HPS) technique which depends on the pattern search (PS) and genetic algorithm (GA). Although PS is widely used in the solution of different engineering problems (Wetter and Polak, 2005; Karakaya and Soykasap, 2009; Amiri et al., 2011; Baghari and Amini, 2013), it is rarely used in the optimization of the tuned mass dampers. Therefore, it is preferred in the solution of the optimization problem in this study.

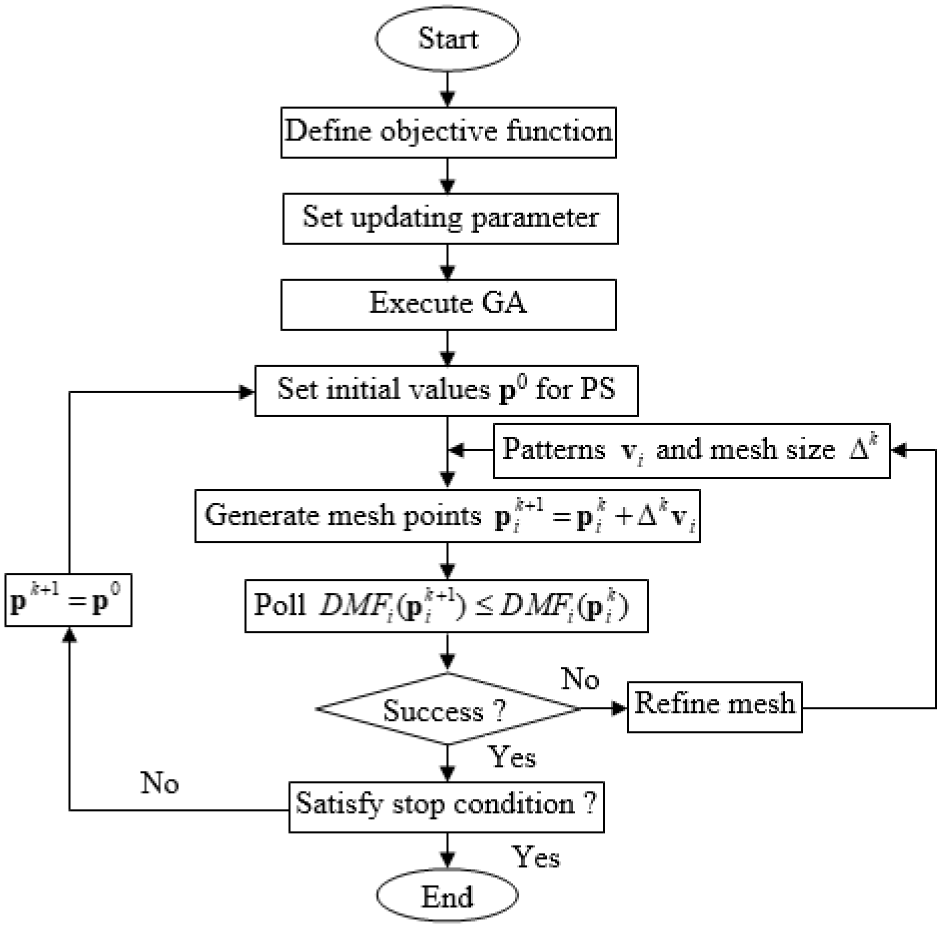

Figure 2:
Flowchart of present optimization method

The PS method consists of two steps, namely the pattern step and the poll step. The pattern consists of a set vectors $\mathbf{v}_i$ (i = 1,2 3 and 4) such as $\mathbf{v}_1$ = [1 0], $\mathbf{v}_2$ = [0 1], $\mathbf{v}_3$ = [-1 0] and $\mathbf{v}_4$ = [0 -1]. A mesh is then created around the search point with the help of the pattern vector. It is expressed as follows:

$$\mathbf{p}_i^{k+1} = \mathbf{p}_i^k + \Delta^k \mathbf{v}_\mathbf{i} \tag{14}$$

where Δ^k denotes the mesh size.

The PS polls the mesh points in the current mesh by calculating their objective function values. When the algorithm finds a point whose objective function value is smaller than that of the current point, it stops polling the mesh points. In this case, the poll is considered succeeded and the point found becomes the current point in the next step. Objective function values and mesh points are calculated by the algorithm up to the point at which it stops the poll.

In Fig. 3, the optimum tuning ratios obtained from the presented method are compared to the results of different optimization methods such as H_∞ optimization method and the fixed-point theory. According to these figures, it is observed that the results show great similarity. The present method gives slightly greater tuning ratios than those of Anh and Nguyen (2014).

Compared with the other two methods for $\xi_s = 2\%$ and 5%, the maximum difference for any tuning ratio value is seen to be 0.14% and 0.75%, respectively.

Fig. 4 shows the relation curves between the mass ratio and the optimum damping ratio. As can be seen, the present method gives greater damping ratios than those of the other method. Compared with the other both methods for $\xi_s = 2\%$ and 5%, the maximum difference for any optimum damping value is seen to be 7.84% and 11.97%, respectively. Since the damping ratios used for the other methods are not optimum, the magnitude of the difference is larger than the tuning ratio.

In the below figures, Ref. (1) and Ref. (2) indicates Liu and Coppola (2010) and Anh and Nguyen (2014), respectively. H_∞ optimization method and the fixed-point theory are used in these references.

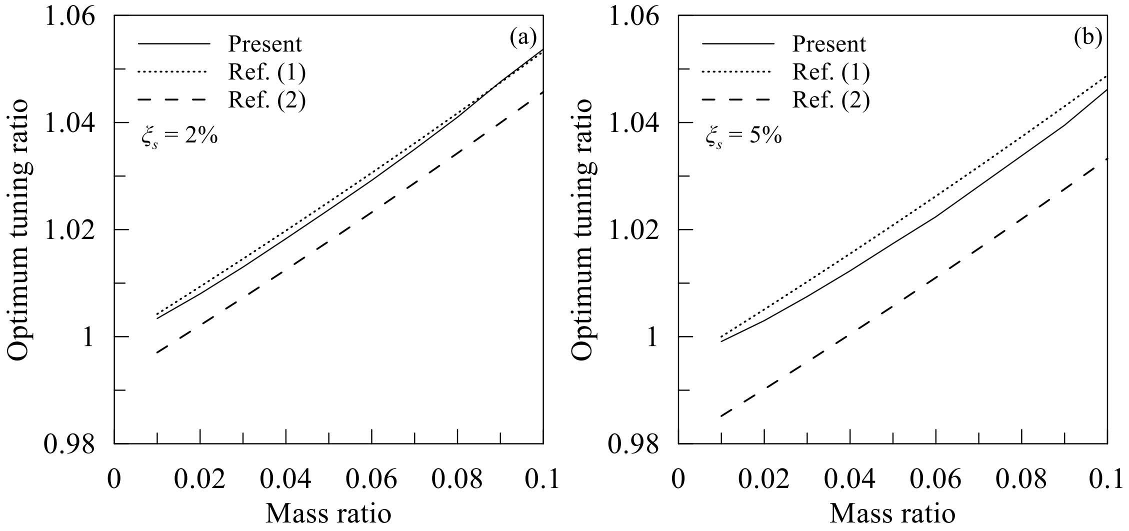

Figure 3:
The change of optimum tuning ratio with respect to mass ratio

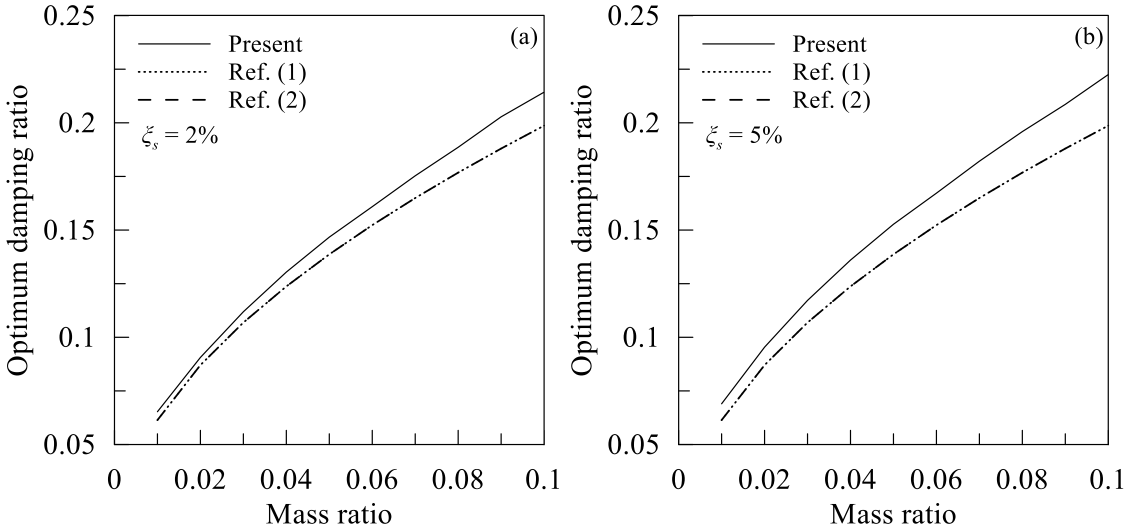

Figure 4:
The change of optimum damping ratio with respect to mass ratio

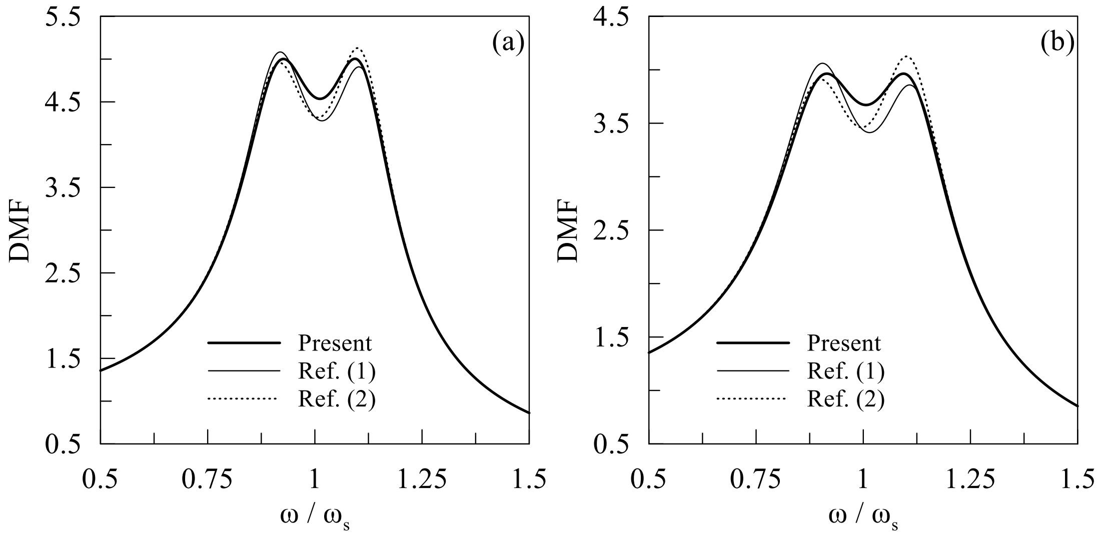

Figure 5:
Response amplitude DMF of the structure with the NT-TMD for $\mu = 0.05$:(a) $\xi_s = 2\%$; (b) $\xi_s = 5\%$

Table 1 Comparison of optimization results with the literature values where $\mu = 0.05$

Variables/ function	$\xi_s = 2\%$			$\xi_s = 5\%$		
	Present	Ref. (1)	Ref. (2)	Present	Ref. (1)	Ref. (2)
α_{opt}	1.0237	1.0252	1.0178	1.0174	1.0208	1.0057
ξ_{opt}	0.1467	0.1368	0.1368	0.1527	0.1368	0.1368
max. DMF	5.0040	5.0835	5.1293	3.9640	4.0610	4.1253

In Figs. 5(a) and (b), response amplitude DMF of the structure with the NT-TMD for the different damping ratios is shown. The peak responses and optimal parameters are also given in Table 1. As can see from Table 1 and Fig. 5, NT-TMD designed with the presented method has better control performance than other methods.

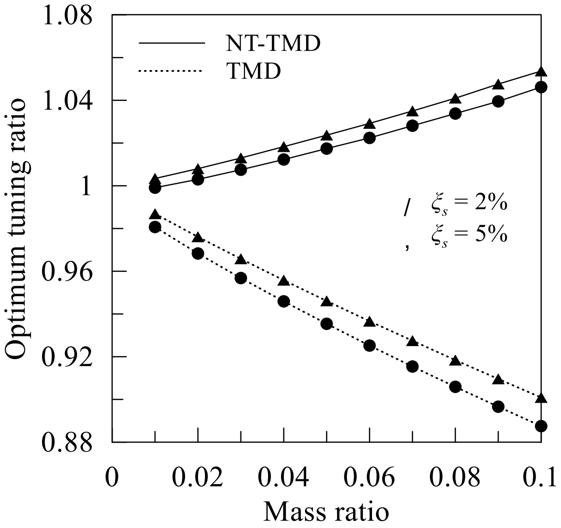

Figure 6:
The optimum tuning ratios obtained for different mass ratios

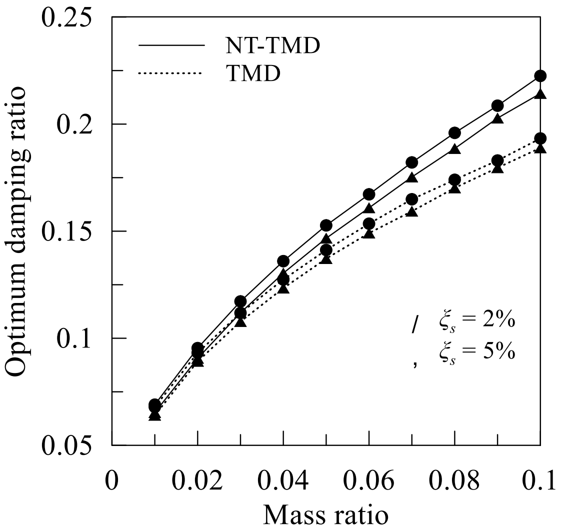

Figure 7:
The optimum damping ratios obtained for different mass ratios

The optimum tuning ratios with different mass ratios are given in Fig. 6. Fig. 6 depicts the optimum values of TMD are less than that of NT-TMD. Figure 6 also shows that the tuning ratio increases with the decrease in the damping ratio of the structure.

The optimum damping ratios of the both devices for various mass ratios are shown in Fig. 7. As can see from Fig. 7, the optimum damping ratio of the TMD is smaller than that of NT-TMD. Fig. 7 also depicts as the damping ratio of the main system and the mass ratio increase, the damping ratio of both devices increases.

4. NUMERICAL ANALYSIS

Numerical analysis is carried out to investigated control effectiveness of the NT-TMD in suppressing the peak DMF values of the structure for two different structural damping ratios (i.e., $\xi_s = 2\%$ and 5%).

Fig. 8 shows the displacement responses of the structure with the NT-TMD and TMD for $\mu = 0.05$. Fig. 8 indicates that the optimum NT-TMD is more effective than classical TMD in suppressing the vibrations around the resonance.

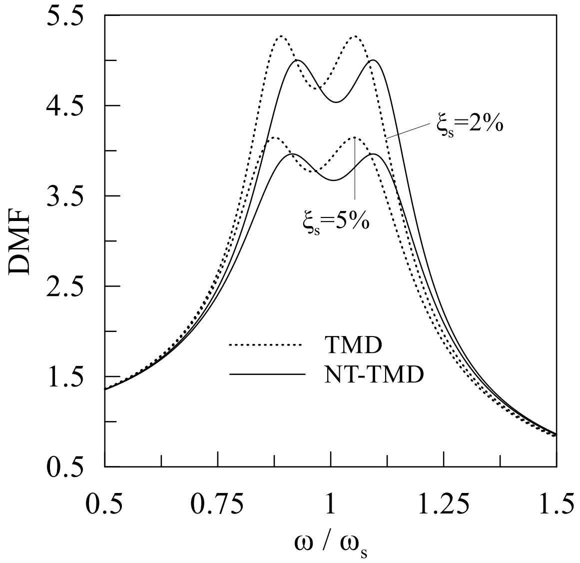

Figure 8:
Comparison of DMF response of the structure with the both devices

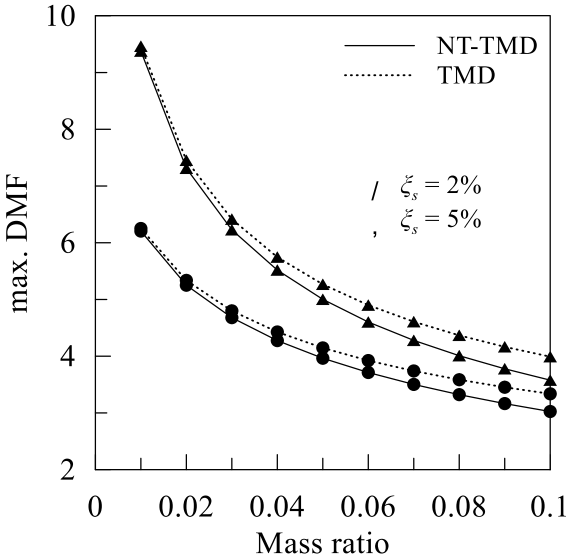

Figure 9:
The maximum DMF values obtained against different mass ratios

For further validation, the maximum DMF curves against the mass ratio for both devices are given in Fig. 9. As seen from this figure, if the mass ratio is bigger than 0.03, NT-TMD offers higher effectiveness than TMD. Thus, it is recommended that NT-TMD has a mass ratio greater than 0.03 for high control performance.

5. CONCLUSIONS

A number of numerical analyzes are carried out to investigate the effectiveness of non-traditional TMDs attached to a damped main structure due to harmonic excitation. Hybrid pattern search (HPS) technique is used to obtain optimum values. The conclusions can be summarized as follows.

- The optimum values obtained for NT-TMD are greater than those obtained for TMD.
- The increase in the mass ratio increases the tuning ratio of NT-TMD while decreases the tuning ratio of TMD.
- The increase in the mass ratio causes the damping ratios of both control devices to increase.
- When the mass ratio is bigger than 0.03, NT-TMD offers higher effectiveness than TMD. Thus, it is recommended that NT-TMD has a mass ratio greater than 0.03 for high control effectiveness.

CONFLICT OF INTEREST

The authors confirm that there is no known conflict of interest or common interest with any institution/organization or person.

AUTHOR CONTRIBUTION

Onur ARAZ: Determination of design processes of the study, data analysis and interpretation, reviewing the intellectual content, final approval and full responsibility of the study.

Volkan KAHYA: Reviewing the intellectual content, final approval and full responsibility of the study.

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