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Optimal design of rotational inertial double tuned mass dampers under random excitation

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ABSTRACT

The rotational inertial double tuned mass damper (RIDTMD) is a type of passive mass damper which includes a physical mass as well as a rotational mass. This rotational mass is produced by an inerter which is capable of providing large effective mass utilizing very little physical mass. By selecting the proper design parameters, the RIDTMD show promise at more effective response reduction of underlying primary systems in comparison to conventional tuned mass dampers (TMDs). However, when the primary system is subjected to random loads, the previously considered optimum design values for harmonic excitation are not effective. This motivates an investigation to determine the exact analytical optimum solution for selecting the stiffness and damping design values of RIDTMD when the primary structure is subjected to random force and base excitation. In this paper, an exact optimization solution procedure is presented with the goal of finding the optimum design values of the RIDTMD when mitigating the response of a structure subjected to random force and base excitation. Utilizing the obtained optimum design values, the effectiveness of the RIDTMD is also studied in comparison to conventional TMDs. The results of this study show that the RIDTMD with optimized stiffness and damping values outperforms the optimized conventional TMD; however, the degree of its increased effectiveness in reducing the main mass response is reliant upon the selection of appropriate pairs of secondary and rotational mass.

1. Introduction

Structures are subjected to various types of dynamic excitations. Of particular interest to structural engineers are wind, which directly loads a structure, and earthquakes, which load the structure as a result of base excitation. Reducing the effects of dynamic loads has motivated many researchers to study the use of supplemental mechanical vibration absorbers. Conventional tuned mass dampers (TMDs), which are designed to damp the vibration of a primary mass [1], are composed of a secondary mass, spring, and viscous damper. TMDs have been developed and used as a reliable device for structural vibration control of loads which can be modeled as a stationary process. The performance of a TMD is highly dependent on three parameters: (1) the ratio of the mass of the TMD to the main mass, (2) the frequency ratio of the TMD to the main mass, and (3) the TMD damping ratio. It has been found that by utilizing optimized parameters, the TMD can be effective at reducing the response of the main mass it is attached to [2].

Design parameters of TMDs have been obtained through the use of the H_∞ optimization criterion, which minimizes the maximum displacement response of the main mass in the frequency domain. The fixed-points theory, which is an approximation of the H_∞ method [2], is commonly used for TMD optimization. This method is based on the existence of two equal magnitude fixed points on the system's frequency response curve that do not depend on the system's damping level and are thus at the same location in either the zero or infinite damping condition. TMD optimum design parameters under harmonic force and base excitation for undamped primary systems have been obtained by applying the fixed points method [3]. In addition, analytical exact solutions for H_∞ optimization of the TMD for damped and undamped primary systems have been developed [4,5].

However, when the main system is subjected to random vibration, it is often the minimization of the mean square of the response that is considered [6]. In the literature, the minimization of the mean square response of the primary structure over all frequencies is called the H_2 optimization criterion. Utilizing this criterion, design formulas have been proposed for random force and base excited systems utilizing a TMD when the main system is undamped [3]. Using residue theory from complex analysis, exact solutions for TMD parameter optimization considering random force and base excitation have been proposed for a TMD attached to a damped single-degree-of-freedom (SDOF) system

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[4,7]. Furthermore, robust H_2 optimization using a numerical approach has been proposed [8] and a mixed analytical and numerical curve fitting approach has been utilized [9].

Variant types of TMD have been proposed, formulated, and investigated to determine optimum design values. Non-traditional vibration absorbers for random force vibration attached to an undamped single-degree-of-freedom primary system have been optimized utilizing an analytical solution [10]. Three-element type TMD [11], which have an additional spring in series with the viscous damper, have been proposed and optimized considering a random force excitation [12]. Another variant of the TMD is the multiple tuned mass dampers (MTMD) system which utilizes multiple separately tuned TMDs attached to a primary system. H_2 optimum parameters for the MTMD have been investigated using numerical methods for single- [13] and multi-degree-of-freedom primary systems [14]. In addition, numerical approaches implemented for minimax optimization of MTMD attached to multi-degree-of-freedom primary systems have also been developed [15].

In an effort to improve the effectiveness of TMDs, the utilization of supplemental rotational masses in TMDs has recently been studied [16–20]. The most important benefit of rotational devices is that they are able to produce a large effective mass by using a relatively small rotational physical mass; therefore, these devices potentially need smaller physical mass than a conventional TMD to be effective. Using the ball screw mechanism concept, the rotational inertial viscous damper (RIVD) was proposed and evaluated for use with toggle bracing for control of a single-degree-of-freedom system [16]. The tuned viscous mass damper (TVMD), which consists of a tuning spring connected to a viscous mass damper that contains a rotational inertial mass, has also been proposed [17]. The results of this study showed that the TVMD attached to a single-degree-of-freedom system was effective under harmonic and seismic loading. The authors of this work also designed and produced a ball screw mechanism, which provides 350 kg of effective mass utilizing only 2 kg of physical mass.

The rotational inertial double tuned mass damper (RIDTMD) is another type of rotational device which has been proposed recently [18]. The RIDTMD consists of a conventional TMD, but the typical viscous damper has been replaced with a tuned viscous damper [18]. The tuned viscous damper consists of a spring, axial dashpot, and a rotational mass, known as an inerter. While the rack and pinion mechanism has been proposed for use in the RIDTMD [18], different alternative mechanisms for transferring translational motion to rotation, such as a ball screw mechanism, could be utilize. The RIDTMD, with a small added rotational mass, has demonstrated effective results compared to the TMD under force harmonic excitation.

Optimum design values of the RIDTMD have been obtain using a numerical method optimization for force harmonic excitation [18]. Furthermore, the RIDTMD is a special case of inerter-base devices which have been optimally design only for random force excitation on the primary system using a numerical optimization method [20]. In addition, compared to other inerter-base devices with the same number of degrees of freedom, The RIDTMD shows better performance in the reduction of the dynamic magnification factor [20]. Because the investigation of optimum design values for this device have been limited to force excitation using numerical methods [18,20,21], it is important to develop the optimum design of this device through the exact analytical solution for both force and base excitations.

In this study, an analytical exact solution procedure for selecting the optimum values for the RIDTMD under random force and base excitation attached to an undamped primary system is provided. The rotational device, attached to a single-degree-of-freedom primary system, is formulated considering a general mechanism, not specifically a rack and pinion system, for transferring the linear motion to the rotational part of the RIDTMD. The analytical solution is performed based on the H_2 optimization criterion and the variance of the output equations are derived and presented analytically. Using the optimum values for

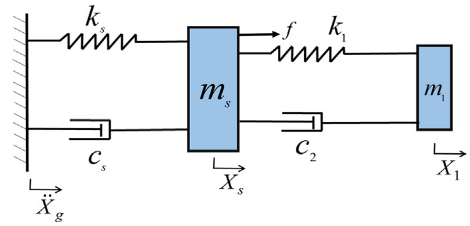

Fig. 1. Traditional TMD.

different mass ratio combinations, the performance of the device at controlling an undamped primary system under random force and base excitation is compared to the TMD. The optimized system response for different mass ratio combinations are also presented to evaluate the optimum mass ratio of this type of device when subjected to random vibration.

In the next section, dynamic modeling and explicit formulation of the RIDTMD is presented. Section 3 presents the optimum design formulas for TMD under random force and base excitation. Sections 4 and 5 present the H_2 optimum design of RIDTMD under random force and base excitation, respectively. Curves showing the optimal RIDTMD parameter design values for both cases are presented in Section 6. Section 7 covers the evaluation of the performance of the device with respect to the H_2 norm and the reduction of the dynamic magnification factor. Comparisons between the performance of the optimal RIDTMD and optimal TMD are also made in Section 7. Section 8 presents a summary and the conclusions of this study.

2. Rotational inertial double tuned mass damper (RIDTMD)

The primary physical difference between TMDs (Fig. 1) and rotational inertial double tuned mass dampers (RIDTMDs) (Fig. 2) is the replacement of the TMD's damper with a parallel rotational mass and viscous damper, often referred to as a rotational inertial device, which is in series with a tuning spring. The rotational mass works by transforming relative translation motion into the localized rotation of a small mass. While many mechanisms could potentially be used to produce this rotation, two alternatives have been primarily considered in the literature: the rack and pinion mechanism [18] and the ball screw mechanism [16,17] (Fig. 3).

The rotational velocity of this rotational mass, usually a flywheel, (ω) is based on the derivative of relative displacement between the two terminal ends of the device and α, a coefficient related to the mechanism's physics.

$$\omega = \alpha(\dot{x}_1 - \dot{x}_2) \tag{1}$$

For the ball screw mechanism,

$$\alpha = 2\pi/\rho \tag{2}$$

where ρ is the ball screw lead. For the rack and pinion mechanism,

$$\alpha = 1/r_c \tag{3}$$

where r_c is the pinion radius.

Utilizing Eq. (3) in calculating the kinetic energy of the rotational

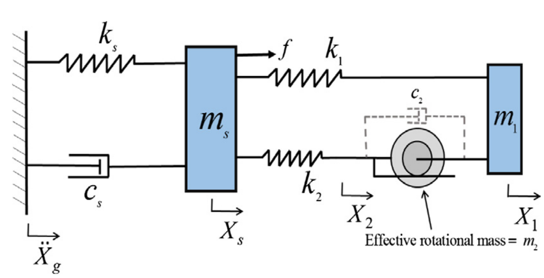

Fig. 2. Rotational inertial double tuned mass damper (RIDTMD).