ADVANCED SUSPENSION SYSTEM WITH TWO-DEGREE-OF-FREEDOM HYBRID MASS DAMPERS

Youssef Alsoufi

Department of Instrumentation and Control Engineering
Faculty of Mechanical Engineering
Czech Technical University in Prague , Czech Republic
Youssef.Alsoufi@fs.cvut.cz
https://orcid.org/0000-0001-5420-9027

Abstrakt:
Klíčová slova:

Abstract: Historical review and Full explanation about types of dampers are presented. The benefits and drawbacks of using two degrees of freedom rather than one degree in vibration suspension are discussed in this study. Dual-Loop Controller (DLC) with two-degree-of-freedom (rotational and displacement) Hybrid Mass Damper (HMD) attached to a single degree of freedom primary system is studied. The proposed model aims to improve the performance of damping and reduce energy consumption with achieving the fail-safe behavior.
Keywords: Dynamic vibration absorber, active suspension system, hybrid vibration absorber, dual loop controller.

1 Introduction

In general, vibration problems are caused by structural resonances or significant harmonic loads. Vibration absorbers can be added to increase structural damping and thus reduce vibrations. Passive and active vibration dampers are used widely to reduce the effects of structure oscillations. As a traditional Passive device, the Tuned Mass Damper (TMD) is an auxiliary mass–spring–damper system that is correctly tuned to a target mode, or a well-known harmonic perturbation. The device is then referred to as a Dynamic Vibration Absorber (DVA) [1]. The main advantages of the passive approach are the overall set-inherent up's stability and the fact that no energy is required to dampen the oscillations. The design of a single-degree-of-freedom (SDOF) TMD to attenuate the vibration of a single mode of the main structure under various conditions is studied in [2]. Using multiple SDOF TMDs to absorb more than one mode of the main structure is discussed in [3]. The problem with DVA is that in practice, the resonant frequency of the primary structure or the disturbance frequency may change during the time. However, the traditional DVA has no ability to automatically adjust its passive parameters or absorption frequency. So, to overcome this problem, an adaptive DVA was developed, which has the ability to self-adjust its absorption frequency. The adaptive DVA can adjust its absorption frequency by modifying its passive parameters. However, the effect of adaptive DVA can only perform well on vibration control with slow time-varying disturbances frequency. Adding an actuator to control the force that the reaction mass applies to the structure allows for more efficient method of treating multiple resonances and improves the vibration absorber's efficiency in controlling structural vibration caused by random disturbance forces. The resulted proposed device is called Active Mass Damper (AMD) which can be efficient on all the controllable modes [4]. However, the drawback of traditional AMD design is that all counter forces are created by the active element and therefore, the power consumption and the size of the active actuator become very large in a case dealing with large structural vibration [5]. In [6], different types of controller designs are reported. The authors in the research proposed a new concept of active vibration absorption, "Delayed Resonator" where a controlled time delay has been used in a feedback loop. In fact, the main difference between this new generation damper and TMD is that this device is mainly developed for harmonic perturbation rejection and thus cannot be considered as a fail-safe. A robust

alternative of the delayed resonator using a spectral approach is proposed in [7] and [8].
Recently, a new definition appeared to refer to the new generation of dampers which combining the behavior of an optimal TMD and the active damping devices. These dampers are named Hybrid Mass Damper (HMD), or Hybrid Vibration Absorber (HVA). The HMD over the last decade is used widely and many control laws have been proposed. The researchers attempted to prove that using HMD will enhance the dynamic systems, decreasing the energy consumption by decreasing the control effort of the actuator, reducing the stroke of the moving mass. Optimal control is used to combine structural damping with a restricted stroke of the actuator to create a HMD in [9]. As a result, the goal of this research is to develop an advanced suspension system that will improve performance while also reducing energy consumption. The advanced suspension system is proposed to be built around a hybrid mass damper, as shown in the following paragraphs.

2 Review of passive tuned mass damper

The passive devices use the internal motion of the structure to enhance the absorption forces and dissipate the vibration energy of a specific resonance. They do not require an external power supply to operate. The control forces applied to the structure are only dependent on the structural motion in ideal passive devices. TMD absorbers are widely used in mechanical systems for vibration control. A mass is attached to a structure via a parallel spring-dashpot system in a TMD device. The mechanism is discussed in [10] where the added mass m of the TMD is usually about 10% of the main structure mass and mounted on a spring and a damper. The equations for stiffness and damping can be found in [11]. In [12], four optimum design methods for a dynamic absorber are compared when they are applied to a single-degree-of-freedom system with primary damping. The undamped natural frequency of the TMD is determined by its mass and stiffness, and it have to be tuned close to the natural frequency of the main structure. As a result, at the resonance frequency, reducing the amplitude of the main structure is achieved by properly adjusting TMD parameters damping. On the other side, because of the high sensitivity of the suppression amplitude to TMD stiffness and damping parameter variations, the challenge in designing TMD is hard tuning correctly the absorber's stiffness and damping parameters. In addition to another drawback in DVAs work which is that they are tuned to damp only one specific resonance. The problem was solved by adding an actuator that controls the force applied to the structure by the reaction mass. we mention it in section 3. In fact, parameter variations in the dynamics and natural frequency variations in the main structure have an impact on the overall performance of designed systems. Some methods designed to deal with this sensitivity. One of them is discussed in [13]. Comparison between passive and active dampers is presented in [14]. In [15], H_2 and H_∞ norms are used to optimize the system response under random and harmonic excitations with two-degree-of-freedom TMD.

3 Review of semi-active and active mass dampers

The controllers must be designed in such a way that the control effectiveness and energy consumption are balanced. Control strategies can be divided into two categories from this perspective: active and semi-active. Semi-active controllers require a small external power source to operate and rely on the structure's motion to generate control force. An external power source can be used to adjust the magnitude of the force. Since its introduction in [16], By bridging the gap between purely passive and purely active suspensions, these systems have attempted to combine the benefits of both passive and active devices. Semi-active devices, on the other hand, have fewer power requirements than active devices. Furthermore, while semi-active devices cannot inject mechanical energy into a controlled structural system, they do have properties that can be controlled to reduce the system's response to an optimal level. In [17], in order to design an optimal suspension, a Two-degree-of-freedom (TDOF) model of a semi-actively suspended vehicle is used as a starting point.

AMD, on the other hand, works by moving the auxiliary mass with an actuator connected between the structure and the auxiliary mass to generate reaction forces. The equations of AMD can be found in [18]. The AMD concept has been used in a variety of applications over the last few decades. To suppress the vibrations of structures, for example, moving a relatively small mass with a limited amplitude is required. This explains why AMDs have a limited ability to absorb large earthquake excitations. However, the goal of structural vibration energy absorption under small earthquake excitations or strong wind is met. As a result of active control, the auxiliary system's response motion is usually increased. Therefore, not only is it desirable to improve control efficiency, but also to limit auxiliary mass motion as much as possible. The allowable amplitude of the AMD is generally determined by the size of the AMD or the installation space, with strict limitations. The limitations of the auxiliary mass's amplitude are one of the main reasons for AMDs' poor performance. In [9], a control law for AMD that effectively suppress the vibrations of a SDOF structural system under the amplitude constraint of the auxiliary mass is studied.

The research [19] discussed the advantages of AMD compared with PSS. Pneumatic, linear motor, hydraulic actuators, and other force actuators can provide the required forces in [20]. A four-degree-of-freedom half-car model with active suspension is studied in [21]. Recently with self-driving vehicles, the suspension of the vertical vibration amplitude of the cameras which mounted on the vehicles is needed to reduce the unwanted motion effects.

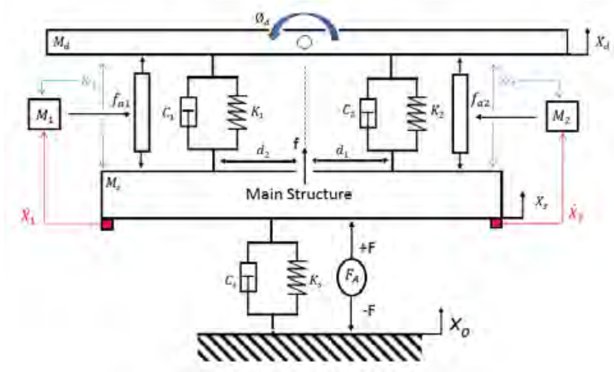


Fig. 1: Advanced suspension system with two degree of freedom hybrid mass damper

4 Review of hybrid mass damper

After decades of research, a new class of dampers has emerged that attempts to combine multiple objectives and features at the same time. These devices are gathered under the common name of Hybrid Mass Dampers (HMDs), or Hybrid Vibration Absorbers (HVAs) which combine passive and active vibration control. Combining passive and active elements the system is fail-safe that the damper will behave as a passive device even when the feedback control is turned off. The goal of using HMD may differ from one to the other. For example, in [22], H_∞ optimal design of HMD is used for the minimization of the resonant vibration amplitude of a SDOF vibrating structure. In [23], a pole placement technique is proposed to ensure performance and stability . And a special pole placement controller is designed such that all vibration modes of the flexible structures become critically damped in [24]. In [25], a dual loop approach is preferred to increase the stability margins. Improving the performance and stability of hybrid mass dampers by creating a hyperstable controller is studied in [26].

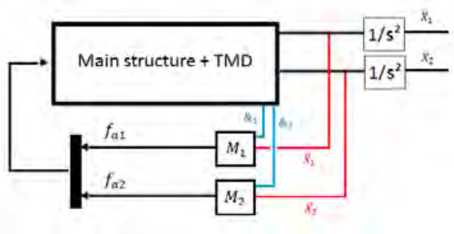


Fig. 2: Controller loops.

5 proposed model

More than one mode of vibration of an absorber body relative to a primary system can be tuned to suppress single-mode vibration of a primary system. Therefore, mitigation of the response under harmonic excitations is proposed in the present model. Fig. 1 represents a model with two-degree of freedom HMD attached to a single degree of freedom primary system. Vertical displacement movement and rotational movement around the center of the auxiliary mass as a basic design is proposed. The results show that an optimal two-degree of freedom TMD outperforms a traditional single-degree of freedom TMD with optimal mass distribution, even when the absorber's rotary inertia approaches zero. We will use this proof as a starting point for converting TMDs to HMDs later.
It is well known that when we use an HMD to attenuate the vibration; the passive part has a positive impact on reducing the amplitudes of the vibrations i.e. damping the oscillations occurs without the need for any external power. The frequency band where the absorber effectively suppresses vibrations, on the other hand, is relatively

narrow, being centered at the absorber's natural frequency. Because the physical absorber is never perfect, i.e. it has nonzero damping, the damper cannot absorb the vibration completely even if the vibration frequency is identical to the absorber's natural frequency. Solving this problem is presented in section 2. The proposed model aims to accomplish three main goals: (1) improve the performance of the suspension system, (2) reduce energy consumption, and (3) ensure fail-safe behavior. The passive parts are TMDs optimally tuned using Den Hartog's law in [1], and the active control forces (f_{a1}) and (f_{a2}) are introduced between the two masses. The general concept of dual-loop controller is introduced in [4] and is used precisely as parallel dual loops and act on the same transducer in [27] which is the closest to our model with the exception that in the previous research a dual loop controller (DLC) with a single degree of freedom was used. The proposed controller employs two inputs: (i) the relative displacement between the inertial mass and the main structure, and (ii) the absolute velocity of the main structure. Subsequently, dual loops on each side of the damper are used. One to detune the HMD (negative stiffness feedback), and the other to damp the main structure (direct velocity feedback). Fig 2 illustrates that since the relative velocity or absolute acceleration will be measured, so integration has to be added to the control laws M1 and M2. The damper's rotational movement allows it to absorb not only vertical but also angle disturbances in some cases. Also, as long as we use an active impact in our design, we have to keep in mind the necessity of restricting the amount of energy required for absorption operation. We mentioned it in section 1. Indeed, increasing the frequency bandwidth where the absorber suppresses the vibrations has been discussed in many previous studies, such as in [8]. So, in this article, we will try to demonstrate the superiority of TDOF over SDOF by comparing them first without any active parts and then improving the concept by adding a single loop PD controller to the TDOF as an active part. Let's start with our basic design shown in Fig. 1. The main structure has a natural frequency $w_s = \sqrt{k_s/m_s}$ and damping ratio $\zeta_s = c_s/2\sqrt{k_s m_s}$ and is subjected to a base excitation x_0, an external disturbance force f. We will use a harmonic disturbance as an external force in this study because it represents a more realistic situation in a TMD's field of application (hybrid or not) .x_d is translation and θ_d is rotation. m_d is the mass of the damper and I_d is the rotational inertia about its center of mass $I_d = m_d\rho^2$ where ρ is the radius of gyration. The absorber is connected to the primary system at distances d_1 and d_2 from its center of mass via springs and dashpots. Let's consider first the case where $d_1 = d_2 = d$. We will replace the spring and the damping device by control-force vector $[u_1, u_2]^{\mathrm{T}}$ where

$$\begin{aligned} u_1 &= k_1(x_1 - x_s) + c_1(x_1' - x_s') \\ u_2 &= k_2(x_2 - x_s) + c_2(x_2' - x_s') \end{aligned} \tag{1}$$

x_1, x_2 are the displacements of the damper in the direction of $x_s$at the connection locations.

$$\begin{aligned} m_d x_D'' &= -u_1 - u_2 \\ I_d \theta'' &= u_1 d - u_2 d \end{aligned} \tag{2}$$

$$m_s x_s'' = u_1 + u_2 - k_s x_s - c_s x_s' + k_s x_0 + c_s x_0' + f \tag{3}$$

According to the proposed model , we note that:

$$x_d = \frac{x_1 + x_2}{2}, \theta_d = \frac{x_2 - x_1}{2d}, I_d = m_d\rho^2 \tag{4}$$

$$\begin{aligned} \frac{m_d}{2}(x_1'' + x_2'') &= -u_1 - u_2 \\ \frac{m_d}{2}(x_2'' - x_1'') &= u_1\frac{d^2}{\rho^2} - u_2\frac{d^2}{\rho^2} \end{aligned} \tag{5}$$

All the previous equations are covered in matrix form:

$$\begin{bmatrix} m_d/2 & m_d/2 & 0 \\ -m_d/2 & m_d/2 & 0 \\ 0 & 0 & m_s \end{bmatrix} \begin{bmatrix} x_1'' \\ x_2'' \\ x_s'' \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & c_s \end{bmatrix} \begin{bmatrix} x_1' \\ x_2' \\ x_s' \end{bmatrix} +$$

$$\begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & k_s \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_s \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ c_s \end{bmatrix} x_0' + \begin{bmatrix} 0 \\ 0 \\ k_s \end{bmatrix} x_0 + \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} f$$

$$+\begin{bmatrix}-1 & -1\\ \frac{d^2}{\rho^2} & -\frac{d^2}{\rho^2}\\ 1 & 1\end{bmatrix}\begin{bmatrix}u_1\\ u_2\end{bmatrix} \tag{6}$$

Or

$$Mq'' + Cq' + Kq = B_{00}X_0 + B_{01}X_0' + B_f f + B_u u \tag{7}$$

$$where \;\; q = [x_1, x_2, x_s]^{\mathrm{T}}$$

If we want to use the model as a HMD then,

$$\begin{bmatrix}0 & -1 & -1\\ 0 & \frac{d^2}{\rho^2} & -\frac{d^2}{\rho^2}\\ 1 & 1 & 1\end{bmatrix} f \;\; where \; f = [f_s, f_{a1}, f_{a2}]^{\mathrm{T}}$$

let us assume that all the initial values equal zero.

$$x = \begin{bmatrix}q\\ q'\end{bmatrix} \tag{8}$$

$$x' = Ax + B_1 f + B_2 u\,;\, u = \begin{bmatrix}u_1\\ u_2\end{bmatrix} \tag{9}$$

$$A = \begin{bmatrix}0 & I\\ -M^{-1}K & -M^{-1}C\end{bmatrix} \tag{10}$$

$$B_1 = \begin{bmatrix}0\\ M^{-1}B_f\end{bmatrix}, B_2 = \begin{bmatrix}0\\ M^{-1}B_u\end{bmatrix} \tag{11}$$

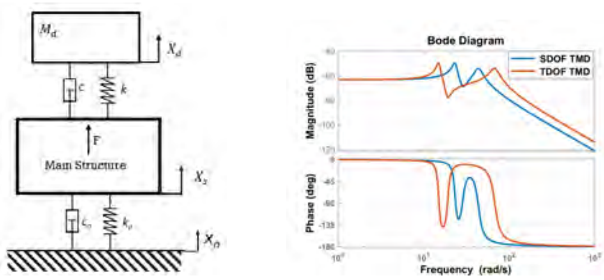
The cost output can be interpreted as the absolute or relative displacement, velocity, or acceleration of the primary system, u is a matrix given by (1) as a state feedback matrix P multiplied by the measurement output y. So:

$$u = \begin{bmatrix}-k_1 & k_1 & c_1 & 0 & 0\\ -k_2 & 0 & 0 & k_2 & c_2\end{bmatrix} y = Py \tag{12}$$

where

$$y = \begin{bmatrix}x_s, & x_1 - x_s, & x_1' - x_s', & x_2 - x_s, & x_2' - x_s'\end{bmatrix}^{\mathrm{T}}$$

To make the comparison easier, we divided the frequency band into many characteristic frequencies after evenly distributing the stiffness and damping device values of the SDOF into two spring-dashpot couples for the TDOF concept. TDOF TMD without any active parts had a better overall performance in the critical frequencies like resonance frequency of the main structure w =35.7 rad/sec where TDOF shows more efficiency than SDOF, see Fig. 4. Let's start our study when $w << w_n$=10 rad/s with arbitrary constant values for stiffness and damping device. The bode plot in this part of frequencies shows constant values for the amplitude and phase in both models Fig.3(b).



(a) a

(b) b

Fig. 3: SDOF TMD

For critical frequencies $w_1 = w_{SDOF}$= 23.3 rad/sec and $w_2 = w_{TDOF}$=14.9 rad/sec, we can see that the magnitude of TDOF TMD at resonance frequency w_1 is around 0.3 $\times 10^{-3}$ (m) with phase (-14.9) deg Fig. 5 (b). On the other side, SDOF TMD reaches around 1 $\times 10^{-3}$ for a peak amplitude at resonance frequency w_2 and phase

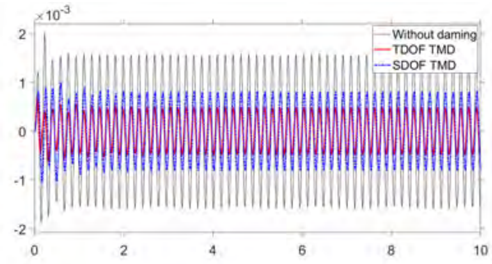
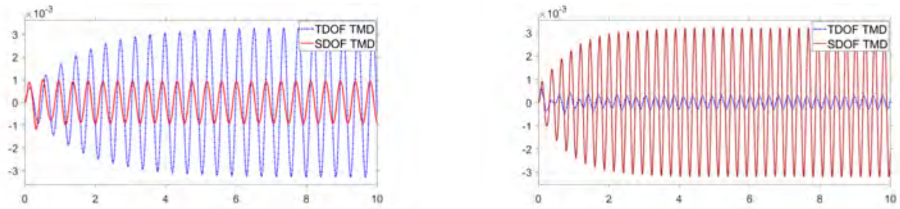


Fig. 4: Absolute displacement X_S at natural frequency of main structure.



(a) a

(b) b

Fig. 5: Absolute displacement x_s/x_0 at critical frequencies (a) 14.9 rad/sec , (b) 23.3 rad/s.

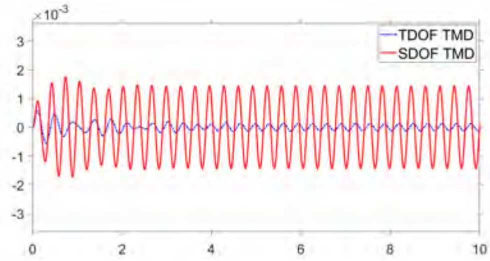


Fig. 6: Absolute displacement X_S at frequency w=19.5 rad/sec

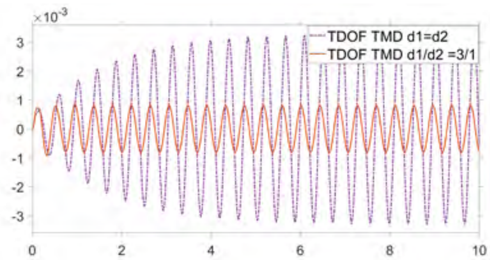
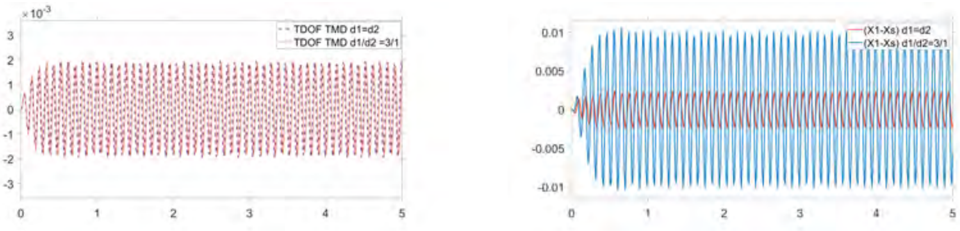


Fig. 7: Absolute displacement X_S at frequency w_n=14.9 rad/sec, $d_1 = d_2$(dashed) and $d_1/d_2 = 3/1$ (line)

(-3.85) deg Fig. 5 (a). Also at frequency w=19.5 rad/s it is clear that TDOF TMD has a higher performance than SDOF TMD Fig. 6 where $Amp_{(SDOF)} = 1.5 \times 10^{-3}$ (m) and $Amp_{(TDOF)} = 0.25 \times 10^{-3}$ (m).

The second case is changing the dimensions d_1, d_2 when $d_1 \neq d_2$. As is obvious in Eq. 6, the performance of the TDOF TMD system depends on the ratio of the absorber's radius of gyration ρ to the distance d from the mount points to the center of mass of the absorber. So, according to Eq. (6) and Fig. 1 when d=0, that corresponds to the optimal SDOF TMD. Even if the ratio 3/1 could be not an optimal rate, let's use it for this study as a ratio d_1/d_2 and substitute it in the equation while keeping ρ constant. The results show that the performance of the damper is increased at the resonance frequency 14.9 rad/sec compared with the same model when $d_1 = d_2$ Fig. 7 with approximately (-80) deg as a phase-shift.

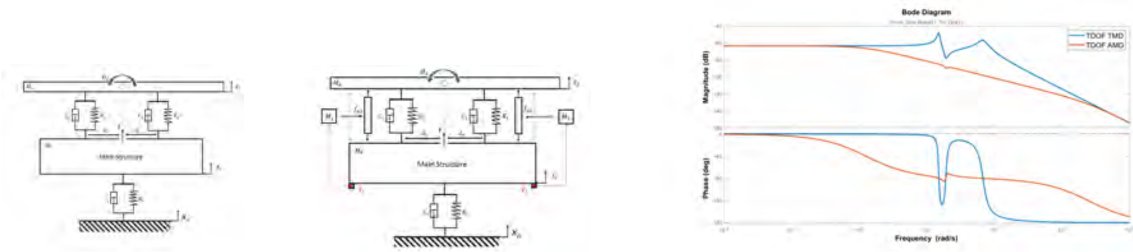
Fig. 8(b) illustrates the relative displacement spectrum (X1-Xs) of TDOF TMD at frequency w=66 rad/sec where the amplitude for both tested systems is the same Fig. 8(a). One can see in the figure that the superior performance of changing the dimensions of d_1, d_2 is obtained at the cost of a much larger relative displacement of



(a) a (b) b

Fig. 8: (a).Absolute displacement x_s/x_0 when d1=d2(dashed) and when d1/d2=3/1(line). (b).Relative displacement (X1-Xs) when d1=d2(red) and d1/d2=3/1 (blue) w=66 rad/s

the moving mass (m_d).



(a) a (b) b (c) c

Fig. 9: (a) TDOF TMD (b) TDOF AMD (c) Bode plot

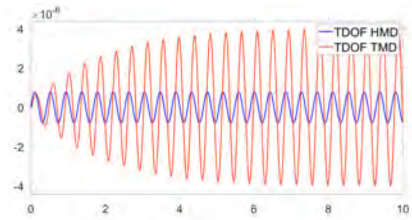


Fig. 10: Absolute displacement X_S at frequency w=14.9 rad/sec TDOF-TMD(red), TDOF-HMD (blue)

In fact, achieving optimal damper performance without combining active and passive parts to create an HMD with fail-safe behavior is impossible. PD controller as a single loop controller is proposed for this study. So the equations:, $M_1 = g_{c1}s + g_{k1}$ and $M_2 = g_{c2}s + g_{k2}$ where $g_{c1} > 0$, $g_{c2} > 0$. In the bode plot Fig. 9, we can see the increase in gain and phase, as well as the response of the damper at the peak amplitude of TDOF TMD in the time domain Fig. 10 where the residual motion of m_s is more than 5 times smaller with the HMD than with the TMD with amplitude 0.8 $\times 10^{-6}$.

6 conclusion

A historical overview of the most commonly created and developed types of dampers is presented. The superiority of the two-degree of freedom (displacement and rotational movements) dampers over a single degree with only displacement flexibility is demonstrated in this article by comparing the performance of TDOF TMD with SDOF TMD for a very wide frequency range. In the end, a hybrid mass damper with dual loop controllers is used to improve the damper's performance in both the time and frequency domains. The industrial application will need to provide derivative control filtering, as well as AMD performance, which will be thoroughly tested on various industrial applications in future research, including comparative performance analysis and energy consumption reduction.

Tab. 1: **Identified parameters of the models:**

Parameters	Values	Units	Parameters	Values	Units
ms	1.1570	Kg	cs	4.4317	N/m/s
md	0.5200	Kg	k	423.9525	N/m
ks	1482.2504	N/m	c	2.4198	N/m/s

Tab. 2: **Nomenclature**

Abbreviation	Definition
AMD	Active Mass Damper
DLC	Dual-Loop-Controller
DVA	Dynamic Vibration Absorber
HMD	Hybrid Mass Damper
HVA	Hybrid Vibration Absorber
PSS	Passive Suspention System
SDOF	Single-Degree-Of-Freedom
TDOF	Two-Degree-Of-Freedom
TMD	Tuned Mass Damper

Acknowledgement

The author has been supported by the Grant Agency of the Czech Technical University in Prague, grant No. SGS20/159/OHK2/3T/12.

References

[1] JP DenHartog. Mechanical vibrations, (1956), 373.

[2] GB Warburton. Optimum absorber parameters for various combinations of response and excitation parameters. *Earthquake Engineering & Structural Dynamics*, 10(3):381–401, 1982.

[3] L Kitis, BP Wang, and WD Pilkey. Vibration reduction over a frequency range. *Journal of Sound and Vibration*, 89(4):559–569, 1983.

[4] André Preumont and Kazuto Seto. *Active control of structures*. Wiley Online Library, 2008.

[5] Andreas H Vonflotow, Andrew Beard, and Don Bailey. Adaptive tuned vibration absorbers: tuning laws, tracking agility, sizing, and physical implementations. *nce*, pages 437–454, 1994.

[6] N Olgac and BT Holm-Hansen. A novel active vibration absorption technique: delayed resonator. *Journal of Sound and Vibration*, 176(1):93–104, 1994.

[7] Dan Pilbauer, Tomáš Vyhlídal, and Wim Michiels. Optimized design of robust resonator with distributed time-delay. *Journal of Sound and Vibration*, 443:576–590, 2019.

[8] Matěj Kuře, Tomáš Vyhlídal, Wim Michiels, and Islam Boussaada. Spectral design of robust delayed resonator by double-root assignment. *IFAC-PapersOnLine*, 51(14):72–77, 2018.

[9] Kazunobu Yoshida and Itaru Matsumoto. Vibration suppression control for a multi-degree-of-freedom structural system using an amd with restricted stroke. In *2009 International Conference on Networking, Sensing and Control*, pages 912–917. IEEE, 2009.

[10] MA Acar and C Yilmaz. Design of an adaptive–passive dynamic vibration absorber composed of a string–mass system equipped with negative stiffness tension adjusting mechanism. *Journal of Sound and Vibration*, 332(2):231–245, 2013.

[11] Jacob Pieter Den Hartog. *Mechanical vibrations*. Courier Corporation, 1985.

[12] YZ Wang and SH Cheng. The optimal design of dynamic absorber in the time domain and the frequency domain. *Applied Acoustics*, 28(1):67–78, 1989.

[13] Hui Zhang, Yang Shi, and Aryan Saadat Mehr. Robust non-fragile dynamic vibration absorbers with uncertain factors. *Journal of Sound and Vibration*, 330(4):559–566, 2011.

[14] Sang-Myeong Kim, Semyung Wang, and Michael J Brennan. Dynamic analysis and optimal design of a passive and an active piezo-electrical dynamic vibration absorber. *Journal of sound and vibration*, 330(4):603–614, 2011.

[15] Lei Zuo and Samir A Nayfeh. The two-degree-of-freedom tuned-mass damper for suppression of single-mode vibration under random and harmonic excitation. 2006.

[16] Dean Karnopp, Michael J Crosby, and RA Harwood. Vibration control using semi-active force generators. 1974.

[17] D Hrovat, DL Margolis, and M Hubbard. An approach toward the optimal semi-active suspension. 1988.

[18] Isao Nishimura, Takuji Kobori, Mitsuo Sakamoto, Norihide Koshika, Katsuyasu Sasaki, and Satoshi Ohrui. Active tuned mass damper. *Smart Materials and Structures*, 1(4):306, 1992.

[19] TPJ Van der Sande, BLJ Gysen, IJM Besselink, JJH Paulides, EA Lomonova, and H Nijmeijer. Robust control of an electromagnetic active suspension system: Simulations and measurements. *Mechatronics*, 23(2):204–212, 2013.

[20] Jiangtao Cao, Honghai Liu, Ping Li, and David J Brown. State of the art in vehicle active suspension adaptive control systems based on intelligent methodologies. *IEEE transactions on intelligent transportation systems*, 9(3):392–405, 2008.

[21] Panshuo Li, James Lam, and Kie Chung Cheung. Multi-objective control for active vehicle suspension with wheelbase preview. *Journal of Sound and Vibration*, 333(21):5269–5282, 2014.

[22] YL Cheung, WO Wong, and L Cheng. Design optimization of a damped hybrid vibration absorber. *Journal of sound and vibration*, 331(4):750–766, 2012.

[23] MH Tso, Jing Yuan, and WO Wong. Design and experimental study of a hybrid vibration absorber for global vibration control. *Engineering structures*, 56:1058–1069, 2013.

[24] MH Tso, Jing Yuan, and WO Wong. Suppression of random vibration in flexible structures using a hybrid vibration absorber. *Journal of sound and vibration*, 331(5):974–986, 2012.

[25] Peter Hagedorn and Gottfried Spelsberg-Korspeter. *Active and passive vibration control of structures*. Springer, 2014.

[26] Christophe Collette and Simon Chesne. Robust hybrid mass damper. *Journal of sound and vibration*, 375:19–27, 2016.

[27] Simon Chesne, G Inquieté, P Cranga, F Legrand, and B Petitjean. Innovative hybrid mass damper for dual-loop controller. *Mechanical Systems and Signal Processing*, 115:514–523, 2019.

Selected article from
Tento dokument byl publikován ve sborníku

Nové metody a postupy v oblasti přístrojové techniky, automatického řízení a informatiky 2021
New Methods and Practices in the Instrumentation, Automatic Control and Informatics 2021

15. 9. – 17. 9. 2021, Žatec

ISBN 978-80-01-06889-2

Web page of the original document:
http://iat.fs.cvut.cz/nmp/2021.pdf

Obsah čísla/individual articles:
http://iat.fs.cvut.cz/nmp/2021/

Ústav přístrojové a řídicí techniky, FS ČVUT v Praze, Technická 4, Praha 6