The tuned mass-damper-inerter for harmonic vibrations suppression, attached mass reduction, and energy harvesting

Laurentiu Marian1 and Agathoklis Giaralis2*

1AKT-II Ltd, 100 St John Street, EC1M 4EH, London, UK
2Department of Civil Engineering, City, University of London, Northampton Square, EC1V 0HB, London, UK

Abstract. In this paper the tuned mass-damper-inerter (TMDI) is considered for passive vibration control and energy harvesting in harmonically excited structures. The TMDI couples the classical tuned mass-damper (TMD) with a grounded inerter: a two-terminal linear device resisting the relative acceleration of its terminals by a constant of proportionality termed inertance. In this manner, the TMD is endowed with additional inertia, beyond the one offered by the attached mass, without any substantial increase to the overall weight. Closed-form analytical expressions for optimal TMDI parameters, stiffness and damping, given attached mass and inertance are derived by application of Den Hartog’s tuning approach to suppress the response amplitude of force and base-acceleration excited single-degree-of-freedom structures. It is analytically shown that the TMDI is more effective from a same mass/weight TMD to suppress vibrations close to the natural frequency of the uncontrolled structure, while it is more robust to detuning effects. Moreover, it is shown that the mass amplification effect of the inerter achieves significant weight reduction for a target/predefined level of vibration suppression in a performance-based oriented design approach compared to the classical TMD. Lastly, the potential of using the TMDI for energy harvesting is explored by substituting the dissipative damper with an electromagnetic motor and assuming that the inertance can vary through the use of a flywheel-based inerter device. It is analytically shown that by reducing the inertance, treated as a mass/inertia-related design parameter not considered in conventional TMD-based energy harvesters, the available power for electric generation increases for fixed attached mass/weight, electromechanical damping, and stiffness properties.

Keywords: tuned mass damper; inerter; passive vibration control; energy harvesting; weight reduction; electromagnetic motor; optimal design

1. Introduction

The concept of the dynamic vibration absorber (DVA) is historically one of the first strategies for passive vibration control of dynamically excited mechanical and civil engineering structures and structural components (Frahm 1911). It relies on attaching a free-to-vibrate mass to the structural system whose motion is to be suppressed (primary structure), such that significant kinetic energy is transferred from the primary structure to the attached mass. Considering a linear spring in parallel with a dashpot (e.g., a linear viscous damper) to attach the vibrating mass to the primary structure, the so-called tuned mass-damper (TMD) is, arguably, the most widely studied passive DVA in the literature (e.g., Ormondroyd and Den Hartog 1928, Brock 1946, Den Hartog 1956, Warburton 1982, Rana and Soong 1998, Asami et al 2002, Krenk 2005, Ghosh and Basu
2007, Bakre and Jangid 2007, Leung and Zhang 2009, Tributch and Adam 2012, Bortoluzzi et al. 2015, Salvi and Rizzi 2016) and the most commonly used in practical applications. The widespread use of the classical linear TMD is mainly due to the existence of simple and well-established design approaches seeking to determine optimal TMD stiffness and damping properties that minimize the response of a given dynamically excited primary structure for an a priori fixed attached mass. Focusing on periodic narrow-band excitations, Den Hartog (1956) established a semi-empirical TMD design approach by relying on the observation that all frequency response functions (FRFs) of a TMD-equipped undamped single degree-of-freedom (SDOF) primary structure pass through the same two points. Based on this “fixed point” theory, Den Hartog (1956) and Brock (1946) reached simple closed-form expressions for the TMD stiffness and damping properties, widely used in practical TMD design, to suppress the peak displacement of sinusoidal force-excited undamped SDOF primary structures (see also Krenk 2005). Further, Warburton (1982) followed the above design approach to derive TMD design formulae minimizing different response quantities of interest for harmonic force and base-excited undamped SDOF primary structures. More recently, Ghosh and Basu (2007) demonstrated that the fixed point theory leads to near-optimal TMD vibration suppression performance for the case of lightly damped SDOF primary structures with critical damping ratio up to 3%, applicable to a wide range of structures and structural components. Notably, the above TMD design formulae can be further applied to suppress the vibratory motion corresponding to a single (e.g. the dominant) structural mode shape in the case of lightly damped multi degree-of-freedom (MDOF) primary structures (e.g. Rana and Soong 1998).

Further to vibration suppression, the potential of the TMD to harvest energy from large-amplitude low-frequency oscillating primary structures has been recently recognized (Rome et al 2005) and explored by various researchers focusing primarily on large-scale (civil engineering) primary structures. In particular, TMDs can achieve simultaneous vibration suppression and energy generation by employing either electromagnetic (EM) devices (e.g. Tang and Zuo 2012, Shen et al 2012, Zuo and Tang 2013, Gonzalez-Buelga et al 2014, Shen et al 2016), or piezoelectric materials (e.g. Adhikari and Ali 2013) to connect the TMD mass to the primary structure as opposed to using only dampers. In this manner, part of the kinetic energy of the primary structure is transformed into electric energy instead of being “lost” at the dampers in the form of heat. The thus generated energy may be stored to batteries for later use (Zuo and Tang 2013), or can be used to achieve energy-autonomous semi-active or even active TMD vibration control strategies (Tang and Zuo 2012, Gonzalez-Buelga et al 2014), or to power wireless sensors for structural health monitoring (Shen et al 2012, Makihara et al 2015).

Despite being widely used in practice, the classical (linear passive) TMD is known to suffer from the problem of “detuning” due to such reasons as nonlinear behaviour of the primary structure (e.g. Domizio et al 2015), and/or uncertainty and variations to the dynamic properties of the primary structure over time (e.g. Wang and Lin 2015). Detuning affects significantly the TMD vibration suppression performance (and consequently its potential for energy harvesting), especially for the case of harmonic/narrow band excitations as its effectiveness depends heavily on ensuring resonance between the primary structure and the TMD. To this end, different strategies have been considered to enhance the robustness to detuning of the passive TMD for the purpose of controlling a single primary structure vibration mode. One such strategy is to use hysteretic/yielding components to attach the TMD mass to the primary structure (e.g. Ricciardeli and Vickery 1999) which widens the operational TMD frequency range around the target primary
structure natural frequency. Nevertheless, optimal design of inelastic TMDs is considerably more challenging compared to the linear TMD. Alternatively, robustness to detuning effects can be achieved by use of multiple TMDs (MTMDs) linked in parallel (e.g. Xu and Igusa, 1992, Yamaguchi and Harnpornchai 1993) or in series (Zuo 2009). In the parallel configuration, each individual TMD is tuned to a different frequency such that the effective frequency band becomes wider. In the series configuration, a chain of two or more appropriately determined masses are attached to the primary structure and tuned to achieve “multiple resonance” at the cost of excessive attached mass displacements. Parallel MTMDs have been considered for wind-induced vibration suppression in piers of cable-stayed bridges (Casciati and Giuliano 2009) and for traffic-induced vibrations suppression in (foot-)bridges (e.g. Lin et al 2005), among other applications. Nevertheless, optimal MTMD design is appreciably more involved than single TMD design (see e.g. Jokic et al. 2011) due to the increased number of design variables, while heuristic/experiential assumptions need to be made for the mass distribution among the TMDs (see e.g. Bandivadekar and Jangid 2012, Yang et al. 2015).

To this end, it is argued that, perhaps, the simplest and most straightforward way to enhance the performance and robustness to detuning of the classical single TMD is to increase the attached mass for which “optimum” stiffness and damping parameters is sought in TMD design. Indeed, the larger the attached TMD mass considered, the more effective an optimally designed linear TMD becomes to suppress excessive primary structure vibrations and the less sensitive to detuning effects (see e.g. De Angelis et al 2012 and references therein). Nevertheless, these benefits come at the cost of an increase total weight of the overall TMD-equipped structural system. To circumvent the latter trade-off, this paper considers coupling the classical linear TMD with an inerter device, introduced by Smith (2002), in a so-called “sky-hook” configuration as has been recently proposed by the authors (Marian and Giaralis 2014). In this manner, the resulting tuned mass-damper-inerter (TMDI) configuration exploits the mass amplification effect of the inerter (i.e., a linear two-terminal device of negligible mass/weight which resists the relative acceleration of its terminals) to increase the inertia of the attached mass, without increasing the overall weight of the controlled structure. In fact, the authors showed that for the same attached mass the TMDI performs better than the classical TMD, treated as a special case of the TMDI, in suppressing the displacement variance of stochastically based-excited SDOF and MDOF primary structures (Marian and Giaralis 2013, 2014). More recently, the potential of the TMDI for the seismic protection of primary structures modelled as SDOF systems has been explored by Pietrosanti et al (2017) and by Masri and Caffrey (2017), while Giaralis and Petrini (2017) considered the use of TMDI for wind-induced vibration mitigation in a benchmark tall building accounting for vortex shedding effects.

Herein, closed-form formulae are derived for optimal TMDI design in harmonically excited undamped SDOF primary structures based on the fixed point theory. These formulae are then used to quantify the gains in terms of vibration suppression and of weight reduction for optimally designed TMDI vis-à-vis the classical TMD. Further, the incorporation of a linear electromagnetic motor shunted by a resistive load is considered to gauge the potential of the TMDI for energy harvesting. This is analytically assessed by assuming the availability of a flywheel-based inerter device with varying mass amplification property. The latter consideration introduces a new “degree of freedom” which allows to vary the apparent inertia of the energy harvester leveraging the trade-off between vibration suppression and energy harvesting at will, without any changes to the attached mass.
Overall, apart from the novel closed-form expressions for the TMDI design for harmonic excitations, this paper makes original contributions by analytically quantifying (1) the vibration suppression performance enhancement of the TMDI compared to the classical TMD in harmonically force-excited and support-excited primary structures, (2) the weight reduction achieved by the TMDI compared to the classical TMD as a function of the inerter mass amplification property for a predefined vibration suppression performance, and (3) the increase of the available electric power to be generated from harmonically excited primary structures by employing a passive energy harvesting enabled TMDI with varying inertance.

The remainder of the paper is organized as follows. In Section 2 the ideal flywheel-based inerter is briefly presented and the governing equations of motion and associated frequency response functions of TMDI equipped SDOF primary structures are furnished. In Section 3, closed-form expressions for the design of the TMDI for harmonically excited primary structures are derived based on the fixed point theory and the benefits of the TMDI vis-à-vis the TMD in terms of vibration suppression and weight reduction are analytically quantified. Section 4 introduces an energy harvesting enabled TMDI and quantifies analytically its vibration suppression and power generation capabilities for harmonically excited primary structures, while Section 5 quantifies the increase to the available energy for harvesting by varying the inerter property of the energy harvesting enabled TMDI. Finally, Section 6 summarizes the main conclusions of the work.

2. The tuned mass-damper-inerter for single-degree-of-freedom (SDOF) structures

2.1 Rack-and-pinion flywheel-based ideal inerter

The ideal inerter was conceptually defined by Smith (2002) as a linear two terminal mechanical element of negligible physical mass/weight developing an internal (resisting) force $F$ proportional to the relative acceleration of its terminals. That is,

$$ F = b(u_1 - u_2) $$

where $u_1$ and $u_2$ are the displacement coordinates of the inerter terminals and, hereafter, a dot over a symbol denotes time differentiation. In the above equation, the constant of proportionality $b$ is the so-called inertance measured in mass units (kg). Importantly, several different inerter prototypes were devised and experimentally characterized over the past decade achieving inertance values $b$ orders of magnitude larger than the devices’ physical mass, while approximating the linear behavior in Eq. (1) within a wide frequency range of practical interest (e.g. Papageorgiou and Smith 2005, Wang et al 2011, Chuan et al 2012, Swift et al 2013, Gonzalez-Buelga et al 2016, Hu et al 2016). For example, the early and most widely-known inerter implementations incorporate rack-and-pinion or ball-screw mechanisms to transform, through gearing, the translational kinetic energy associated with the relative motion of the device terminals into rotational kinetic energy at a lightweight fast-spinning disk or “flywheel” (Smith 2002, Papageorgiou and Smith 2005). The inertance in such flywheel-based inerters depends primarily on the number of gears and on the gearing ratio used to drive the flywheel, rather than on the mass of the flywheel.
To elaborate further on this point, consider a typical mechanical realisation of the inerter comprising a flywheel linked to a rack-and-pinion via \( n \) gears. Figure 1 depicts such a device for the special case of \( n=4 \). The inertance of this device is given by

\[
b = m_f \frac{\gamma_f^2}{\gamma_{pr}^2} \left( \prod_{k=1}^{n} \frac{r_k^2}{pr_k^2} \right)
\]

where \( m_f \) and \( \gamma_f \) are the mass and the radius of gyration of the flywheel, respectively, \( \gamma_{pr} \) is the radius of the flywheel pinion, and \( r_k \) and \( pr_k \) \((k=1,2,...,n)\) are the radii of the \( k \)-th gear and its corresponding pinion, respectively, linking the rack to the flywheel pinion (see also Fig. 1). Assuming a flywheel of 10kg mass with a ratio \( \gamma_f / \gamma_{pr} = 3 \) driven by a single gear (i.e., \( n=1 \)) with a \( r_1 / pr_1 = 4 \) gear ratio, the inertance computed from Eq. (2) is \( b = 1440\)kg (see also Smith 2002). Adding two more gears with a common gear ratio equal to 3, yields an inerter with \( b = 116640\)kg, that is, a device with a physical mass three orders of magnitude smaller than its inertance. The above simple example illustrates the scalability of flywheel-based inertia through gearing. It also suggests that it is practically feasible to achieve inerter with adjustable/varying inertance without any change to their weight either in a stepped manner, by means of standard gearboxes with fixed gear ratios, or, continuously, by means of continuously varying transmission gearboxes, similar to those used in automotive engineering applications (Dhand and Pullen 2015).

![Fig. 1 Schematic representation of a rack-and-pinion flywheel-based inerter device with 4 gears.](image)

In view of Eqs. (1) and (2), it is seen that the ideal (linear) inerter can be construed as an inertial/mass amplification device whose gain depends on \( b \) and on the relative acceleration observed by its terminals. In fact, in the special case where one of the inerter terminals is “grounded” (i.e., linked to a stationary point), the inerter behaves as a “weightless” mass equal to \( b \). For instance, by setting \( \ddot{u}_1 = 0 \) in Eq. (1), the inertance \( b \) is added to the physical mass associated with the dynamic degree-of-freedom (DOF) corresponding to the displacement \( u_1 \) within a dynamical system. This inerter property was originally recognized by Smith (2002) and motivates the consideration of the so-called tuned mass-damper-inerter (TMDI) configuration (Marian and Giaralis 2014) reviewed in the following sub-section.
2.2 Equations of motion for TMDI equipped SDOF primary structures

Consider the class of dynamically excited structures amenable to be modelled as single-degree-of-freedom (SDOF) systems. The TMDI aims to suppress the motion of such systems (primary structures) by coupling the classical tuned mass-damper (TMD) with a grounded inerter in a skyhook configuration (Marian and Giaralis 2014). Specifically, the TMDI comprises a mass $m_2$ attached to the primary structure via a linear spring of stiffness $k_2$ and a viscous damper with damping coefficient $c_2$, along with an inerter device with inertance $b$ linking the attached mass to the ground as shown in Fig. 2. It is emphasized, in passing, that the TMDI is different from the various inerter-based DVAs considered by Hu and Chen (2015) and optimally designed in Hu et al. (2015) for harmonic excitation. In the latter DVAs, motivated mostly by suspension systems in vehicle engineering applications, the inerter is sandwiched in between the primary structure and the attached mass is conjunction with damper and spring elements in different layouts. Nevertheless, the TMDI considers a sky-hooked (grounded) inerter aiming to suppress vibrations in stationary (i.e., non-moving) primary structures. A practical example is the case of highway truss bridges oscillating along their longitudinal direction in which the deck is interpreted as the attached mass $m_2$ connected to the main truss of mass $m_1$ through bearings modelled via the spring $k_2$ and dashpot $c_2$ as considered by Hoang et al. (2008). In this case, the inerter can link the bridge deck to the ground at the abutments and the dynamical system of Fig.2 applies to find the optimal bearing system that would minimise the truss vibrations in the longitudinal direction of the bridge.

The equations of motion of a TMDI equipped undamped SDOF primary structure with mass $m_1$ and stiffness $k_1$ are written in matrix form as

$$
\begin{bmatrix}
 m_2 + b & 0 \\
 0 & m_1
\end{bmatrix}
\begin{bmatrix}
 \ddot{x}_2 \\
 \ddot{x}_1
\end{bmatrix} +
\begin{bmatrix}
 c_2 & -c_2 \\
 -c_2 & c_2
\end{bmatrix}
\begin{bmatrix}
 \dot{x}_2 \\
 \dot{x}_1
\end{bmatrix} +
\begin{bmatrix}
 k_2 & -k_2 \\
 -k_2 & k_1 + k_2
\end{bmatrix}
\begin{bmatrix}
 x_2 \\
 x_1
\end{bmatrix} =
\begin{bmatrix}
 F_2(t) \\
 F_1(t)
\end{bmatrix}
$$

(3)

under the assumption that the physical mass of the inerter, the damper, and the spring are negligible compared to the $m_1$ and $m_2$ masses. In the previous equations, $x_1$ and $x_2$ are the displacement response histories relative to the ground of the primary structure and of the attached mass, respectively. Furthermore, the forcing vector in the right hand size of Eq. (3) specializes as
\[
\begin{bmatrix}
F_2(t) \\
F_1(t)
\end{bmatrix} = \begin{cases}
0 & \text{or} \\
F(t)
\end{cases} \quad \text{or} \quad \begin{bmatrix}
F_2(t) \\
F_1(t)
\end{bmatrix} = -\begin{bmatrix} m_2 \\
m_1
\end{bmatrix}a_s(t) \tag{4}
\]

The first vector in Eq. (4) corresponds to a force-excited primary structure subject to a load \(F(t)\) as shown in Fig. 2(a). The second vector in Eq. (4) corresponds to a base-excited primary structure subject to the ground acceleration time-history \(a_g(t)\) as shown in Fig. 2(b).

In view of Eqs. (3) and (4), it is readily seen that for the case of force-excited primary structures, the TMDI coincides with a classical TMD with attached mass \(m_2+b\). In this regard, all known approaches and formulae for vibration control and energy harvesting for force-excited SDOF primary structures equipped with the classical TMD are applicable for the TMDI as well: one needs only to replace the attached TMD mass, \(m_2\), by the sum of the attached mass and the inertance, \(m_2+b\), as required in the various expressions derived for the classical TMD (e.g., Den Hartog 1956, Krenk 2005, Salvi and Rizzi 2016). However, this is not the case for acceleration base-excited primary structures in which the effective (inertial) force applied to the attached mass due to the ground acceleration is proportional to \(m_2\) and not to \(m_2+b\). To this end, only the case of acceleration base-excited TMDI equipped primary structures is explicitly considered in the ensuing mathematical development as the associated expressions quantifying the performance for vibration suppression and energy harvesting cannot be trivially derived by substitution to known results applicable to the classical TMD. Still, certain plots and final analytical formulae pertaining to force-excited TMDI equipped primary structures will also be presented and discussed in subsequent sections for the sake of completeness and comparison, as deemed essential.

Denote by \(\omega_{TMDI}\) and \(\zeta_{TMDI}\) the natural frequency and the critical damping ratio of the TMDI, respectively, defined as

\[
\omega_{TMDI} = \sqrt{\frac{k_2}{m_2+b}}, \quad \zeta_{TMDI} = \frac{c_2}{2(m_2+b)\omega_{TMDI}}. \tag{5}
\]

Further, consider the mass ratio \(\mu\), frequency ratio \(\nu_{TMDI}\), and inertance ratio \(\beta\) expressed as

\[
\mu = \frac{m_2}{m_1}, \quad \nu_{TMDI} = \frac{\omega_{TMDI}}{\omega_1}, \quad \text{and} \quad \beta = \frac{b}{m_1}, \tag{6}
\]

respectively, where \(\omega_1 = (k_1/m_1)^{1/2}\) is the natural frequency of the primary structure. Using the above dimensionless quantities, the complex frequency response function (FRF) in terms of the relative displacement \(x_i\) of the base-excited primary structure in Fig. 2(b) can be written as

\[
G_i(\omega) = \frac{x_i}{a_s} = \frac{(1+\mu)\omega_{TMDI}^2 - \omega^2 + i2\zeta_{TMDI}(1+\mu)\omega_{TMDI}\omega}{\left(1 - \frac{\omega^2}{\omega_1^2}\right)(\omega_{TMDI}^2 - \omega^2 + i2\zeta_{TMDI}\omega_{TMDI}\omega) - \frac{\beta + \mu}{\omega_1^2}(\omega_{TMDI}^2 + i2\zeta_{TMDI}\omega_{TMDI}\omega)\omega^2} \tag{7}
\]

in the domain of frequencies \(\omega\) by considering the normalized acceleration input \(a_s/\omega_1^2\). In the latter equation and hereafter \(i = \sqrt{-1}\). Moreover, the complex FRF for the same dynamical system in terms of the relative displacement \(x_2\) of the attached mass is written as
\[ G_2(\omega) = \frac{x_2^2}{a_i^2} = \frac{(1 + \mu)\left(\alpha_{\text{TMDI}}^2 + i2\zeta_{\text{TMDI}}\alpha_{\text{TMDI}}\omega_0\right) + \frac{\mu}{\beta + \mu}\left(\omega_i^2 - \omega^2\right)}{\left(1 - \frac{\omega^2}{\omega_i^2}\right)\left(\omega_{\text{TMDI}}^2 - \omega^2 + i2\zeta_{\text{TMDI}}\alpha_{\text{TMDI}}\omega_0\right) - \frac{\beta + \mu}{\omega_i^2}\left(\omega_{\text{TMDI}}^2 + i2\zeta_{\text{TMDI}}\alpha_{\text{TMDI}}\omega_0\right)\omega^2} \] (8)

Note that by setting \( b = \beta = 0 \) in Eqs. (7) and (8), the FRFs in terms of the relative displacements \( x_1 \) and \( x_2 \), respectively, for an undamped SDOF primary structure equipped with the classical TMD are retrieved. In this regard, the classical TMD may be viewed as a special case of the TMDI.

In the following section, optimal TMDI design for undamped harmonically excited SDOF primary structures is sought by considering the minimization of the peak value attained by the magnitude of the FRF in Eq. (7), \( |G_2(\omega)| \), hereafter referred to as the dynamic amplification factor (DAF). This is the most common design criterion adopted in the literature for vibration suppression under harmonic excitation (Krenk 2005). Further, in Section 4, the kinetic energy of the TMDI equipped SDOF primary structures available to be transformed into electric energy via a standard electromagnetic energy harvester is quantified. The latter requires the consideration of both the FRFs in Eqs. (7) and (8).

3. Optimal TMDI design and performance for vibration suppression in harmonically excited SDOF structures

3.1 Derivation of TMDI parameters in closed-form based on the fixed point theory

Assume that the TMDI equipped structure in Fig. 2(b) is subjected to a harmonic ground acceleration excitation \( \alpha_c \). Given fixed values for the \( \mu \) and \( \beta \) ratios defined in Eq. (6), it is sought to determine optimal values for the TMDI stiffness coefficient \( k_2 \) and damping coefficient \( c_2 \), or, equivalently, for the dimensionless frequency and damping ratios \( \nu_{\text{TMDI}} \) and \( \zeta_{\text{TMDI}} \) defined in Eqs. (6) and (5), respectively, such that the peak relative displacement of the primary structure is minimized. To this aim, the optimal tuning/design approach of Den Hartog (1956) is herein adopted. This approach is based on the “fixed point theory” which relies on the empirical observation that the DAF curves \( |G_i(\omega)| \) in Eq. (7) for \( b = \beta = 0 \) (i.e., for the classical TMD) and for fixed attached mass and frequency ratio pass through two specific points, the location of which is independent of the damping coefficient \( c_2 \). Importantly, this observation holds for TMDI equipped harmonically base-excited primary structures (case of \( \beta \neq 0 \)), as well. For illustration, Fig. 3 plots the DAF \( |G_i(\omega)| \) in Eq. (7) for several values of the TMDI damping ratio \( \zeta_{\text{TMDI}} \) and for fixed values \( \mu, \beta \), and \( \nu_{\text{TMDI}} \). Evidently, there exist two “stationary” points, denoted by \( P_1 \) and \( P_2 \), where the DAF curves intersect for all damping coefficient values \( c_2 \); or, equivalently, for all TMDI damping ratios \( \zeta_{\text{TMDI}} \).
Fig. 3 Relative displacement response amplitude of undamped support excited TMDI equipped SDOF primary structure with mass ratio \( \mu = 0.1 \), inertance ratio \( \beta = 0.1 \), frequency ratio \( \nu_{\text{TMDI}} = 0.5 \), and for various damping ratios \( \zeta_{\text{TMDI}} \).

The location of \( P_1 \) and \( P_2 \) points on the frequency axis can be found by considering the equation

\[
\lim_{\omega \to \omega_0} |G_1(\omega)|^2 = \lim_{\omega \to \omega_0} |G_i(\omega)|^2
\]  (9)

By collecting the real and imaginary parts in the numerator and denominator in Eq. (7), the square magnitude of the FRF \( G_1(\omega) \) can be expressed as

\[
|G_1(\omega)|^2 = \frac{A^2 + 4\zeta_{\text{TMDI}}^2 B^2}{C^2 + 4\zeta_{\text{TMDI}}^2 D^2}
\]  (10)

where

\[
A = (1 + \mu)\omega_{\text{TMDI}}^2 - \omega_0^2, \quad B = (1 + \mu)\omega_{\text{TMDI}} \omega_0,
\]

\[
C = \frac{\omega_0^4}{\omega_0^2} - \omega_0^2 \left[ 1 + \frac{\omega_{\text{TMDI}}^2}{\omega_0^2} (1 + \beta + \mu) \right] + \omega_{\text{TMDI}}^2, \quad \text{and} \quad D = \omega_0 \omega_{\text{TMDI}} [1 - \frac{\omega_0^4}{\omega_0^2} (1 + \beta + \mu)].
\]  (11)

By substituting Eq. (10) in Eq. (9) and upon some algebraic manipulation, one obtains

\[
AD = \pm BC
\]  (12)

Adopting the positive sign in Eq. (12) and making use of the expressions in Eq. (11), the trivial (static) solution \( \omega = 0 \) is reached, which is not of interest. However, by adopting the negative sign in Eq. (12) together with Eq. (11) yields the following quadratic equation in \( \omega^2 \)

\[
(2\mu + \beta + 2)\omega^4 - [\alpha_0^2 (2 + \mu) + 2\omega_{\text{TMDI}}^2 (1 + \beta + \mu)(1 + \mu)]\omega^2 + 2(1 + \mu)\alpha_0^2 \omega_{\text{TMDI}}^2 = 0
\]  (13)

The two roots, \( \omega_{p1}^2 \) and \( \omega_{p2}^2 \), of the last equation are the squared frequencies corresponding to the stationary points \( P_1 \) and \( P_2 \).

The tuning approach of Den Hartog [3] suggests that the peak response of the considered primary structure is minimized when the following two conditions hold:

(I) \( |G_i(\omega)| \) attains the same value at the points \( P_1 \) and \( P_2 \), and
(II) \(|G_1(\omega)|\) attains a local maximum at the points \(P_1\) and \(P_2\).

By enforcing condition (I) for the limiting value \(\zeta_{\text{TMDI}} \to \infty\), that is,

\[
\lim_{\zeta_{\text{TMDI}} \to \infty} |G_1(\omega_{P_1})| = \lim_{\zeta_{\text{TMDI}} \to \infty} |G_1(\omega_{P_2})|
\]

the following expression for the sum of the roots of Eq. (13) is reached

\[
\omega_{P_1}^2 + \omega_{P_2}^2 = \frac{2\omega_1^2}{1 + \beta + \mu}
\]

Further, a second expression for the sum of the roots of Eq. (13) can be readily written as

\[
\omega_{P_1}^2 + \omega_{P_2}^2 = \frac{\omega_1^2(2+\mu) + 2\omega_{\text{TMDI}}^2(1+\beta+\mu)(1+\mu)}{(2\mu+\beta+2)}
\]

This is obtained by taking the ratio of the linear coefficient over the quadratic coefficient in Eq. (13) with the negative sign. Making use of Eqs. (15) and (16) the following formula for the optimal frequency ratio in Eq. (6) is obtained in closed-form as a function of the (given) ratios \(\mu\) and \(\beta\)

\[
\zeta_{\text{TMDI}}^{\text{OPT}} = \frac{1}{1 + \beta + \mu} \sqrt{\frac{(1 + \mu)(2 - \mu) - \mu\beta}{2(1 + \mu)}}
\]

The above frequency ratio ensures that \(|G_1(\omega)|\) in Eq. (7) attains the same value at frequencies \(\omega_{P_1}\) and \(\omega_{P_2}\) for any \(\zeta_{\text{TMDI}}\) since it satisfies condition (I) through Eq. (14).

Next, condition (II) of the Den Hartog design approach is enforced by setting the first derivative of \(|G_1(\omega)|\) at the two stationary points equal to zero. That is,

\[
\left. \frac{d|G_1(\omega)|}{d\omega} \right|_{\omega=\omega_{P_1}} = \left. \frac{d|G_1(\omega)|}{d\omega} \right|_{\omega=\omega_{P_2}} = 0
\]

Application of Eq. (18) yields two different values for \(\zeta_{\text{TMDI}}\) which make the gradient of the DAF curve zero at the two stationary points. Following Brock (1946), the “optimal” TMDI parameter \(\zeta_{\text{TMDI}}^{\text{OPT}}\) is taken as the average of these two values (though other alternatives are possible Krenk (2005)), yielding

\[
\zeta_{\text{TMDI}}^{\text{OPT}} = \frac{\beta^2\mu + 6\mu(1 + \mu)^2 + \beta(1 + \mu)(6 + 7\mu)}{8(1 + \mu)(1 + \beta + \mu)[2 + \mu(1 - \beta - \mu)]}
\]

Substituting in Eq. (7) the TMDI tuning parameters in Eqs. (17) and (19), the following expression for the DAF at points \(P_1\) and \(P_2\) is reached

\[
\max_{\omega} \left| G_1(\omega) \right| = \left| G_1(\omega_{P_1}) \right| = \left| G_1(\omega_{P_2}) \right| = \sqrt{\frac{(1 + \mu)(\beta + 2\mu + 2)}{\beta + \mu}}
\]
Note that by setting $\beta = b = 0$ to Eqs. (17), (19), and (20) the closed-form expressions for optimal parameters and DAF of the classical TMD for undamped harmonic base acceleration excited SDOF systems are retrieved (Warburton 1982). In the remainder of this section, the potential of the TMDI vis-à-vis the classical TMD to achieve enhanced vibration suppression for the same attached mass and attached mass/weight reduction for the same level of vibration suppression is assessed. In doing so, pertinent plots based on the herein considered optimal design approach are provided and discussed.

### 3.2 Vibration suppression performance of TMDI vis-à-vis the classical TMD

To facilitate a comparison between the TMDI configuration of Fig. 2(b) and the classical TMD, Table 1 collects the previously derived formulae for the optimal TMDI tuning parameters and the corresponding peak DAF for undamped SDOF primary structures subjected harmonic base acceleration with the known formulae for the classical TMD ($b=0$). Furthermore, closed-form expressions for optimal tuning parameters and peak DAF for the case of TMDI-equipped force excited primary structures (Fig. 2(a)) are also included in Table 1 for the sake of completeness. In the latter case, the expressions for the TMDI are trivially derived from the known expressions of the classical TMD (also included in Table 1) with attached mass $m_2 + b$.

Table 1 Closed-form expressions for optimally tuned TMDI and classical TMD for undamped SDOF structures subjected to harmonic excitation.

|                      | Optimal frequency ratio ($\nu_{TMDI}$) | Optimal damping ratio ($\zeta_{TMDI}$) | Peak dynamic amplification factor (max $\left|G_\nu(\omega)\right|$) |
|----------------------|---------------------------------------|---------------------------------------|-----------------------------------------------------------------|
| Force excited TMD$^*$ | $\frac{1}{1+\mu}$                      | $\frac{3\mu}{8(1+\mu)}$              | $\frac{2+\mu}{\mu}$                                           |
| Force excited TMDI   | $\frac{1}{1+\beta+\mu}$               | $\frac{3(\mu+\beta)}{8(1+\mu+\beta)}$ | $\frac{2+\mu+\beta}{\mu+\beta}$                               |
| Base excited TMD$^{**}$ | $\frac{1}{1+\mu} \sqrt{\frac{2-\mu}{2}}$ | $\frac{3\mu}{8(1+\mu)(1-\mu/2)}$ | $(1+\mu)\frac{2}{\mu}$                                       |
| Base excited TMDI    | $\frac{1}{1+\beta+\mu} \sqrt{\frac{(1+\mu)(2-\mu)-\mu\beta}{2(1+\mu)}}$ | $\frac{\beta^2\mu+6\mu(1+\mu)\beta+\beta(1+\mu)(6+7\mu)}{8(1+\mu)(1+\beta+\mu)(2+\mu(1-\mu))}$ | $\frac{(1+\mu)(\beta+2\mu+2)}{\beta+\mu}$ |

$^*$ Den Hartog (1956); $^{**}$ Warburton (1982)

Further, Figs. 4(a) and 4(b) plot the optimal design parameters in Eqs. (17) and (19), respectively, for several different values of the mass ratio $\mu$ as a function of the inertance ratio $\beta$. The latter quantity takes values within a suggested interval of practical interest [0,1], with $\beta=0$ being the limiting value for which the TMDI degenerates to the classical TMD. It is observed that the optimal frequency ratio $\nu_{TMDI}$ decreases as $\beta$ increases for all values of $\mu$ considered, while it also decreases as the attached $m_2$ mass increases. Further, the optimum damping ratio $\zeta_{TMDI}$ increases monotonically with the normalized inertor constant $\beta$ for all considered values of $\mu$, while it also increases as the attached $m_2$ mass increases. The rate of change of both the TMDI
optimum parameters with $\beta$ is higher for smaller values of $\beta$ and $\mu$, while for $\mu>0.2$ the rate of change is almost constant. Similar trends are observed for the optimal parameters for the case of force-excited primary structures in Figs. 4(c) and 4(d), though a more prominent trend of saturation (i.e., decrease rate of change) is seen with $\beta$, especially for the relatively small values of mass ratio considered. A comparison between figs. 4(a) and 4(b), and figs. 4(c) and 4(d), respectively, suggests that for $\mu<0.1$ the optimal TMDI parameters are practically the same for the force-excited and the base-acceleration-excited primary structures across the considered range [0 1] of $\beta$ values, despite the differences in the derived analytical formulae in Table 1. This observation suggests that $\beta$ and $\mu$ ratios are not interchangeable in treating different types of excitations for relatively large attached mass ratios.

![Diagram showing optimum TMDI parameters](image)

Fig. 4 Optimum TMDI frequency ratio $\nu_{TMDI}$ and damping ratio $\zeta_{TMDI}$ for varying inertance ratio $\beta$ and for several mass ratio values $\mu$ for undamped SDOF primary structures

To assess the achieved level of vibration suppression by the TMDI vis-à-vis a same-weight classical TMD, Fig. 5(a) plots the DAF $|G_1(\omega)|$ for optimally designed (i.e., using the formulae in Table 1) TMDI-equipped undamped SDOF primary structure under harmonic base acceleration excitation with mass ratio $\mu=0.1$ and for different values of the inertance ratio $\beta$, including the $\beta=0$ value corresponding to the classical TMD. The frequency axis is normalized by the natural frequency of the uncontrolled primary structure $\omega_1$. It is seen that the larger the inertance of the optimally designed TMDI is, the more significant DAF reduction is achieved compared to the TMD case at the natural frequency $\omega_1$ of the primary structure as well as at the frequencies $\omega_{p1}$ and $\omega_{p2}$ of the stationary points. Note, however, that as the inertance increases, the location of the stationary points shifts to lower frequencies and the distance of the two points increases. As a result, the DAF values for relatively low excitation frequencies (i.e., lower than 70% the resonant frequency $\omega_1$) may increase with increasing inertance. Nevertheless, in practical applications,
dynamic vibration absorbers are used to suppress excessive oscillations in harmonically excited structures due to resonance and, therefore, their vibration suppression performance is normally gauged within a relatively narrow frequency band centered at the natural frequency of the uncontrolled structure. In this regard, it is observed that optimally designed TMDIs perform remarkably better than a same-weight optimally designed TMD within a substantially wide frequency (wider than $[0.8\omega_1, 1.2\omega_1]$ for the considered case of $\mu=0.1$) and, more importantly, the DAF curves become flatter across frequencies as the inertance ratio increases. The latter observation demonstrates that TMDIs with larger inertance ratios are also more robust to detuning effects and to uncertainty in the excitation frequency and/or in the primary structure properties than a same-weight TMD.

In light of the above discussion and plots in Figure 5(a), it can be intuitively argued that an increase of the inertance in the TMDI has the same positive effects as an increase of the mass ratio in the TMD (see e.g. De Angelis et al 2014), without, however, any substantial increase to the overall weight. To further elaborate on this important practical aspect, Fig. 5(b) plots DAF curves for optimally designed TMDs for different attached mass values. A comparison between Figs. 5(a) and 5(b) establishes that better vibration suppression close to resonance and increased robustness to detuning effects and uncertainty can be achieved either by increasing the attached mass (and therefore the added weight) of the classical TMD or by increasing the inertance of the TMDI (for a fixed attached mass/weight). Interestingly, for base acceleration excited primary structures (i.e., the case considered in Fig. 5) an optimally designed TMD with attached mass ratio $\mu_{TMD}$ performs worse than an optimally designed TMDI having a sum of the attached mass and inertance ratio, $\mu_{TMDI} + \beta$ equal to $\mu_{TMD}$. Nevertheless, for force excited primary structures the previous two dynamic vibration absorbers yield the same DAF curve.

Further to the above discussion, it is observed in Fig. 5(a) that the positive influence of the inerter tends to saturate with increasing inertance values. To better quantify this trend, Figure 6 plots the peak DAF (i.e., $\max \{G_{\omega}(\omega)\}$ in Table 1) for optimally designed TMDIs as a function of the inertance ratio $\beta$ and for several attached mass ratios normalized by the peak DAF for optimally designed TMDs. It is seen that the rate of reduction of the peak DAF achieved by the TMDI compared to same-weight TMDs at the stationary points (note that the location of these points varies for each structure, since $\omega_{p1}$ and $\omega_{p2}$ frequencies are functions of $\mu$ and $\beta$ as seen by Eqs. (13) and (17)), reduces as larger inertance ratio values are considered. Furthermore, it is
also deduced from Fig. 6 that for a fixed value of inertance the positive impact of the inerter is more prominent as TMDs with smaller attached mass are considered. In other words, the positive influence of increasing the attached TMDI mass saturates for larger mass ratios, as in the case of the classical TMD (see also Fig. 5(b)). The practical significance of this observation is that the inerter is more effective/beneficial for vibration suppression when it is coupled with more lightweight TMDs. Importantly, similar observations and trends on the improved level of vibration suppression achieved by the TMDI vis-à-vis the classical TMD as a function of the attached mass and inertance ratio hold for randomly base-excited primary structures (Marian and Giaralis 2014). As a final remark, the curves in Figs. 6(a) (base acceleration excitation) and 6(b) (force excitation) practically coincide even for excessively large attached mass ratio values.

3.3 Attached mass/weight reduction of TMDI vis-à-vis the classical TMD

The previous discussion quantified the improved vibration suppression capabilities of the TMDI vis-à-vis the classical TMD in a performance-assessment context. However, the TMDI bears a significant advantage over the TMD within the more practical performance-based design context: it achieves the same level of vibration suppression with significantly smaller attached mass ratios than the classical TMD and therefore with significantly reduced added weight to a given primary structure. This aspect is quantified in Fig. 7 which plots the peak DAF in a TMDI design bar-chart format. These design charts provide for the required attached mass ratio to achieve a target (i.e., pre-specified by the design engineer) peak DAF for different values of inertance including the limiting case of \( \beta=b=0 \) corresponding to the classical TMD. For illustration, suppose that it is sought to achieve a peak DAF of 4 for a particular base acceleration excited primary structure. From Fig. 7(a), it is seen that this value of DAF can be achieved by an optimally designed TMDI with 60% smaller attached mass than the one required by an optimally designed classical TMD and an inertance ratio of \( \beta=0.05 \). Further, an optimally designed TMDI with double the previous inertance (i.e., \( \beta=0.1 \)) achieves the target peak DAF of 4 with a 4.5 times smaller attached mass than the one required by the TMD yielding an overall significantly lighter dynamic vibration absorber. To further support this argument, assume that the mass of the primary structure under consideration is \( m_1 = 360t \). A flywheel-based rack-and-pinion inerter with inertance \( b=36t \) (i.e., corresponding to \( \beta=0.1 \)) can be achieved by using a flywheel with mass equal to 10kg and ratio \( \gamma_{fr}=3 \) connected to the rack by two gears (\( n=2 \) in Eq. (2)) with
transmission ratios: \( r_1/pr_1 = 5 \) and \( r_2/pr_2 = 4 \) (see also Fig. 1). Clearly, the total weight of such an inerter device is negligible compared to the achieved inerter \( b \).

4. Energy harvesting in harmonically excited TMDI equipped structures

4.1 An energy harvesting enabled TMDI

Having established the benefits of the TMDI for vibration suppression, this section explores its potential for harvesting energy from primary structure oscillations. To this aim, the linear dissipative damper of the TMDI is substituted by a linear translational electromagnetic motor (EM) shunted by a purely resistive load, as shown in Fig. 8. Compared to the standard TMD-based energy harvesters proposed in the literature for electric generation from low-frequency large-amplitude oscillations (see e.g., Tang and Zuo 2012, Gonzalez-Buelga et al 2014), the herein considered energy harvesting-enabled TMDI considers additionally a grounded inerter. This consideration enables leveraging the inertia of the attached mass, without changing the DVA total weight. In this respect, the functionality of the inerter in the proposed configuration is significantly different from the various energy harvesters found in the literature which utilize rack-and-pinion (e.g. Tang and Zuo 2012) or ball-screw mechanisms (e.g. Cassidy et al 2011, Hendijanizadeh et al 2013), similar to those used in flywheel-based inerters, to enable the use of rotational EMs by transforming the translational kinetic energy to rotational kinetic energy.
The dashpot with coefficient $c_M$ shown in the mechanical configurations of Fig. 8 is included to model the mechanical parasitic damping leading to energy losses. A standard EM comprising a moving magnet DC voice coil linear actuator is assumed (e.g. Zhu et al 2012, Gonzalez-Buelga et al 2014). The moving magnet observes the relative motion of the primary structure and of the attached mass and travels within a magnetic field of constant flux density $J$ generating a voltage $V$ expressed as

$$V = J(\dot{x}_1 - \dot{x}_2).$$

(21)

The EM resists the relative motion between the primary structure and the attached mass by developing a damping force $F_{EM}$ in the mechanical domain written as

$$F_{EM} = c_{EM}(\dot{x}_1 - \dot{x}_2),$$

(22)

where $c_{EM}$ is the electromechanical damping coefficient. The above damping force is linearly proportional to the generated electric current $I$, that is,

$$F_{EM} = JI.$$

(23)

Using Eqs. (21) to (23) in conjunction with Ohm's law $I=V/R$, which relates the electric current $I$ through a circuit with total resistance $R$ due to a voltage $V$, the electromechanical damping coefficient $c_{EM}$ is expressed as

$$c_{EM} = \frac{J^2}{(R_C + R_L)}.$$  

(24)

In the last equation, $R_C$ represents the internal “parasitic” resistance of the EM modeling the energy losses within the device, while $R_L$ is the resistive load. In deriving Eq. (24), the inductance of the EM is neglected (e.g. Zhu et al 2012). A comparison between the dynamical systems in Figs. 2 and 8 suggests that the equations of motion and the FRFs of section 2.2 are applicable to the herein considered energy harvesting enabled TMDI by setting

$$c_2 = c_{EM} + c_M.$$  

(25)

4.2 Quantification of the available energy for harvesting

In this section, the available energy to be harvested from the vibrating system of Fig. 8 is quantified by assuming that the energy harvesting enabled TMDI is optimally designed for vibration suppression under harmonic excitation as detailed in section 3.1. (Table 1). Specifically, the available power to be harvested through the resistive load $R_L$ is given by the standard relationship in the electrical domain

$$P = I^2 R_L.$$  

(26)

Using the above relationship in conjunction with Eq. (21) and Ohm’s law, the following expression for the available power to be harvested from the dynamical systems in Fig. 8 under harmonic excitation is reached
\[ P(\omega) = \frac{J^2}{(R_c + R_L)^2} \left[ G_{RV}(\omega) \right]^2 R_L. \] (27)

In the above equation, \( G_{RV} \) is the relative velocity FRF between the \( m_1 \) mass of the primary structure and the attached \( m_2 \) mass given as

\[ G_{RV}(\omega) = i\omega \frac{G_1(\omega) - G_2(\omega)}{\omega^2}, \] (28)

where the FRFs \( G_1 \) and \( G_2 \) have been defined in Eqs. (7) and (8), respectively, for base acceleration excitation. Similar expressions for \( G_1 \) and \( G_2 \) readily follow from Eqs. (3) and (4) for force excited primary structures.

Figures 9(a) and 9(c) plot the magnitude of the \( G_{RV} \) FRFs against the normalized frequency by \( \omega_1 = (k_1/m_1)^{1/2} \) for optimally designed TMDI-equipped undamped SDOF primary structures under base acceleration and force excitations, respectively, with mass ratio \( \mu = 0.1 \) and for different values of the inertance ratio \( \beta \). It is seen that the values of these FRF spectra reduce for increasing inertance ratios which achieve an overall improved level of vibration suppression (see also Figs. 5(a) and 6). However, the reduction of \[ \left| G_{RV} \right| \] is not beneficial in terms of energy harvesting as is readily seen in Eq. (27). The effect of the increased inertance ratio \( \beta \) to the energy harvesting potential of the proposed TMDI system is quantified in Figs. 9(b) and (d) plotting the magnitude of the power in Eq. (27) as a function of the normalized \( \omega/\omega_1 \) frequency. These plots have been obtained by taking \( J=11.34 \text{ N/A} \) and \( R_c = 2.96 \Omega \) which corresponds to a particular off-the-shelf
In certain practical applications, it may be desired to increase electric power generation from primary structure oscillations during times when vibration suppression requirements are relaxed. In conventional TMD-based energy harvesters, such considerations are addressed by varying the damping property of the EM (e.g., Cassidy et al 2011, Zhu et al 2012, Gonzalez-Buelga et al 2014), to achieve a desirable trade-off between energy harvesting and vibration suppression. However, in the case of the energy harvesting enabled TMD of Fig. 8 it is viable to achieve a trading between the above two objectives by varying its total apparent inertia, intuitively defined as $m_2 + b$. This can be accomplished by considering a typical flywheel-based inerter, as the one shown schematically in Fig. 1, with varying inerterance $b$ in Eq. (2) via standard transmission gearboxes to switch gearing ratios $r_i / p_i$ and/or the number of gearing stages $n$.

To illustrate the usefulness of treating the inerterance property of the energy harvesting-enabled TMD as a “degree of freedom” leveraging the trade-off between energy harvesting and vibration suppression, Fig. 10 plots DAF spectra and available power for harvesting spectra for one optimally designed TMD for vibration suppression with mass ratio $\mu = 0.1$ and inerterance ratio $\beta = 0.6$ and for several sub-optimal TMDs. The optimal TMD parameters are determined as $c_{\text{TMD} \mu = 0.1, \beta = 0.6}^{\text{OPT}} = 0.5651$ and $\zeta_{\text{TMD} \mu = 0.1, \beta = 0.6}^{\text{OPT}} = 0.4132$ using Eqs. (17) and (19), respectively. It is observed that as $\beta$ reduces, the (sub-optimal) TMD allows for more energy to be harvested across a range of excitation frequencies centered at the primary structure natural frequency $\omega_1$, at the cost of increased oscillations to the primary structure at the same range of frequencies. Therefore, by keeping constant all the TMD properties but the inerterance $b$ leverages effectively the trade-off between energy harvesting and vibration suppression. This aspect is further quantified in Fig. 11 which plots the normalized peak DAF and peak available power for harvesting for non-optimal TMDs as the inerterance ratio $\beta$ changes for four different values of the mass ratio $\mu$ and for constant optimal TMD parameters ($\beta = 0.6$) reported in Table 2. As the inerterance is reduced below $\beta = 0.6$ (i.e., departing from the optimum design point for vibration suppression performance and available energy for harvesting of the TMD-based energy harvesters (e.g., Tang and Zuo 2012, Ali and Adhikari 2013 and Gonzalez-Buelga et al 2014) is confirmed for the proposed energy harvesting enabled TMDI, as well. In passive optimally designed TMD-based energy harvesters, this trade-off depends heavily on the assumed TMD inertial property governed by the fixed mass ratio $\mu$ (e.g., Gonzalez-Buelga et al 2014). Nevertheless, the inertial property of the herein considered TMDI system, depends not only on the a priori fixed mass ratio $\mu$, but also on the inerterance ratio $\beta$. To this end, the next section explores the potential of considering passive sub-optimal TMDIs with varying inerterance to achieve increased available energy for electric power generation.

5. Enhanced energy harvesting TMDI performance through varying inerterance

In conventional TMD-based energy harvesters, such considerations are addressed by varying the damping property of the EM (e.g., Cassidy et al 2011, Zhu et al 2012, Gonzalez-Buelga et al 2014), to achieve a desirable trade-off between energy harvesting and vibration suppression. However, in the case of the energy harvesting enabled TMD of Fig. 8 it is viable to achieve a trading between the above two objectives by varying its total apparent inertia, intuitively defined as $m_2 + b$. This can be accomplished by considering a typical flywheel-based inerter, as the one shown schematically in Fig. 1, with varying inerterance $b$ in Eq. (2) via standard transmission gearboxes to switch gearing ratios $r_i / p_i$ and/or the number of gearing stages $n$.

To illustrate the usefulness of treating the inerterance property of the energy harvesting-enabled TMD as a “degree of freedom” leveraging the trade-off between energy harvesting and vibration suppression, Fig. 10 plots DAF spectra and available power for harvesting spectra for one optimally designed TMD for vibration suppression with mass ratio $\mu = 0.1$ and inerterance ratio $\beta = 0.6$ and for several sub-optimal TMDs. The optimal TMD parameters are determined as $c_{\text{TMD} \mu = 0.1, \beta = 0.6}^{\text{OPT}} = 0.5651$ and $\zeta_{\text{TMD} \mu = 0.1, \beta = 0.6}^{\text{OPT}} = 0.4132$ using Eqs. (17) and (19), respectively. It is observed that as $\beta$ reduces, the (sub-optimal) TMD allows for more energy to be harvested across a range of excitation frequencies centered at the primary structure natural frequency $\omega_1$, at the cost of increased oscillations to the primary structure at the same range of frequencies. Therefore, by keeping constant all the TMD properties but the inerterance $b$ leverages effectively the trade-off between energy harvesting and vibration suppression. This aspect is further quantified in Fig. 11 which plots the normalized peak DAF and peak available power for harvesting for non-optimal TMDs as the inerterance ratio $\beta$ changes for four different values of the mass ratio $\mu$ and for constant optimal TMD parameters ($\beta = 0.6$) reported in Table 2. As the inerterance is reduced below $\beta = 0.6$ (i.e., departing from the optimum design point for vibration
control), the available energy for harvesting increases significantly (for fixed attached mass, stiffness, and damping properties).

It is important to note that in the above presented numerical results and discussion the damping and stiffness properties of the TMDI are purposely kept constant, for the following two reasons: (i) to isolate the effect of a varying inertance to the achieved levels of vibration suppression and of available energy for harvesting, and (ii) to by-pass the need of posing any particular, and therefore non-general, optimization criterion balancing between the conflicting objectives of minimizing the oscillation amplitude of the primary structure and maximizing energy generation. Nevertheless, it is possible to vary the stiffness and/or the damping properties, as well, to achieve an overall optimal retuning of the device assembly as a whole, yet such considerations fall outside the scope of this study and are left for future work.

<table>
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<tr>
<th>Mass ratio $\mu$</th>
<th>Frequency ratio $\nu_{\text{TMDI}}^{\text{OPT}}$</th>
<th>Damping ratio $\zeta_{\text{TMDI}}^{\text{OPT}}$</th>
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6. Concluding remarks

The TMDI configuration, recently introduced by the authors for vibration suppression of stochastically base-excited structures, have been considered for vibration control and energy harvesting in harmonically excited structures. Closed-form analytical expressions for optimal TMDI parameters, stiffness and damping, given mass and inertance ratios have been derived by application of Den Hartog semi-empirical approach widely used for the design of the classical TMD to suppress the motion of harmonically excited undamped SDOF structures. Based on pertinent analytically derived results, it was shown that the TMDI is more effective from a same mass/weight classical TMD to suppress vibrations close to the natural frequency of the uncontrolled structure, while it is more robust to de-tuning effects and uncertainties in estimating the structural properties of the primary structure. This is because the TMDI exploits the mass amplification effect of a grounded inerter: the larger the inerter constant (inertance), the more reduction to the peak response displacement of the primary structure is achieved over a wider band of frequencies for the same attached mass. Moreover, it was demonstrated that the mass amplification effect of the inerter coupled with the herein derived optimum TMDI design parameters achieves significant weight reductions for a target/predefined level of vibration suppression in the context of performance-based design compared to the classical TMD. It is expected that this aspect of the TMDI can lead to simple and cost-effective robust vibration suppression in demanding practical applications enjoying many practical benefits over large-mass passive TMDs. Furthermore, the potential of simultaneous energy harvesting and vibration suppression in passive mode by means of a novel energy harvesting enabled TMDI has been explored utilizing a typical electromagnetic motor for electric energy generation. It was shown that the inertance leverages the available power to be harvested in an optimally designed TMDI for vibration suppression. This was achieved by treating the inertance as an inertial/mass related degree of freedom, not normally considered in the design of conventional TMDs for energy harvesting, assuming the availability of a flywheel-based inerter device implementation with varying inertance through mechanical gearing.

Overall, the herein furnished analytical results have quantified the benefits of coupling a grounded linear inerter device with the classical TMD for vibrations suppression, attached weight reduction, and energy harvesting in harmonically excited SDOF structures. In this respect, it is envisioned that this study will pave the way for further developments, through theoretical and experimental research, towards adaptive DVAs and energy harvesters with varying inertial/mass properties, besides stiffness and damping, yielding smart structures and structural components. Nevertheless, further research is warranted to gauge the gains of the TMDI over the classical TMD in terms of weight reduction and energy harvesting in multi-mode MDOF structures such as in wind-excited tall buildings. In such structures, the TMDI mass is attached towards the top floors via dampers and linear stiffeners, or hangers in case of pendulum-like TMD implementations and, therefore, the inerter cannot be grounded: it needs to be connected to a different floor from the one that the mass damper is attached to (Marian and Giaralis 2014, Giaralis and Petrini 2017).
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