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A Multi Tuning-Frequency Passive/On-demand Active Pendulum Tuned Mass Damper

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Abstract— **The use of tuned mass dampers (TMDs) in high-rise buildings, towers, and other tall structures for mitigating wind-induced vibration has resulted in significant improvements in serviceability of such structures. In applications where more than one mode of vibration with distinctly different natural frequencies are in need of damping, multiple passive TMDs each tuned to one of the natural frequencies are needed. In many such applications, the excessive cost, weight, and space requirement of multiple passive TMDs are not acceptable. This paper presents a novel passive pendulum tuned mass damper (PTMD) which on-demand can switch to an active PTMD. In its default passive state, the proposed device is tuned to the first mode of the structure and acts as a traditional passive PTMD. In its active state, while staying tuned to the first mode passively, the PTMD simultaneously tunes itself to multiple higher order modes adding tuned damping and providing a multi-directional vibration control.**

Index Terms: **multi-tuning frequency, passive/on-demand active, tuned mass damper, pendulum TMD, structural vibration mitigation.**

I. INTRODUCTION

As tall structures such as high-rise buildings become taller, lighter, and more slender, they become more susceptible to excessive resonant vibration during wind events [1], [2], [3]. The same is true for air traffic control towers, steel stacks in power plants, and other tall structures subject to wind-induced perturbation. Pendulum tuned mass dampers are commonly used for introducing supplemental damping to the target modes of such structures. Although by no means exhaustive, a sample of such work is presented in [4], [5], [6], [7], [8].

PTMDs are made up of a small mass (relative to the mass of the structure they are appended to) suspended by multiple parallel links (commonly flexible links using steel wire ropes). In addition to the inertia (mass) and restoring (pendulum) elements, damping devices are also built into the make-up of PTMDs to dissipate the vibration energy [4], [9].

The PTMD mass translates in the directions of X and Y. The extent of motion in the vertical, Z, direction is insignificant and irrelevant in tuned damping of tall structures. Moreover, it rotates around the vertical Z axis, i.e., rolls. The other two rotational motions, i.e., yaw and pitch are constrained by parallel suspension links. The translational natural frequency of the PTMD (in both X and Y directions) is set by the length of the pendulum and the rotational natural frequency is set by the length of the pendulum as well as the geometry of the mass.

Considering that the first translational (sway) mode of a tall structure plays the dominant role in its dynamic response, PTMDs are normally tuned to the first natural frequency of the structure [2] [10] [9] [3]. In certain applications multiple modes with distinctly different natural frequencies are in need of being damped, necessitating the use of multiple PTMDs with different dynamic characteristics, each tuned to one of the target modes.

The extent of vibration energy that a TMD dissipates as well as its frequency range (bandwidth) increase with increase in the mass ratio (the ratio of the TMD mass to the modal mass of its target mode) of the TMD and the corresponding size of the viscous dampers used in its make-up [2] [11]. The TMDs used in tall buildings have a very small mass ratio (less than 1%) and correspondingly small optimal damping ratio; note that the optimal damping ratio decreases with the mass ratio [12]. Such TMDs a) have a rather narrow bandwidth making their damping effectiveness highly sensitive to their tuning accuracy and b) experience large excursions (relative motion of the TMD mass with respect to the structure) even when the structure is subject to moderate perturbations. Due to the change, over time, in the resonant frequencies of a structure caused by deterioration, change in the operating conditions of the structure, remodeling, etc. such TMDs are normally in

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need of frequent re-tuning [2], [8]. Moreover, due to the aging of the viscous dampers and potential change in the modal mass of the structure, the damping ratio of the TMD optimized at the design stage does not necessarily remain optimal during the life of the TMD.

More recently active tuned mass dampers (ATMDs) have been used for adding tuned damping to large structures. ATMDs can add damping to multiple modes, have high damping effectiveness, self-tune itself to the changing structure's frequency, and use smaller mass blocks than an equivalent passive TMD [13].
Chang and Soong [14] and Isao [15] showed that the effectiveness of TMDs can be considerably increased by introduction of active control. The control forces calculated using an optimal scheme demonstrated that a) significant reduction in building vibration can be reached and b) reduction in TMD stroke or the mass ratio can be achieved. Linear quadratic regulation (LQR) and linear quadratic Gaussian (LQG) optimal schemes have been successfully used by Alavinasab and Moharrami [16] and Ki-Pyo [17] as the control algorithm of ATMDs. "Ref. [18]" used pole placement method in designing effective controllers for ATMDs.

For the structures with model uncertainties, ATMDs using fuzzy logic and neural network controllers have been proposed by Bijan [19] [20]. Robust control schemes such as sliding mode control has been used by [21] as the ATMD controllers.

The smaller mass of an ATMD compared to a passive TMD with equivalent effectiveness normally comes at the expense of its larger excursion. [22] showed that the excursion of ATMDs can be reduced by augmenting their control by the rate feedback of their excursion.

In this work, a novel PTMD is presented which can, on-demand, turn into an active PTMD (APTMD) with multiple, adjustable tuning frequencies and corresponding damping ratios in multiple directions. This device is designed as a passive PTMD tuned to the first natural frequency of the structure. It also has the look of a passive PTMD, but instead of using shock absorber type viscous dampers as its dissipative elements it uses hydraulic cylinders which along with flow control valves act as passive viscous dampers. When need arises, the flow control valves are automatically by-passed and servo-valves come on-line turning the hydraulic cylinders into actuators and in turn the passive PTMD into an active PTMD (APTMD). In its active mode, the control algorithm of the APTMD tunes it to more than one mode in more than one direction. The multi-tuning frequency attribute of the proposed APTMD can lower the cost, weight and space requirement associated with dampening multiple modes using multiple passive PTMDs.

A. *Passive/on demand active pendulum tuned mass damper*

Figure. 1 depicts the proposed passive/on-demand active PTMD with the mass suspended by 6 steel wire ropes. The passive/on-demand active PTMD uses 6 hydraulic cylinders in place of the viscous dampers commonly used in passive PTMDs. The hydraulic cylinders along with the moving mass form a 6-legged (hexapod) closed-chain mechanism with 6 degrees of freedom, commonly known as Stewart platform, capable of moving in any direction and orientation, generating controllable dynamic motions (Tosatti, 1998) (Luh, 1996) (Wen, 1994).

Except for the use of hydraulic cylinders in place of passive viscous dampers, the proposed PTMD resembles a passive PTMD commonly used in many modern high-rise buildings and airport control towers. In its default state, the hydraulic cylinders are configured to act as passive viscous dampers turning the device to a passive PTMD. When needed (depending on the frequency content of the structure's vibration) the control logic switches the hydraulic cylinders to active actuators and turn the device into an active PTMD with multiple tuning frequencies.

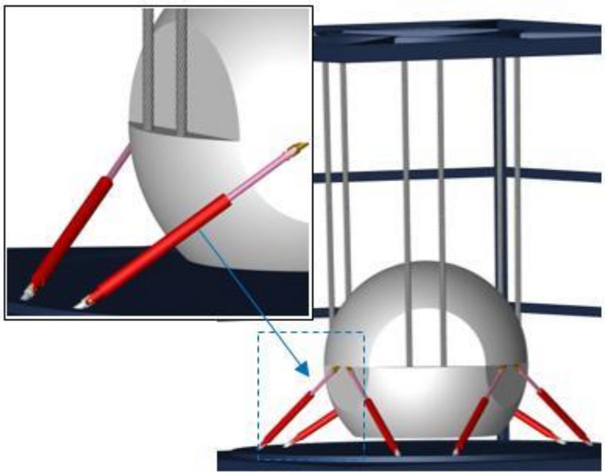

Figure 1 The passive/on-demand active PTMD

II.CONTROL STRATEGY

The block diagram presentation of the structure and the passive/on-demand active PTMD is shown in Fig. 2.

The structure is subject to the perturbation force (the term 'force' is used for force and moment) u1. As shown in the block diagram, the PTMD interacts with the structure in a feedback manner by providing a reactive force vector R, applied at the PTMD installation location, which is at the top floor, in response to the vibratory motion of the structure.

The supervisory controller shown in Figure. 2 which has continuous access to the vibration attributes of the structure decides whether to run the device as a passive or an active PTMD. When the perturbation is such that mainly the first mode of the structure is excited, the supervisory controller configures the PTMD as a passive device by disengaging the hydraulic cylinders from the hydraulic power supply and routing the hydraulic fluid in each cylinder from one side of the piston to the other side, thru its corresponding flow control, temperature control, and normally-open 3-way solenoid valves, making the cylinders to behave as viscous dampers.

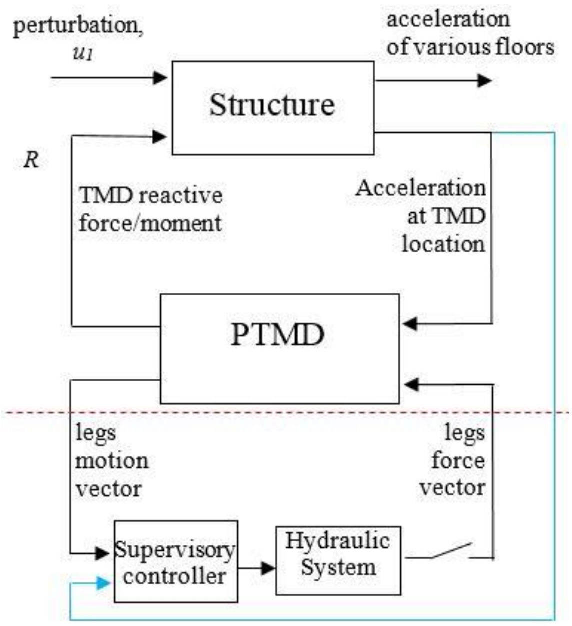

Figure 2 Block diagram of the structure + passive/on-demand active PTMD

When the perturbation becomes multi-directional and the structure vibrates in its higher order modes, the supervisory controller reverts the PTMD into its active state by bypassing the flow control valves and connecting the hydraulic cylinders to the power supply, through their corresponding servo-valves. Proper adjustment of the servo-valves by the active control scheme (made up of a combination of one centralized and six distributed controllers) enables the PTMD, in its active state, to tune itself to multiple frequencies, adding tuned damping to multiple modes, concurrently.

In the unlikely event of power loss, the PTMD reverts to its default passive state (with hydraulic cylinders acting as passive viscous dampers) tuned to the mode with the lowest natural frequency. As such, the proposed system at the minimum level of effectiveness is similar to that of a passive PTMD of the same size, tuned to the same frequency.

A. CONTROL SCHEME

With its spatial force and position control ability Stewart platform with hydraulic cylinders as its legs, is selected as the manipulation mechanism of choice for the proposed passive/on-demand active PTMD.

The strategy employed in the active control algorithm of the PTMD in its active state is the on-going adjustment of its frequency-dependent stiffness and damping. This is achieved by simultaneous actuation of the hydraulic cylinders subjecting the PTMD mass to the combined stiffness plus damping force vector shown in " (1)"

$$U_2 = KP + C\dot{P} \quad (1)$$

where U_2 is the actuation vector defined in the global Cartesian coordinate system installed on the structure, K is the required (desired) additional stiffness matrix (beyond what the pendulum provides) required for multi-frequency tuning, C is the corresponding damping coefficient matrix, and P and $\dot{P}$ are the position and velocity vectors of the center of mass of the PTMD mass measured in the global Cartesian coordinate system installed on the structure.

In the absence of any constraints, the 6-dof nature of a Stewart platform mechanism enables it to adjust its stiffness and damping in up to 6 directions (3 translational and 3 rotational). But considering that in PTMDs, a) the rotation around X and Y coordinates (pitch and yaw) are restrained and b) the motion in the vertical direction is negligible, the number of directions in which the resilience and damping of the PTMD are adjusted reduces to 3, i.e., X, Y, and γ directions.

Some tall structures, including rectangular high-rise buildings, have the directionality of their first 3 modes nearly decoupled from each other, i.e. each mode has most of its activities in one direction, only. Other structures have a rather strong directionality coupling in their first 3 modes. An APTMD considered for a structure of the former type would have three decoupled degrees of freedom, with the actuation vector U_2 shown in "(2)"

$$U_2 = \underbrace{\begin{bmatrix} K_X & 0 & 0 \\ 0 & K_Y & 0 \\ 0 & 0 & K_\gamma \end{bmatrix}}_{K} \begin{pmatrix} X_{rel} \\ Y_{rel} \\ \gamma_{rel} \end{pmatrix} + \underbrace{\begin{bmatrix} C_X & 0 & 0 \\ 0 & C_Y & 0 \\ 0 & 0 & C_\gamma \end{bmatrix}}_{C} \begin{pmatrix} \dot{X}_{rel} \\ \dot{Y}_{rel} \\ \dot{\gamma}_{rel} \end{pmatrix} \quad (2)$$

where K_i and C_i are the *diagonal* elements of the desired stiffness and damping coefficients matrices K and C in i=X, Y, and γ directions. Moreover, X_{rel}, Y_{rel}, γ_{rel} and their corresponding derivatives are the components of the relative displacement and velocity of the PTMD mass with respect to the structure in X, Y, and γ directions. An APTMD considered for a structure of the latter type would have its 3 degrees of freedom coupled with non-zero off-diagonal elements in the desired stiffness and damping coefficient matrices K, and C. In the current work, APTMDs for the structures of the former type are considered.

To meet the dimensional compatibility of the matrices in conducting matrix algebra required for the computation of the forces of the 6 legs, the 3x3 stiffness and damping coefficient matrices in "(2)" are padded with enough zeros to become 6x6 matrices, as shown in "(3)"

$$U_2 = \underbrace{\begin{bmatrix} K_X & 0 & 0 & 0 & 0 & 0 \\ 0 & K_Y & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & K\gamma \end{bmatrix}}_{K} \underbrace{\begin{pmatrix} X_{rel} \\ Y_{rel} \\ Z_{rel} \\ \alpha_{rel} \\ \beta_{rel} \\ \gamma_{rel} \end{pmatrix}}_{P} + \underbrace{\begin{bmatrix} C_X & 0 & 0 & 0 & 0 & 0 \\ 0 & C_Y & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & C\gamma \end{bmatrix}}_{C} \underbrace{\begin{pmatrix} \dot{X}_{rel} \\ \dot{Y}_{rel} \\ \dot{Z}_{rel} \\ \dot{\alpha}_{rel} \\ \dot{\beta}_{rel} \\ \dot{\gamma}_{rel} \end{pmatrix}}_{\dot{P}} \quad (3)$$

where Z_{rel} is the negligible vertical motion and α_{rel} and β_{rel} are the constrained pitch and yaw angular motions of the PTMD mass with respect to the structure.

Implementing the control scheme of "(3)" requires collocated arrangement of motion sensors and actuators in the global coordinate system which if not impossible, is rather impractical to realize. The more practical and convenient alternative is transforming the control force U2 of "(3)" from the global coordinate system to the control force defined in the local coordinate systems of the legs shown in "(4)"

$$u_2 = kp + c\dot{p} \tag{4}$$

where u_2 is the control force vector, p and $\dot{p}$ are the displacement and the velocity vectors of the legs and k and c are the stiffness and damping coefficient matrices of the legs all defined in local, leg space coordinate systems. The transformation from the global coordinate system installed on the structure to the local coordinate systems of the legs is done using a) the principle of virtual work equating the actuation virtual work in the local and global coordinate systems and b) the relationship between the motion in the local and global coordinate systems, as described below:

Let $U_2 = \left[U_{2X} \quad U_{2Y} \quad U_{2Z} \quad \cdots \quad U_{2\gamma}\right]^T$ present the vector of force and moment that the legs collectively impart on the PTMD mass in the global coordinate system and $u_2 = [u_{21} \quad u_{22} \quad u_{23} \quad \cdots \quad u_{26}]^T$ present the vector of actuated leg forces (in the local leg coordinate systems). Moreover, let $\delta p = [\delta l_1 \quad \delta l_2 \quad \delta l_3 \quad \cdots \quad \delta l_6]^T$ present the vector of virtual displacements of the legs and $\delta P = [\delta X \quad \delta Y \quad 0 \quad 0 \quad 0 \quad \delta\gamma]^T$ present the vector of virtual displacement of the PTMD mass. The zeroes in δP vector are to account for negligible translation along the Z axis and restrained rotation around the X and Y axes, of the pendulum mass.

Assuming that the friction forces in the joints and the gravitational effects of the legs are negligible and considering that the main gravitational effect due to the weight of the PTMD mass is taken up by the pendulum wire ropes (not the legs of the Stewart platform mechanism), the virtual work done by the actuated legs equals the virtual work done by the forces and moments that the legs impart on the mass, i.e.,

$$u_2^T \delta p - U_2^T \delta P = 0 \tag{5.1}$$

Taking into account the relationship between the virtual displacements δp and δP, via the 6x6 Jacobian matrix of the closed-chain mechanism J, i.e.,

$$\delta p = J \delta P \tag{5.2}$$

and substituting it in Eq. (5.1) results in

$$\left(u_2^T J - U_2^T\right)\delta P = 0$$

Since the elements of virtual displacement vector δP are independents, the coefficient of δP in the equation above has to vanish, resulting in

$$u_2 = J^{-T} U_2 \tag{5.3}$$

Combining "(3)"-"(4)" –"(5.2)" –"(5.3)" results in " (6.1)" –"(6.2)" relating the stiffness and damping coefficients in the global coordinates to those in the leg coordinates

$$k = J^{-T} K J^{-1} \tag{6.1}$$

$$c = J^{-T} C J^{-1} \tag{6.2}$$

where the superscript *-T* signifies the 'inverse of transpose'.

Having stiffness and damping coefficient matrices in the local coordinate systems of the legs (hydraulic cylinders) i.e., *k* and *c*, allows for the evaluation of the instantaneous desired force vector of the legs $u_2 = [u_{21} \quad u_{22} \quad u_{23} \quad \cdots \quad u_{26}]^T$. This is done by a) feeding the measured displacement vector of the legs to a centralized Proportional + Derivative (PD) scheme in which *k* and *c* are the corresponding proportional and derivative gain matrices and b) evaluating u_2 vector according to "(4)". The evaluated u_2 is then used as the reference input vector to 6 distributed Proportional (P) controllers generating the control signals to the servo-valves. The feedback signals to the distributed P controllers are the measured force of each leg. The control strategy is depicted in the block diagram of Figure. 3.

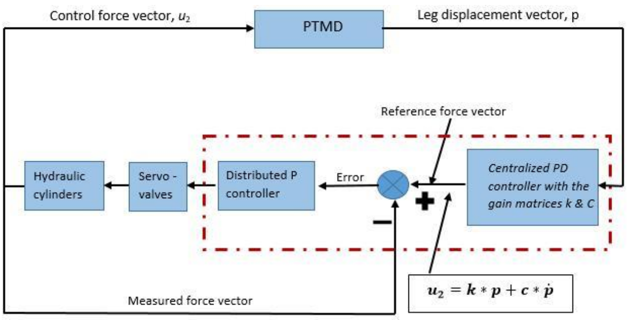

Figure 3. Block diagram of the PTMD in its active state

With the desired force vector of the legs u_2 realized, the PTMD mass experiences the desired global force vector U_2. This results in the PTMD in its active state, experiencing the desired global stiffness *K* and damping coefficient *C*. This in turn leads to the PTMD (in its active state) realizing the desired tuning frequencies along with the corresponding damping, in different directions.

III.ILLUSTRATIVE EXAMPLE

The multi-frequency tuned damping effectiveness of the proposed passive/on-demand active PTMD is numerically demonstrated by incorporating the PTMD model into the model of a tall, nearly uniform high-rise building, perturbed by wind in multiple directions. The building has 41 stories with each floor having three degrees of freedom, two translational along the *X* and *Y* axes and one rotational, denoted by γ, around the *Z* axis (perpendicular to the *X*-*Y* plane). Due to the near uniformity in geometry and mass distribution in each floor, the first 3 vibration modes of the structure are almost directionally decoupled from each other; that is, the activities of each mode is mainly in one direction, only.

Tuning the pendulum to the first mode (mainly vigorous in Y direction with the natural frequency of 0.18 Hz) results in the selection of the pendulum length. Modes 2 and 3 with vibration mainly in X and γ directions, respectively, have the natural frequencies of 0.29 and 0.56 Hz which are 60% and 300% higher than the natural frequency of mode 1. With the first three natural frequencies this far apart from each other, the only effective way of adding tuned damping to the first three modes passively would be using three PTMDs, each one targeting one of the modes. In addition to being economically unattractive such solution would have a large weight and space requirement penalty. Alternatively, a single passive-on-demand active PTMD proposed in this work can optimally be tuned to all 3 modes, simultaneously.

B. Modeling the Structure

The numerical model of the building is constructed using the first 15 modes of the structure evaluated by finite element modal analysis. The nearly directionally decoupled modes of vibration of the building enable one to associate each mode with vibration in one direction, only. For example, the 1st, 4th, 7th, 10th and 13th modes vibrate mainly in Y direction. The 2st, 5th, 8th, 11th, 14th modes vibrate mainly in X direction. And the remaining modes vibrate mainly in γ direction (rotation around Z axis).

Assuming the structure vibrates in its linear region, the state space model of the building is formulated. "(7)" presents the general form of this model

$$\dot{z} = \begin{pmatrix} 0 & I \\ -\omega_n^2 & -2\xi\omega_n \end{pmatrix} z + \begin{pmatrix} 0 \\ \psi' \end{pmatrix} u_1 \qquad \textbf{(7)}$$
$$y = [\psi \ \ 0] z + [D] u_1$$

where n is the number of modes included in the model, $\omega_n = diag([\omega_{n1} \ \omega_{n2} \ \cdots \ \omega_{nn}])$ is the diagonal natural frequency matrix, and $\xi = diag([\xi_1 \ \xi_2 \ \cdots \ \xi_n])$ is the diagonal modal damping ratio matrix. Moreover, 0 and I are the n x n zero and identity matrices, $z = [q \ \dot{q}]^T$ is 2n x 1 state vector in which $q = [q_1 \ \ q_2 \ \ \cdots \ \ q_n]'$ is the n x 1 modal displacement vector and $\dot{q}$ is the n x 1 modal velocity vector. y is the output vector of physical displacements.

Using the state space model of "(7)", the frequency response functions of the building are evaluated. Harmonic perturbations, with spatially varying amplitudes along the height of the structure (zero at the first and maximum at the top floor) mimicking the effect of vortex shedding are used to excite all the floors, simultaneously. The modal damping ratio of 1.5% is used for all the modes. Also,

$$\psi = \begin{bmatrix} \psi_{1x}(floor_1) & \psi_{2x}(floor_1) & \cdots & \psi_{nx}(floor_1) \\ \psi_{1x}(floor_2) & \psi_{2x}(floor_2) & \cdots & \psi_{nx}(floor_2) \\ \vdots & \vdots & & \vdots \\ \psi_{1x}(floor_r) & \psi_{2x}(floor_r) & \cdots & \psi_{nx}(floor_r) \\ \\ \psi_{1y}(floor_1) & \psi_{2y}(floor_1) & \cdots & \psi_{ny}(floor_1) \\ \psi_{1y}(floor_2) & \psi_{2y}(floor_2) & \cdots & \psi_{ny}(floor_2) \\ \vdots & \vdots & & \vdots \\ \psi_{1y}(floor_r) & \psi_{2y}(floor_r) & \cdots & \psi_{ny}(floor_r) \\ \\ \psi_{1\gamma}(floor_1) & \psi_{2\gamma}(floor_1) & \cdots & \psi_{n\gamma}(floor_1) \\ \psi_{1\gamma}(floor_2) & \psi_{2\gamma}(floor_2) & \cdots & \psi_{n\gamma}(floor_2) \\ \vdots & \vdots & & \vdots \\ \psi_{1\gamma}(floor_r) & \psi_{2\gamma}(floor_r) & \cdots & \psi_{n\gamma}(floor_r) \end{bmatrix} \qquad (8)$$

is the 3rxn modal matrix and r is the number of floors, i.e. 41. $u_1 = [u_{1x} \ \ u_{1y} \ \ \ M_{1\gamma}]^T$ is the 3r x 1 vector of the perturbation input in which u1x and u1y are the two rx1vectors of disturbance forces acting in the X and Y directions and M1γ is the rx1 vector of torsional disturbance moment. And lastly, D is the direct through matrix which is a 0 matrix when any of the states or their linear combination is the output.

With 15 modes included in the model of the 41 story building, the modal (eigenvector) matrix ψ, shown in "(8)", has the dimension of 123 x 15. Note that each floor has 3 planar degrees of freedom; the flexibility of each floor is ignored.

The magnitudes of the frequency response functions (FRFs) of the 41th floor acceleration in *Y*, *X*, and *γ* directions, over the frequency range of 0.1-1 Hz, are shown in Figure. 4.

As stated earlier and shown in the FRFs of Figure. 4, the modes are nearly directionally decoupled in this tall building; note that at any natural frequency the corresponding mode has most of its vibration in one direction of Y, X, or γ. In addition, the natural frequencies of the first 3 modes corresponding to vibration in Y, X, and γ directions are far enough apart (in percentage terms) from each other such that a single passive PTMD cannot effectively add damping to all three modes.

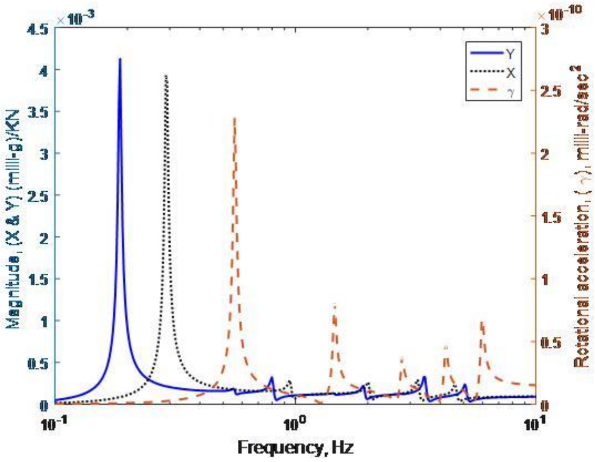

Figure 4. FRFs mapping perturbations to the structure's top floor accelerations in all 3 directions

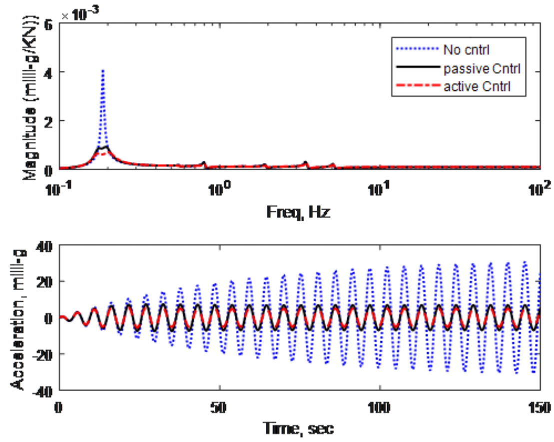

Figure 5. FRFs and resonant time traces of the structure's top floor acceleration along Y direction, without and with the TMD operating in its passive and active modes

C. Application of the Proposed Passive/On-demand Active PTMD

The proposed passive/on-demand active PTMD is synthesized to add damping to the first three modes of the aforementioned tall building, simultaneously. The pendulum length is selected so that the PTMD, with its hydraulic cylinders configured as passive viscous dampers, is optimally tuned to the lowest natural frequency of the structure targeting the first mode. The passive/on-demand active PTMD defaults to its passive state when the structure is perturbed mainly along the Y direction vibrating mostly in its first mode.

The model of the passive/on-demand active PTMD is interfaced with the model of the structure and a number of frequency and time-domain simulations are conducted.

The three traces in Figure. 5 show the frequency response functions as well as the resonant time traces of the structure's acceleration measured at the top floor along Y direction, with no tuned damping (blue/dotted-line), passive tuned damping (black/solid-line), and active tuned damping (red/center-line). Harmonic perturbation in Y direction with spatially varying amplitude along the height of the structure is used to perturb all the floors, simultaneously. The severity of the harmonic perturbations is selected corresponding to the 10 year return period wind (where the building is located) inducing 32 milli-g peak accelerations in Y direction.

Comparison of the black/solid-line traces with red/center-line traces in Fig. 5 shows that the tuned damper in its passive state is almost as effective as it is in its active state. Considering that the proposed TMD is tuned to the frequency of the first mode and designed to operate passively when the structure is mainly vibrates in that mode, the control efforts of the ATMD (TMD in active state) for quieting that mode is minimal.

The comparison of the TMD's performance, acting in passive and active modes is continued by assessing the excursions of the hydraulic cylinders' of the TMD in its two operating states. Figure. 6_a and Figure. 6_b depict the frequency response functions mapping the force perturbing the structure to the excursions for all hydraulic cylinders (legs) acting as viscous dampers (with the TMD in its passive state) and acting as actuators (with the TMD in its active state). Clear from these figures, the excursions of the hydraulic cylinders are not substantially higher when the TMD acts in its active state. This is to be expected considering that a) the proposed TMD is optimally designed as a passive device tuned to the first mode, and b) the main objective for introducing active controls into the TMD is multi-frequency/multi-directional tuning and not so much increasing its damping effectiveness.

When modes 2 and 3 of the structure are also perturbed, the hydraulic control circuits are automatically reconfigured making the cylinders act as the active elements turning the device to an active PTMD. The hydraulic cylinders, arranged similar to the 'legs' of a Stewart platform mechanism and controlled by the control scheme described earlier, introduce direction-dependent damping and additional stiffness (beyond what the length of the pendulum provides) to the passive/on-demand active PTMD. The control scheme extends the tuning of the device beyond the first mode to the second and third modes, enabling *the* passive/on-demand active PTMD to dampen the first three modes of the structure, simultaneously.

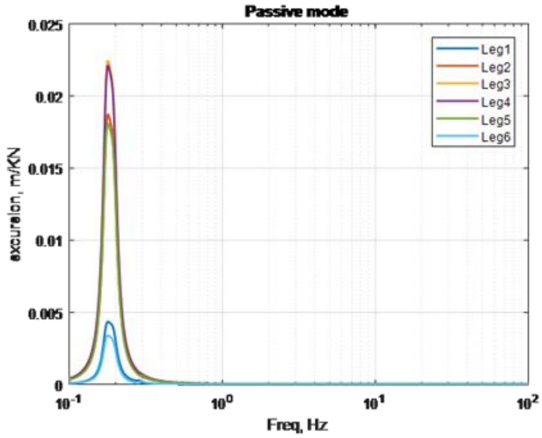

Fig. 6_a. FRFs mapping the force perturbing the structure to the excursions for all hydraulic cylinders (legs) acting a) as viscous dampers (with the TMD in its passive state)

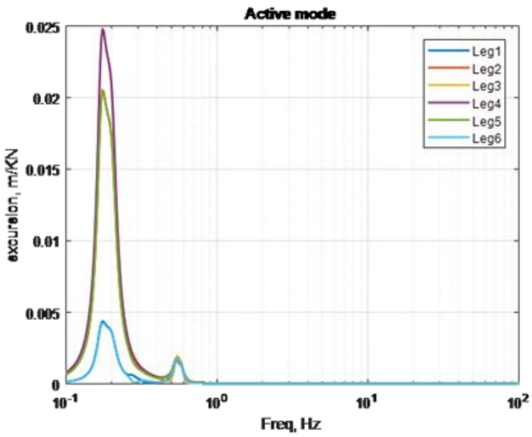

Fig. 6_b. FRFs mapping the force perturbing the structure to the excursions for all hydraulic cylinders (legs) acting as actuators (with the TMD in its active state)

As in Figure. 5, Figures. 7, 8, show the frequency response functions as well as the resonant time traces of the structure's acceleration along the X and around the Z axes, measured at the top floor. Harmonic perturbations with spatially varying amplitude along the height of the structure, are used to perturb all the floors.

Clear from Figures. 5, 6, 7, when operating in its active state, the proposed passive/on-demand active PTMD can be tuned to the first three natural frequencies of the structure, adding sizeable damping to the three corresponding modes, simultaneously. One such PTMD can effectively replace multiple passive tuned mass dampers with different tuning frequencies, in applications where more than one mode are in need of damping. Moreover, when operating in its passive state the proposed PTMD performs as robustly and as effectively as an optimally designed passive PTMD of the same size adding tuned damping to its target mode (normally the first mode) of the structure, with no need for external power.

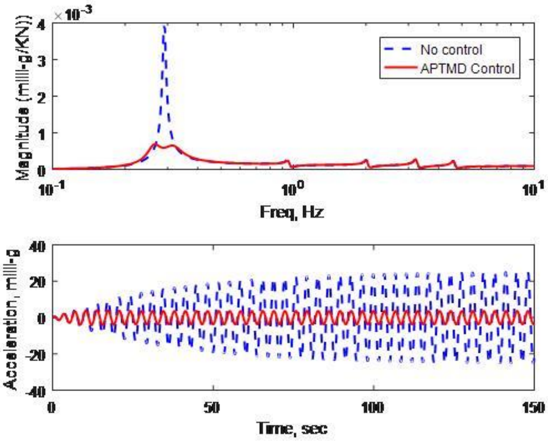

Figure 7. FRFs and resonant time traces of the structure's top floor acceleration along X direction, without and with the TMD operating in its active mode

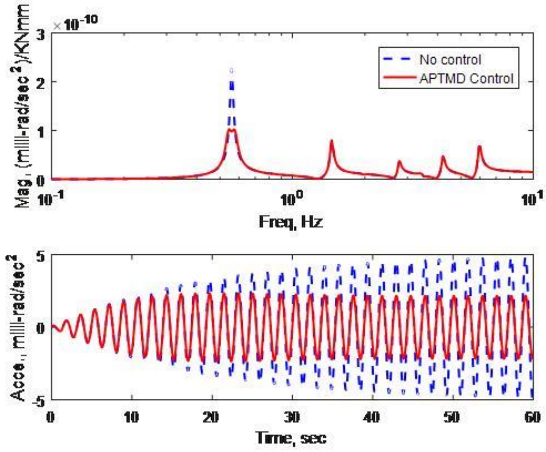

Figure 8. FRFs and resonant time traces of the structure's top floor acceleration around Z direct without and with the TMD operating in its active mode

IV.SAMMARY

A novel passive/on-demand active pendulum tuned mass damper is presented. In its default passive state, the PTMD which is tuned to the first mode of the structure acts as a traditional passive PTMD. In its active state, while staying tuned to the first mode passively, the control scheme tunes the device to higher order modes in multiple directions, simultaneously.

The effectiveness of the proposed tuned mass damper is numerically demonstrated, by interfacing its model with the model of a high-rise building. In its passive state, the PTMD adds tuned damping to the first mode of the structure to which it is passively tuned. In its active state, the device tunes itself to two additional higher order modes adding tuned damping to multiple modes of the structure, simultaneously.

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