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Numerical and Experimental Analyses of a Semi-Active Pendulum Tuned Mass Damper

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Abstract**—**Modern structures such as floor systems, pedestrian bridges and high-rise buildings have become lighter in mass and more flexible with negligible damping and thus prone to vibration. In this paper, a semi-actively controlled pendulum tuned mass dampers (PTMD) is presented that uses air springs as both the restoring (resilient) and energy dissipating (damping) elements; the tuned mass damper (TMD) uses no passive dampers. The proposed PTMD can readily be fine-tuned and re-tuned, via software, without changing any hardware. Almost all existing semi-active systems have the three elements that passive TMDs have, i.e., inertia, resilient, and dissipative elements with some adjustability built into one or two of these elements. The proposed semi-active air suspended TMD, on the other hand, is made up of only inertia and resilience elements. A notable feature of this TMD is the absence of a physical damping element in its make-up. The required viscous damping is introduced into the TMD using a semi-active control scheme residing in a micro-controller which actuates a high-speed proportional valve regulating the flow of air in and out of the air springs. In addition to introducing damping into the TMD, the semi-active control scheme adjusts the stiffness of the TMD. The focus of this work has been the synthesis and analysis of the control algorithms and strategies to vary the tuning accuracy, introduce damping into air suspended PTMD, and enable the PTMD to self-tune itself. The accelerations of the main structure and PTMD as well as the pressure in the air springs are used as the feedback signals in control strategies. Numerical simulation and experimental evaluation of the proposed tuned damping system are presented in this paper.

Keywords**—**Tuned mass damper, air spring, semi-active, vibration control.

I. Introduction

When the dynamic attributes of a structure cannot be modified either by increasing mass, stiffness, or inherent damping attributes, auxiliary damping can be introduced into the structure to abate its undesirable vibration. TMD and viscous dampers are commonly used for adding damping to structures. Being tuned devices, passive tune mass dampers (TMDs) target a particular structural mode and thus are effective only in a small frequency range. They tend to become detuned, and thus less effective, as the dynamic characteristics of structures vary with time.

A passive TMD is an energy-absorbing system consisting of an auxiliary mass, a spring, and a viscous or frictional damper. Generally, a TMD is installed at the location on the structure where amplitude of vibration, corresponding to the mode the TMD is tuned to, is the highest. It abates the vibration of the structure corresponding to its target mode by transferring the vibrational energy to the damping element of the TMD and dissipating it as heat.

Reference [1] derived a simple formula to determine the TMD parameters, i.e., damping ratio, stiffness, and tuning frequency to optimally abate the vibration of the undamped systems subject to harmonic perturbation. Since then, passive tuned mass damping has become one of the most investigated fields in controlling structural vibration. One such investigation was made by [2] who conducted theoretical and experimental study of tuned damping application using a passive TMD to reduce the vibration in two pedestrian bridges. Considering that passive TMDs are unable to adapt to changes in structural parameters, their effectiveness depends strongly on how accurately they are tuned [3].

Semi-active TMDs, also called adjustable passive TMDs, have the same configuration as passive TMDs while possessing parametric adjustability mainly on their damping and/or stiffness [4]. Semi-active TMD (SATMD) was proposed by [5]. The authors investigated the SATMD to alleviate the wind-induced vibration in high-rise buildings. They used the concept of adjustable damper by controlling the fluid flow across the piston of a hydraulic cylinder using a flow control valve. They compared the performance of SATMD with an equivalent passive TMD and demonstrated a considerable improvement with the use of SATMD.

Reference [6] designed and investigated a SATMD capable of adjusting its initial displacement as well as its damping, to protect the structure it is appended to from seismic effect. The control algorithm was derived in closed form using numerical simulations of the impulse response.

Djajakesukma et al. investigated the performance of different control algorithms to control a SATMD. These algorithms were resetting and switching controllers, linear quadratic regulator (LQR), and its modified version. The results showed that the structure design and method of perturbation had strong impact on the control performance. Also, the modified LQR was more efficient and robust than other algorithms [7].

References [8] and [9] used the ground-hook tuned mass dampers (GHTMD) to reduce human-induced vibration of a floor system with varying mass and damping. He used magnetorheological damper as the adjustable element of SATMD. He concluded that this system can perform 14% better than the passive TMD.

Reference [10] designed and implemented SATMD using a variable stiffness device. This device contained spring elements arranged on a rhombus shape and connected via pivots joints. The electromechanical actuator connected on the lower pivots

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of the rhombus shaped device adjusts the aspect ratio of the rhombus which in turn changes the stiffness of the configuration.

All semi-active systems, including the ones reviewed above, have the three elements that passive TMDs have, i.e., inertia, resilient, and dissipative elements with some adjustability built into one or two of these elements. Reference [11], on the other hand, proposed a semi-active air suspended pendulum TMD made up of only inertia and resilience elements. A notable feature of this TMD is the absence of a damping element in its make-up. The required viscous damping is introduced into the TMD using a semi-active control scheme residing in a micro-controller which actuates a high-speed proportional valve regulating the flow of air in and out of the air springs. In addition to introducing damping into the TMD, the control scheme adjusts the stiffness of the TMD.

The authors continued the work started in [11], by further analyzing and experimentally verifying the effectiveness of the semi-active air suspended TMD. The numerical aspect of the study is done in two steps: a) mathematical modeling of the individual components that make up the air suspended TMD, including the air spring, the proportional valve, and the controller and b) the assembly of the components' models to construct the model of the air suspended TMD. The experimental aspect of the study was conducted using a 220 lb TMD, with pendulum configuration (PTMD), suspended by air springs. Using this device, a series of tests were conducted to a) verify the model developed in the numerical study aspect of the research, and b) demonstrate the self-tuning and effectiveness-enhancement aspects of the semi-active air suspended TMD.

II. Air Suspended PTMD Configurations

The air suspended PTMD can be configured either as horizontal or vertical pendulum, depending on the application; the horizontal configuration is shown in Fig. 1. The pendulum hinged at the end and supported by an air spring has a discrete mass of 220 pounds. A high-speed proportional valve is used to a) maintain the desired posture of the PTMD and b) introduce active damping into it. The sensors used in the system are one or two pressure sensor(s) and an accelerometer.

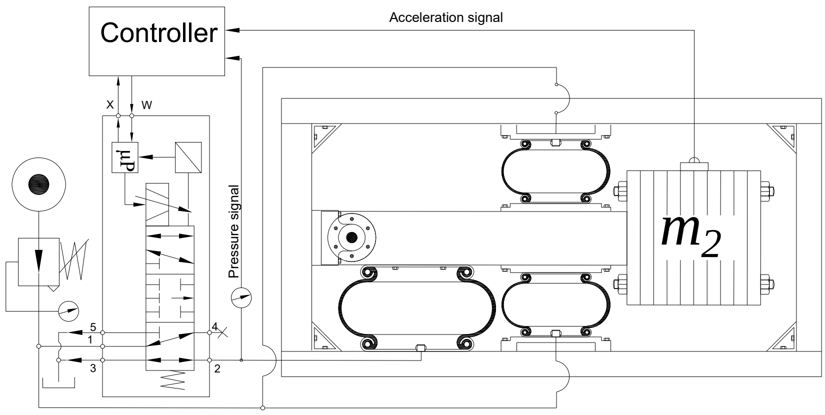

Fig. 1 Schematic diagram of PTMD using three air springs

The control strategy used in this study has two feedback loops: a) PTMD mass acceleration feedback inner loop and b) air springs pressure feedback outer loop. The pressure feedback is to maintain the static posture of the PTMD and the acceleration feedback is to introduce active damping into the air springs.

III. Numerical Modeling and Experimental Studies

The individual components making up the semi-actively controlled air suspended PTMD with pendulum configuration were modeled first. Following the verification of the component models, the interaction between them were established resulting in the system model of the air suspended pendulum TMD.

Prior to introducing the air suspended PTMD into the numerical model of a structure to assess its tuned damping effectiveness, the TMD itself has gone thru an extensive numerical and experimental evaluation.

Control Strategy: Adjusting the Flow to and from the Auxiliary Air Springs

In this set up, the auxiliary air springs (the two small air springs) are connected to the proportional pneumatic valve, which adjusts the air flow to and from the air springs, while the static pressure of the main air spring is maintained constant at 84 psi to have the PTMD hold its horizontal posture. The static pressure of the auxiliary air springs is varied in the steps of 20 psi from 0 to 80 psi.

In each run, the frequency response function (FRF) corresponding to that run is generated by perturbing the PTMD using an instrumented hammer and measuring the impact force with the impact force using the 1 mv/lb force sensor built into the hammer as well as the vibratory response of the PTMD measured with a capacitive accelerometer with the calibration factor of 1000 mV/g placed on the pendulum link. Both force and acceleration signals are fed into a dynamic signal analyzer to evaluate the FRF mapping the force to acceleration.

The two time traces in Fig. 2 show the PTMD's impulse response, with the damping control off and on. Clear from this figure, the vibration in the PTMD with the controller 'off' (gray line) lingers for many cycles, but with the controller 'on' (blue line), it stops almost instantly. The measured acceleration of Fig. 2 is used in estimating the PTMD dynamic parameters of damping ratio, stiffness, damping coefficient and natural frequency.

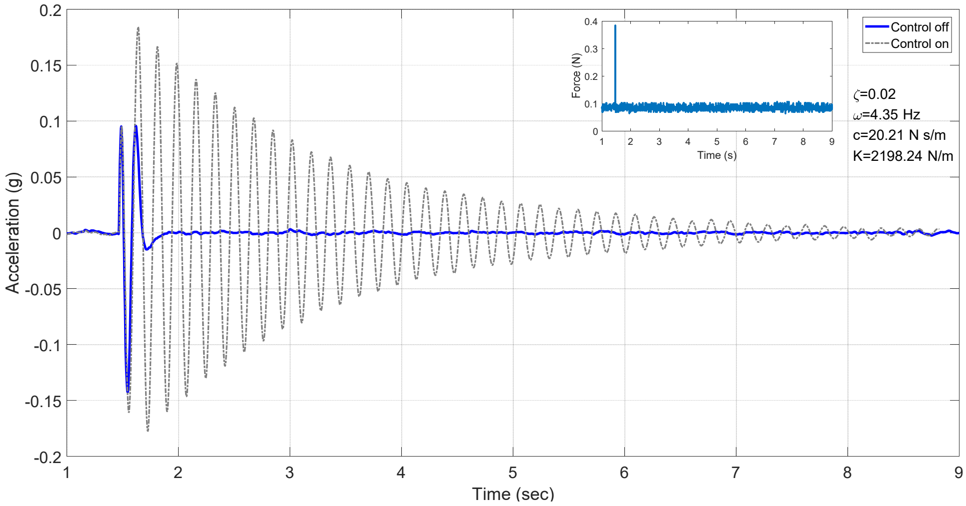

Fig. 2 Acceleration of the PTMD with the control off and on, in response to an impulse perturbation

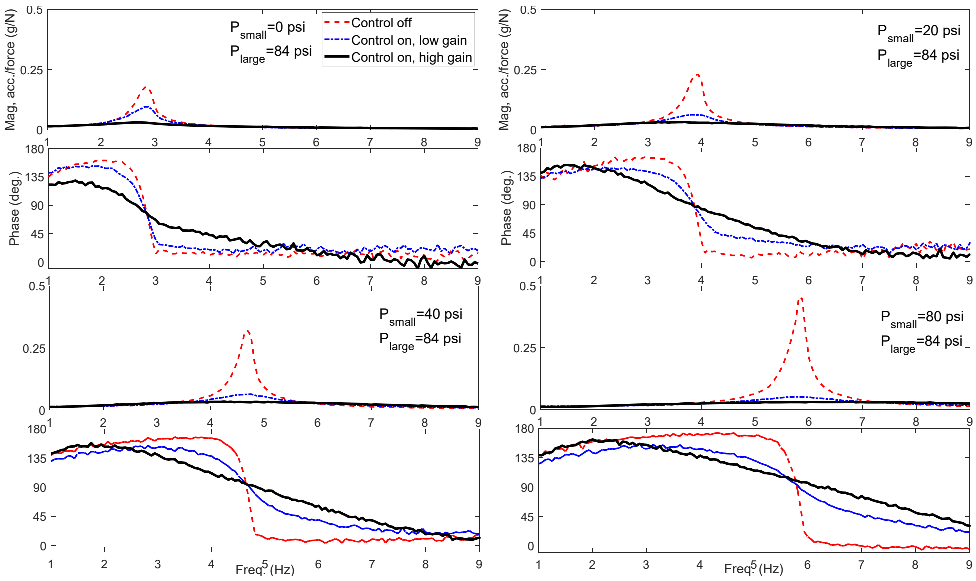

Fig. 3 FRFs of the PTMD with controller off and on with the small air springs at various static pressures

As stated earlier, in a number of impact tests, at various static pressures in the auxiliary air springs, the FRFs of the PTMD are evaluated and presented in Fig. 3. The dashed-line (red) traces present the FRFs with the controller off and the other two traces present the FRFs with the controller on with two different control gains.

Model Verification

The model of the PTMD is verified by comparing the numerically evaluated FRFs presented in Fig. 4 to the experimentally measured FRFs of Fig. 3. The close resemblance of the two sets of FRFs, points to the fidelity of the numerical model.

Examining the experimentally evaluated FRFs of Fig. 3 and the numerically evaluated FRFs of Fig. 4 the following observations are made:

1. With the controller off, the capacity of the PTMD in varying its tuning frequency is studied with different static pressures in the small air springs. The results (red dashed lines) clearly show that increasing the static pressure of the small air springs increases the overall stiffness and thus the resonant frequency of the PTMD. As expected, the increase in the stiffness also results in a reduction in inherent damping in the system; this is evident from the increase in height and sharpness of the peaks of the FRFs as the

stiffness increases.

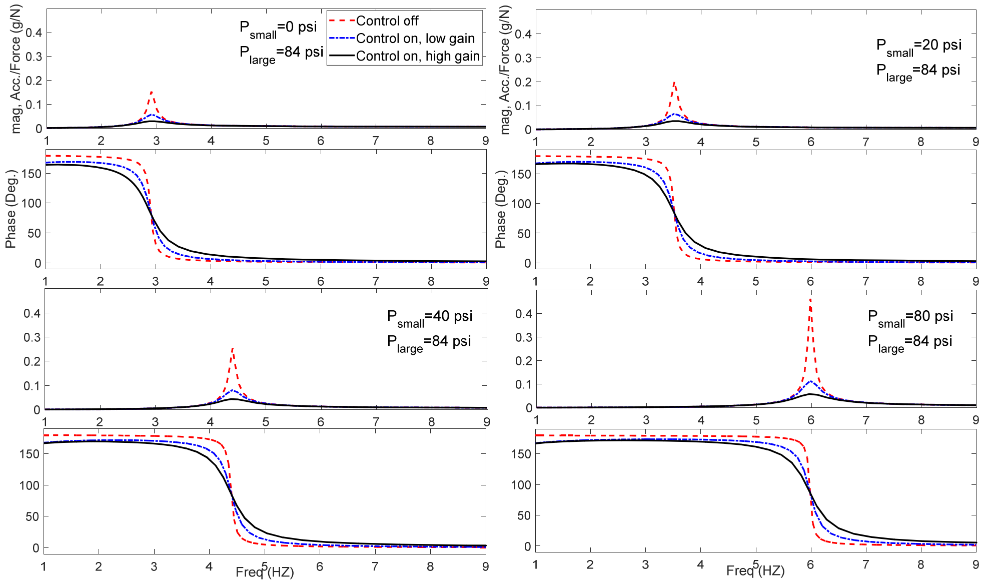

Fig. 4 Simulation FRF with/without controller using two small air spring and one large air spring is passive

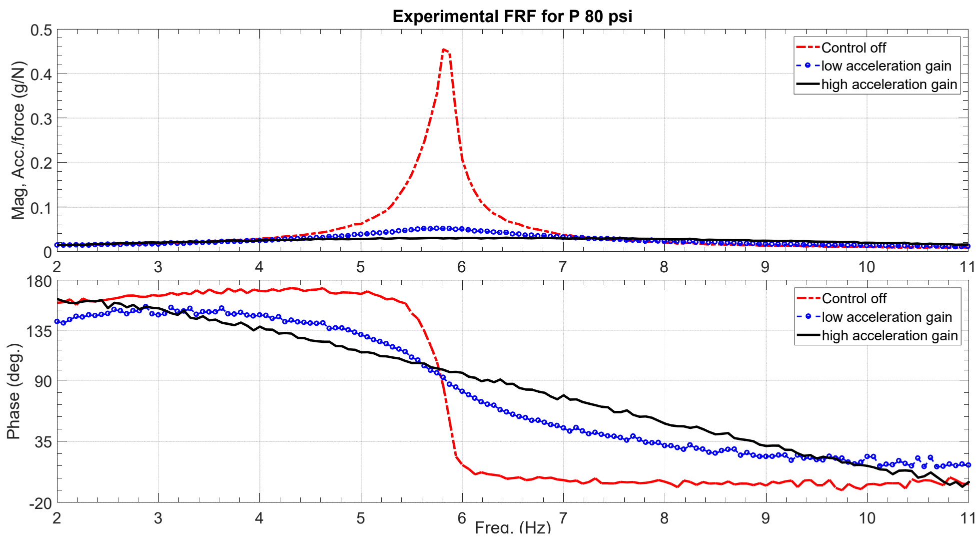

Fig. 5 FRFs with/without controller adjusting the two small air springs at P_{small} 80 psi with the large air spring passive

2. With the controller on, the same experiments are repeated. The FRFs corresponding to these experiments are presented by the blue and magenta lines of Figs. 3 and 4. The reduction in peak magnitudes of the FRFs as well as the sharpness of the peaks of the FRFs with the controller 'on' compared to the corresponding FRFs with the controller 'off' point to the introduction of a substantial amount of damping by the controller to the PTMD. A similar conclusion can also be made by comparing the phase of the two sets of FRFs. The gradual 180 degree drop in phase angle of the FRFs of the system with the controller 'on' compared to the sharp 180 degree drop in the phase of the corresponding FRFs of the system with the controller 'off' points to the effectiveness of the controller in adding damping to the PTMD.

Placing the aforementioned FRFs as subplots with multiple ordinates would have done more justice in judging the effectiveness of the controller in adding damping to the PTMD, but at the expense of having too many traces on one plot and thus a very busy and confusing one. To avoid the clutter but still benefit from such direct and distinctive comparison, only one set of FRFs (corresponding to the nominal pressure 80 Psi) with the controller 'off', 'on with low gain', and 'on with high gain' are placed on the same coordinates and presented in Fig. 4. Evident from the flattening of the peak of the magnitude plots of Fig. 4: a) the controller adds damping to the PTMD and b) the extent of the added damping can be varied by changing the gain on the controller. As stated earlier, the same judgment on damping effectiveness of the controller can be made by observing how gradual the 180 degree phase shift in the phase plots becomes when the controller is turned on.

Fig. 6 shows the simulated results and reveals good agreements with the experimental data.

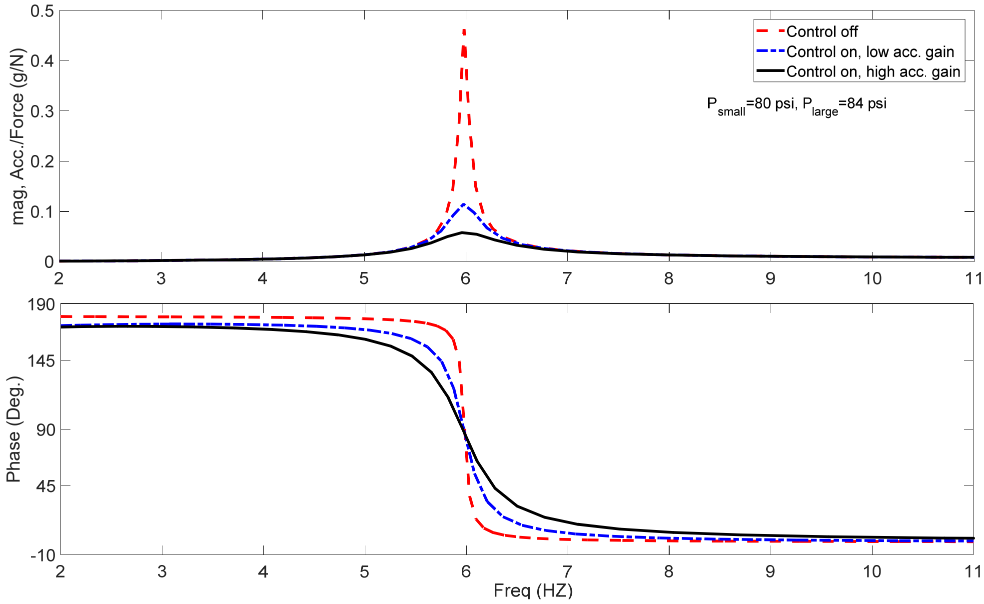

Fig. 6 Simulation FRFs with/without controller adjusting the two small air springs at Psmall 80 psi

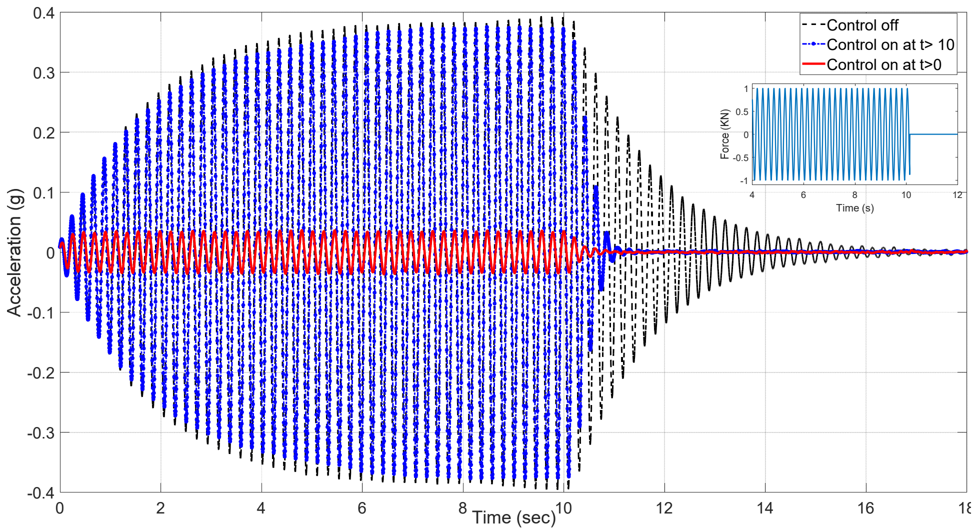

Fig. 7 Resonant responses of the uncontrolled and controlled PTMD

Time Domain Sinusoidal Response

An alternative way to evaluate the damping effectiveness of the controllers is to resonate the uncontrolled PTMD using a sinusoidal perturbation and observe the decay of its free vibration with the perturbation off. In such evaluation, the perturbation at the resonant frequency of the uncontrolled PTMD stays on until the study state response is reached. At that point in time, the perturbation is turned off and simultaneously the controller is turned on. The two black and blue traces in Fig. 7 compare the free vibration of the PTMD with the controller gains of zero (in other words ‘uncontrolled’) and non-zero (‘controlled’). With zero controller gain (‘uncontrolled’) the free vibration of the PTMD decays with a long settling time but with the non-zero controlled gain PTMD settles quickly (within two cycles of oscillation).

In another experiment, the damping effectiveness of the controller is evaluated by having the controller on during the entire durations of resonant build up as well as decay of resonant vibration; see the red trace in Fig. 7. Comparison of the red and black traces show that a) the amplitude of resonant vibration of the controlled PTMD, in response to the same perturbation is many times smaller than that of the uncontrolled vibration and b) the resonant vibration of the controlled PTMD reaches its steady state in a short period of time (and over less than 2 cycles of oscillation) as opposed to the buildup of the resonant vibration of the uncontrolled PTMD which takes a long time (and over many cycles of oscillation) to reach steady state. Both of these two observations which point to the damping effectiveness are valid for the two control strategies 1 and 2.

Tuned Damping Effectiveness of Air Suspended PTMD

Tuned damping effectiveness of the semi-active air suspended PTMD is numerically demonstrated by coupling the model of the PTMD with the model of a one degree of freedom structure.

Fig. 8 depicts the block diagram model of the entire system in which the model of each component is placed inside a subsystem block.

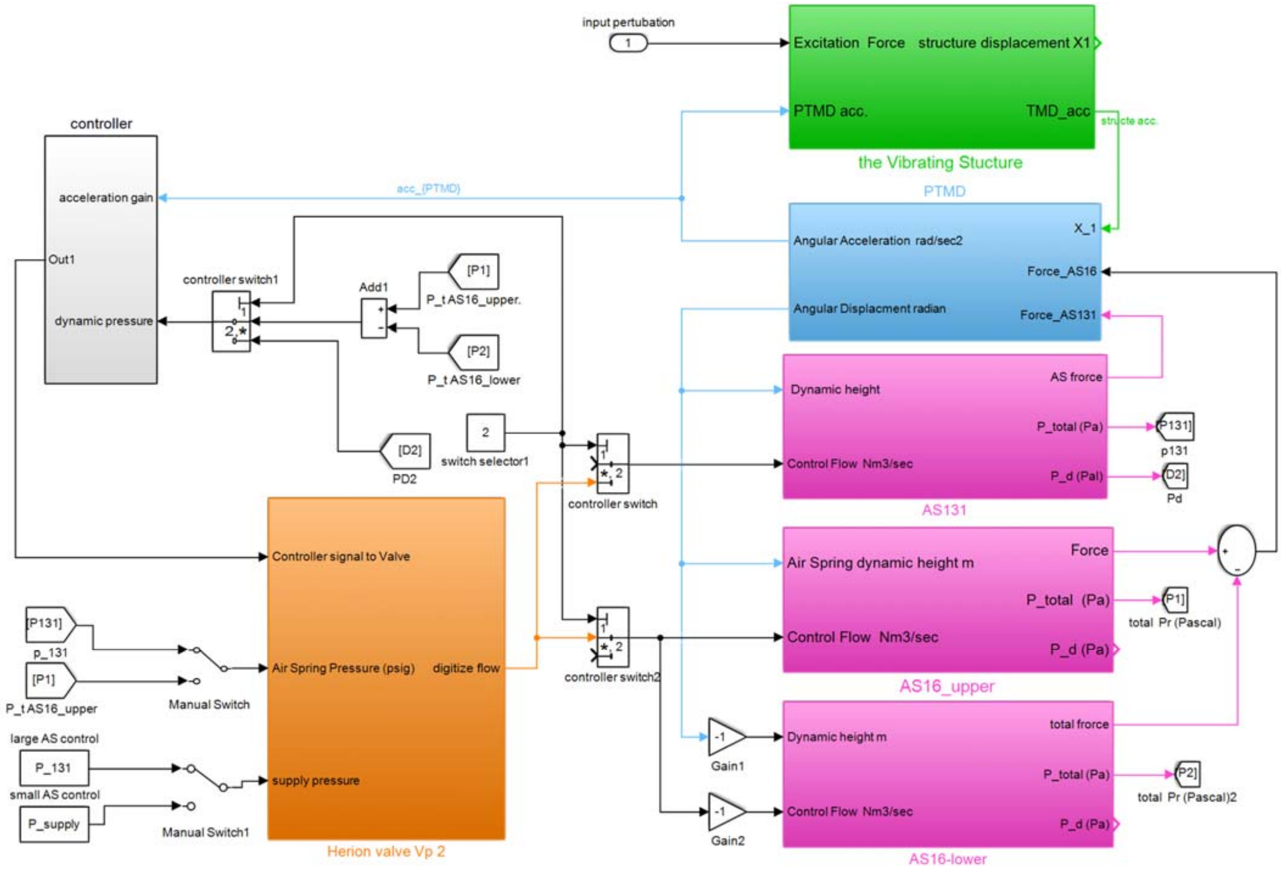

Fig. 8 Combined PTMD and structure model

The tuning process is accomplished by changing the natural frequency of the PTMD via changing the nominal pressure in the two small air springs; the higher air spring pressure the higher its stiffness and thus the higher the natural frequency of the PTMD.

Fig. 9 shows the FRFs of the structure with (blue and black lines) and without (red line) PTMD with different air spring nominal pressure settings. In this figure, the natural frequency of the main structure is changed and the PTMD is tuned by varying the nominal pressure to match the new natural frequency of the structure. Looking at the peaks of the plots, it is obvious that the blue and the black lines (with PTMD) show significant damping compared to the red line (without PTMD).

IV. Summary and Conclusions

An air-suspended PTMD configurable in both horizontal and vertical orientations is presented. Introduction of semi-active control to the PTMD enables it to adapt itself to the potential changes in the primary structure's parameters and maintain its tuning by varying its stiffness as well as its damping. Adjusting the flow of air in and out of the air springs by a high-speed proportional flow control valve varies the damping of the SATMD. Moreover, change in the pressure of the air springs varies its stiffness and thus its tuning frequency. The on-site tuning and re-tuning (if needed) of the proposed SATMD, without any change in the hardware, result in accurate tuning of the PTMD even in face of variation in the parameters of the target structure.

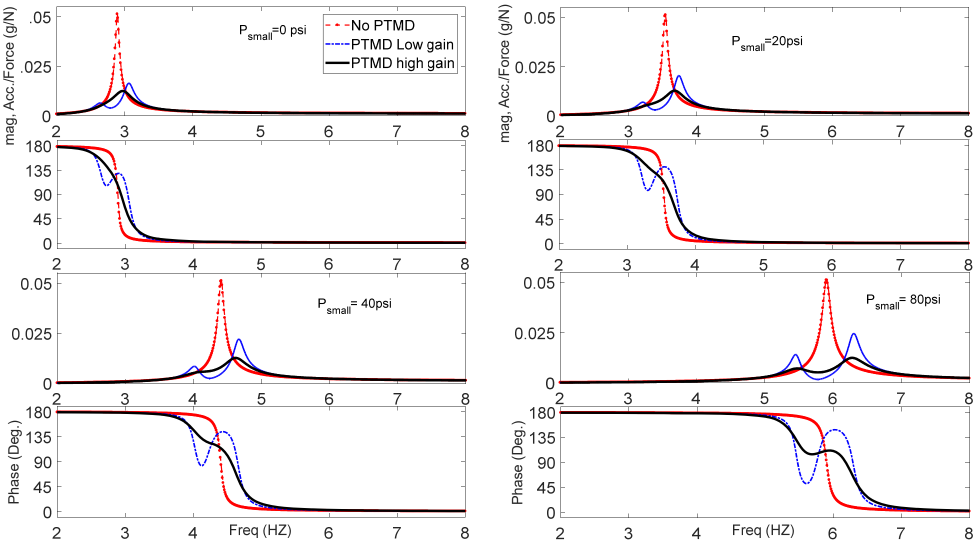

Fig. 9 FRFs of main structure w/wo PTMD

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