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Passive and active mass damper control of the response of tall buildings to wind gustiness

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Abstract

The performance of passive (TMD), active (AMD) and hybrid (ATMD) mass dampers for the reduction of the buffeting response of tall buildings is investigated. First a simple 1 + 1 DoF system is considered to model the main structure provided with a mass damper, and the wind buffeting force is simplified into a white noise excitation. A linear quadratic regulator (LQR) control law is used, and in the case of the ATMD, the results are compared to those obtained with a closed form design procedure from the literature. Second a 64-storey building is considered, and modelled accounting for its first four longitudinal modes. In the latter case a more realistic buffeting excitation is considered, accounting for the frequency distribution of the atmospheric turbulence, and for its vertical correlation. It is pointed out how the performance of each device is strongly related to the response parameter to be mitigated, and how simplified 1 + 1 DoF models can inaccurately estimate the system response, and therefore the control performance.

Keywords: Tall buildings; Structural control; Mass dampers; Wind excitation

1. Active vs. passive control of wind excited structures

Whether passive or active control is the most effective way to mitigate the response of slender civil structures exposed to the wind action is still an open question. Many papers have appeared in recent years dealing with the performance of particular active control strategies in terms of response reduction, but very seldom is their feasibility discussed. Therefore it is reasonable to question whether an active control strategy, whose performance is theoretically demonstrated, is really the optimum solution to a design problem. On the other hand, the many implementations recently published, especially in Japan [1,2], give evidence that there are circumstances in which active control performs better than passive control.

The performance that can be obtained from a particular control strategy is related to the characteristics of the excitation, to the response parameters to be controlled and to the specific design constraints. Generally speaking, for wind-excited structures the total response is the sum of a static component, a background dynamic component and resonant dynamic components in the relevant modes of vibration. The static response can be reduced by either increasing the structural stiffness, or by reducing the mean exciting force, through an improvement of the structure aerodynamics. Both solutions are clearly of a passive nature. Due to its quasi-static nature, the reduction of the background dynamic response can also be achieved increasing the stiffness or improving the aerodynamics, but no success is obtained modifying the structure dynamics (i.e. by changing mode shapes, natural frequencies and damping characteristics). Though theoretically possible, the reduction of the static and background response through active control is practically unfeasible, as the control forces would have to directly counteract the external action, and, therefore, their magnitude would need to be extraordinarily high. Conversely, even though an increase in stiffness and an improvement in aerodynamics are both beneficial, the

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reduction of the resonant response can be more conveniently achieved by modifying the structure dynamics. This can be done using either passive or active devices.

Among structural control strategies are mass dampers (MDs), in their passive, active and hybrid versions. The passive MD is the tuned mass damper (TMD), in which the auxiliary mass is connected to the main structure through a spring and a damping device. The active version of the MD is the active mass driver (AMD), in which the mass is connected to the structure through an actuator. Finally, the hybrid MD is an active tuned mass damper (ATMD), in which the auxiliary mass is connected to the structure through an actuator, a spring and, eventually, a damping device.

The success of a given control strategy strongly depends on the response parameters to be mitigated. For example, the alongwind tip displacement of a tall building can be reduced with the addition of a TMD tuned to the first natural frequency of the structure. This brings a relevant reduction of the resonant response in the first mode, but does not alter the static and background response, nor the resonant response in the higher modes. If the performance of the device is measured in terms of the reduction of the peak displacement, an intrinsic limit exists due to the presence of non-controllable components of the response. On the other hand, if the control aims at reducing the tip acceleration, the relative weight of the resonant response tends to increase due to the absence of a mean acceleration, and because the background component is of a lower magnitude. However, contribution of the higher modes is larger, and therefore more than one TMD may be necessary, in order to control more than one mode of vibration. In the latter case active control can prove to be advantageous, as one single device can control more than one mode of vibration.

In the following sections the performance of TMDs, AMDs and ATMDs for the mitigation of the buffeting response of tall buildings will be compared. Criteria will be set for the design of the active control law, in the framework of LQR control strategy [3,4].

2. Effectiveness of passive and active mass damper control: 1 + 1 DoF models

A simple approach to the design of MDs is that based on a 1 + 1 DoF model of the system (Fig. 1). The building is modelled as a SDoF system corresponding to the mode to be controlled, with m_1, $k_1 = m_1\omega_1^2$ and ξ_1 being the modal mass, stiffness, and damping ratio, respectively, and ω_1 being the associated angular frequency.

When the building is provided with a passive device (Fig. 1a), the mass $m_2 = \mu m_1$ (μ being the mass ratio) of the damper is chosen based on practical requirements, and the damper frequency $\omega_2 = \Omega\omega_1$ (Ω being the tuning ratio) and damping ratio $\xi_2 = c_2/2\omega_2 m_2$ are chosen such to minimise some response parameter. The equations of motion for the 1 + 1 DoF system of Fig. 1a are:

$$\begin{aligned} &m_1\ddot{x}_1 + (c_1 + c_2)\dot{x}_1 + (k_1 + k_2)x_1 = f(t) + c_2\dot{x}_2 + k_2x_2 \\ &m_2\ddot{x}_2 + c_2\dot{x}_2 - c_1\dot{x}_1 + k_2x_2 - k_1x_1 = 0 \end{aligned} \tag{1}$$

When the external excitation is modelled as a stationary Gaussian white noise, closed form expression for the TMD optimum parameters can be found. Neglecting the damping of the main structure, McNamara [5] and Luft [6] define the optimum TMD parameters as those maximising the equivalent damping ξ_e, (i.e. the damping to be provided to the uncontrolled structure in order to make it experience the same RMS displacement as the controlled structure). Again neglecting the damping of the main structure, Warburton [7] found the TMD parameters that minimise the RMS of main structure displacement or acceleration. Literature optimisation criteria do not account for the device stroke, and in many practical cases it is necessary to overdamp the device in order to limit the auxiliary mass displacements. In any case, once the size of the auxiliary mass is chosen, the performance of the TMD is bound by its passive nature, and its design to the choice of the spring and dashpot constants.

Better performances can be obtained using active devices, which allow arbitrary forces to be applied to the structure. The damping capacity of an AMD is theoretically unbounded, but it is obtained at the expense of the actuation force and power, and of the device stroke. The latter quantities can easily increase beyond technical ranges, making the design unpractical, and also for active devices a trade-off solution between performance and design constrains has to be found.

The equations of motion for the structure provided with an AMD (Fig. 1b) are:

$$\begin{aligned} &m_1\ddot{x}_1(t) + c_1\dot{x}_1(t) + k_1x_1(t) = f(t) - u(t) \\ &m_2\ddot{x}_2(t) = u(t) \end{aligned} \tag{2}$$

or, in a state space formulation:

$$\begin{aligned} &\dot{\mathbf{x}}(t) = \mathbf{A}\mathbf{x}(t) + \mathbf{B}_u u(t) + \mathbf{B}_f f(t) \\ &\mathbf{y}(t) = \mathbf{C}\mathbf{x}(t) + \mathbf{D}_{yu} u(t) + \mathbf{D}_{yf} f(t) \end{aligned} \tag{3}$$

where $\mathbf{x} = [x_1\ x_2\ \dot{x}_1\ \dot{x}_2]^{\mathbf{T}}$ and $\mathbf{y} = [x_1\ x_2\ \dot{x}_1\ \dot{x}_2\ \ddot{x}_1\ \ddot{x}_2]^{\mathbf{T}}$ are the state and response vectors, $u(t)$ is the control force, and where:

$$\mathbf{A} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -k_1/m_1 & 0 & -c_1/m_1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

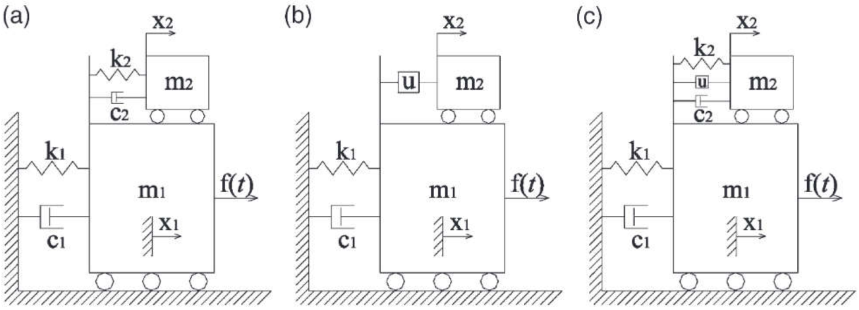

Fig. 1. Schematic of the 1 + 1 DoF system with (a) TMD, (b) AMD, and (c) ATMD.

$$\mathbf{B}_u = \left[0 \;\; 0 \;\; -\frac{1}{m_1} \;\; \frac{1}{m_2}\right]^{\mathrm{T}} \quad \mathbf{B}_f = \left[0 \;\; 0 \;\; \frac{1}{m_1} \;\; 0\right]^{\mathrm{T}} \tag{4}$$

$$\mathbf{C} = \begin{bmatrix} \mathbf{I}_{4\cdot 4} \\ [\mathbf{O}_{2\cdot 2} \;\; \mathbf{I}_{2\cdot 2}]\cdot\mathbf{A} \end{bmatrix}$$

$$\mathbf{D}_{yu} = \begin{bmatrix} \mathbf{O}_{4\cdot 1} \\ [\mathbf{O}_{2\cdot 2} \;\; \mathbf{I}_{2\cdot 2}]\cdot\mathbf{B}_u \end{bmatrix} \qquad \mathbf{D}_{yf} = \begin{bmatrix} \mathbf{O}_{4\cdot 1} \\ [\mathbf{O}_{2\cdot 2} \;\; \mathbf{I}_{2\cdot 2}]\cdot\mathbf{B}_f \end{bmatrix}$$

Different criteria can be used to choose the feedback control law. If a state feedback approach is adopted, the control force is set to $u(t) = -\mathbf{K}\mathbf{x}(t)$, $\mathbf{K} = [\bar{k}_1 \; \bar{k}_2 \; \bar{k}_3 \; \bar{k}_4]$ being a $1 \cdot 4$ gain vector. Closed form expressions of the optimum gains could be found, following the approach used by Ayorinde and Warburton [8] for TMDs, by minimising the RMS response of the main structure. When applied to AMDs, however, this procedure brings a set of four non-linear algebraic equations, whose analytical solution is not easy. In real life a solution is found through the implementation of an optimal control law. In particular, linear quadratic regulator (LQR) control proved to be effective in many cases [9], which involves finding an appropriate controller minimising the cost functional:

$$J = \int_0^\infty (\mathbf{x}^{\mathrm{T}}\mathbf{Q}\mathbf{x} + Ru^2)\mathrm{d}t = \int_0^\infty (x_1^2(t)q_1 + x_2^2(t)q_2 + \dot{x}_1^2(t)q_3 + \dot{x}_2^2(t)q_4 + u^2(t)r)\mathrm{d}t \tag{5}$$

where $\mathbf{Q} = diag[q_1 \; q_2 \; q_3 \; q_4]$ and R are the time invariant weights of the state and of the control force, respectively. The optimal control law is then expressed as:

$$u(t) = -R^{-1}\mathbf{B}_u^{\mathrm{T}}\mathbf{P}\mathbf{x}(t) = -\mathbf{K}\mathbf{x}(t) \tag{6}$$

in which $\mathbf{P}$ is a solution of the algebraic Riccati equation (ARE):

$$\mathbf{A}^{\mathrm{T}}\mathbf{P} + \mathbf{P}\mathbf{A} - \mathbf{P}\mathbf{B}_u R^{-1}\mathbf{B}_u^{\mathrm{T}}\mathbf{P} + \mathbf{Q} = \mathbf{0}. \tag{7}$$

A third possibility is the use of a hybrid device (Fig. 1c). The equations of motion for the structure provided with an ATMD are:

$$\begin{aligned} &m_1\ddot{x}_1 + (c_1 + c_2)\dot{x}_1 + (k_1 + k_2)x_1 = f(t) - u(t) \\ &\quad + c_2\dot{x}_2 + k_2x_2 \\ &m_2\ddot{x}_2 + c_2\dot{x}_2 - c_1\dot{x}_1 + k_2x_2 - k_1x_1 = u(t). \end{aligned} \tag{8}$$

In the state space Eq. (8) takes the same form of Eq. (3), where:

$$\mathbf{A} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{(k_1 + k_2)}{m_1} & \frac{k_2}{m_1} & -\frac{(c_1 + c_2)}{m_1} & \frac{c_2}{m_1} \\ \frac{k_1}{m_2} & -\frac{k_2}{m_2} & \frac{c_1}{m_2} & -\frac{c_2}{m_2} \end{bmatrix}$$

$$\mathbf{B}_u = \left[0 \;\; 0 \;\; -\frac{1}{m_1} \;\; \frac{1}{m_2}\right]^{\mathrm{T}} \quad \mathbf{B}_f = \left[0 \;\; 0 \;\; \frac{1}{m_1} \;\; 0\right]^{\mathrm{T}} \tag{9}$$

$$\mathbf{C} = \begin{bmatrix} \mathbf{I}_{4\cdot 4} \\ [\mathbf{O}_{2\cdot 2} \;\; \mathbf{I}_{2\cdot 2}]\cdot\mathbf{A} \end{bmatrix}$$

$$\mathbf{D}_{yu} = \begin{bmatrix} \mathbf{O}_{4\cdot 1} \\ [\mathbf{O}_{2\cdot 2} \;\; \mathbf{I}_{2\cdot 2}]\cdot\mathbf{B}_u \end{bmatrix} \qquad \mathbf{D}_{yf} = \begin{bmatrix} \mathbf{O}_{4\cdot 1} \\ [\mathbf{O}_{2\cdot 2} \;\; \mathbf{I}_{2\cdot 2}]\cdot\mathbf{B}_f \end{bmatrix}$$

ATMDs can be seen either as a way of improving the performance of a TMD by adding an actuator or as a way of reducing the actuation force and power consumption of an AMD by adding a spring (and, eventually, a dashpot), between the auxiliary mass and the main structure. In the first case, the design of the ATMD can be performed, for a white noise excitation, in closed form following a procedure similar to that suggested by Ayorinde and Warburton [8] for TMDs, if some simplifications are made [10]. The spring stiffness k_2 and dashpot constant c_2 are chosen following any criterion for the design of TMDs. Then the active control force is set proportional to the response vector $[\ddot{x}_{rel} \; \dot{x}_{rel} \; x_{rel}]^{\mathrm{T}}$, x_{rel} being the device stroke, and the feedback control law is designed so as to minimise the RMS displacement of the main structure. Once the acceleration gain μ_o is chosen, it is possible to derive the optimum values of the velocity and displacement gains. As an alternative, the complete state feedback gains $\mathbf{K} = [\bar{k}_1\ \bar{k}_2\ \bar{k}_3\ \bar{k}_4]$ are evaluated with a LQR control procedure.

3. Comparison of the performance of TMDs, AMDs and ATMDs: 1 + 1 DoF model under white noise excitation

As an example, the performance of a TMD and of an AMD, attached to a 64-storey building (details are given in Section 5), were first compared on a 1 + 1 SDoF system model. The modal mass, damping ratio and natural frequency are $m_1 = 1.09 \cdot 10^7$ kg, $\xi_1 = 0.01$ and $f_1 = 0.16$ Hz, respectively.

The main structure is subjected to a stationary Gaussian white noise input with spectral density $S_o = 3 \cdot 10^{11}$ N^2s, corresponding to the value of the spectrum of the building first modal buffeting force at the first resonant frequency (see Section 5).

First a TMD was considered, with a mass ratio $\mu = 0.01$, and optimal tuning and damping evaluated following the criterion by Warburton which minimises the RMS of the main structure displacements ($\Omega = 0.99$ and $\xi_2 = 0.05$). For the design of the AMD gain matrix, the LQR approach was used. The choice of the weights in Eq. (5) was done following a heuristic procedure, based on the expected magnitude of the system response and of the control force. The control force weight R was set equal to 10^{-12}. Then, it was noticed that for a narrowband response the ratio between the displacement and velocity amplitudes is approximately equal to the angular frequency of the system. Therefore, to make velocities and displacements provide similar contributions to the cost functional of Eq. (5), one has to set $q_1/q_3 = q_2/q_4 = \omega_1^2 = (2\pi f_1)^2$, which in this particular case leads to $q_1 = q_3$ and $q_2 = q_4$. The choice of the displacement weights q_1 and q_2 was done based on the expected values of the response and of the control force. In order for the device to work properly, the displacements of the auxiliary mass have to be five to 10 times larger than those of the main structure (for the TMD $\tilde{x}_{rel}/\tilde{x}_1 \cong 7$ was found), which would require the ratio q_1/q_2 to be in the range of 25 to 100. This suggested a first choice of the ratio q_1/q_2 of 50 (AMD1). Two more values of the ratio q_1/q_2 were considered of 120 (AMD2) and 50 000 (AMD3), which slightly and heavily overweight the main structure response, respectively. Finally, it was observed that the RMS displacement of the main structure is expected to be in the order of a few centimetres ($\tilde{x}_1 = 6$ cm for the structure provided with the TMD) and the RMS control force in the order of about 50 KN (159 KN is the peak passive reaction of the TMD). Therefore, for the addends in Eq. (5) to be well balanced, the ratio of q_1 to R would need to be in the order of $(50000/0.06)^2 = 7 \cdot 10^{11}$. It was found, however, that this brings very low control forces. A trial and error procedure indicated that the control force has to be underweighted in the cost function for the control to be effective.

In Table 1 the statistics of the response to a 54-min excitation time history are shown, together with the values of the maximum control force and power required by the AMDs. The value of the maximum reaction of the TMD is also given, which is the maximum force exerted by the dashpot and the spring, i.e. $\max(c_2\dot{x}_{rel} + k_2x_{rel})$. The TMD reduces the RMS response of the uncontrolled main structure by 42%. AMD1 brings to almost the same main structure response as the TMD, but the stroke is 22% lower. AMD2 and AMD3 bring a moderate and strong reduction of the main structure response, respectively, which are obtained at the expense of larger strokes, control forces and actuation power.

In Fig. 2 the spectra of the control force are shown, together with the spectrum of the external excitation. While the TMD, AMD1 and AMD2 operate in a narrow range of frequencies around resonance, AMD3 operates in broad range of frequencies, and the spectrum of the control force eventually tends to match the spectrum of the external excitation. This behaviour indicates that as the control gains increase, the control force tends to counteract the external excitation, rather than the inertia force of the vibrating main structure. This is a pathological behaviour, as beyond some threshold value a large increase in the control force corresponds to a small reduction of the response. It is clear that some upper bound has to be given to the control force for the device to be effective.

In Fig. 3 samples of the time histories of control force are shown. It appears that the control force of AMD1 is similar to the reaction of the TMD, but it also includes higher frequency fluctuations. The control force of AMD3 is of a completely different nature, with much larger fluctuations and peak values. In Figs. 4 and 5 the spectra of the main structure displacements and of the auxiliary mass relative displacements are shown, respectively. AMD3 prevents any resonant behaviour, but this requires unacceptable strokes.

As a third step, three different ATMDs were designed with the LQR procedure, and the performance of each compared to that obtained with a closed form design, following the procedure of Ankireddi and Yang [10]. For the ATMDs, the spring and the dashpot act as restrains to the motion of the auxiliary mass, and this suggests that the weights q_2 and q_4 on the auxiliary mass state be set equal to zero, which leads $\bar{k}_2 = 0$. Following the same criterion used for the AMDs, it is set $q_1 = q_3$, and q_1 is chosen such that $\bar{k}_1$ be not too large, as this would result in a detuning of the device and in oversized control forces. The control force gain R was again set equal to 10^{-12}, and three different values of the ratio q_1/R were considered of 10^{10} (ATMD1), 10^{11} (ATMD2) and 10^{13} (ATMD3). The acceleration gain μ_o in the closed form

Table 1
Response of the 1 + 1 DoF system with AMDs and ATMDs

	Uncontrolled	TMD	AMD1	ATMD1		AMD2	ATMD2		AMD3	ATMD3	
			$\frac{q_1}{q_2}=50$	$\frac{q_1}{R}=10^{10}$	$\mu_\circ=-0.004$	$\frac{q_1}{q_2}=120$	$\frac{q_1}{R}=10^{11}$	$\mu_\circ=-0.007$	$\frac{q_1}{q_2}=50000$	$\frac{q_1}{R}=10^{13}$	$\mu_\circ=-0.0097$
$\tilde{x}_1[m]$	0.105	0.061	0.060	0.056	0.056	0.049	0.048	0.048	0.018	0.028	0.028
$\hat{x}_{rel}[m]$	–	1.48	1.15	1.60	1.83	1.58	2.10	2.42	12.96	4.93	5.70
$\ddot{\tilde{x}}_1[ms^{-2}]$	0.119	0.079	0.079	0.075	0.075	0.071	0.069	0.070	0.055	0.058	0.058
$F_{max}[kN]$	–	159	155	8	17	228	36	42	751	285	332
$P_{max}[kW]$	–	–	101	8	22	197	45	66	2600	623	904

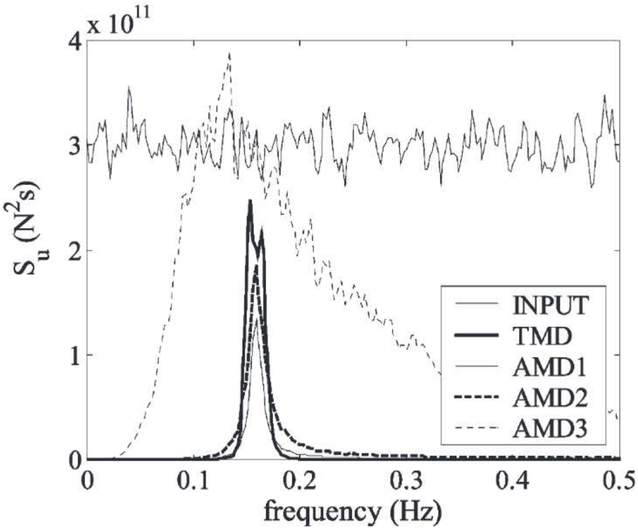

Fig. 2. Spectra of the AMDs control force for the 1 + 1 DoF model.

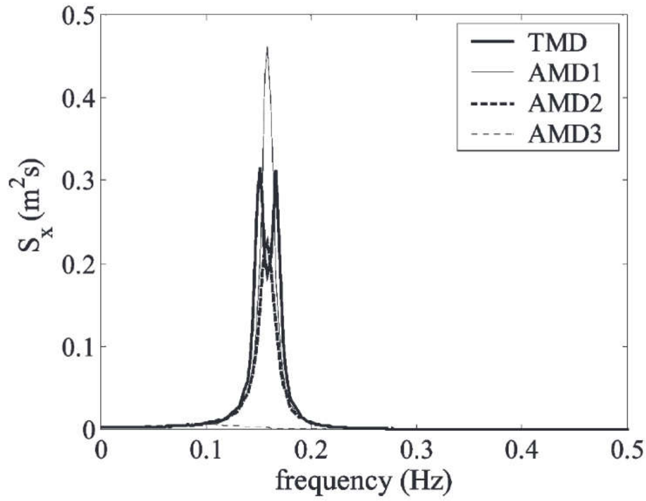

Fig. 4. Spectra of the main system displacements evaluated on the 1 + 1 DoF model.

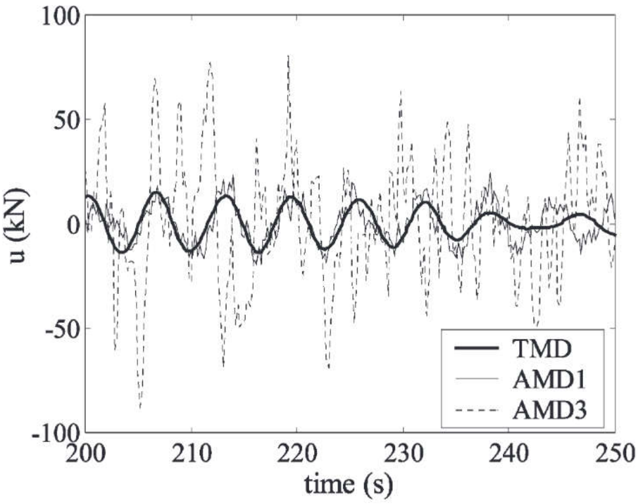

Fig. 3. Sample time histories of the AMDs control force for the 1 + 1 DoF model.

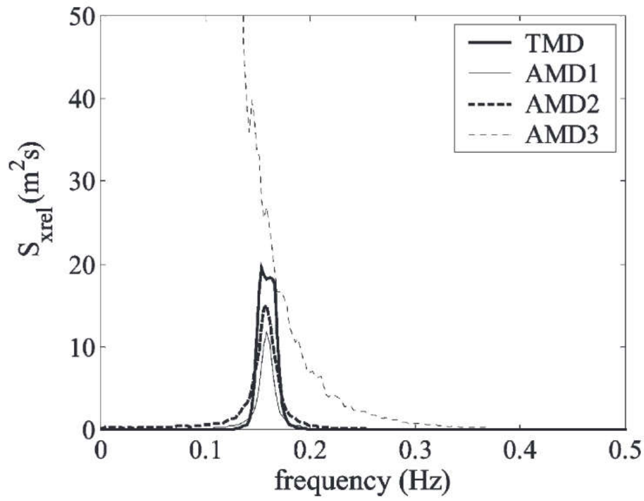

Fig. 5. Spectra of the device relative displacements evaluated on the 1 + 1 DoF model.

procedure was chosen so as to obtain the same main structure RMS response as that obtained with the corresponding device designed with the LQR. This allows comparison of the performance of the two procedures in terms of actuator stroke, control force and actuation power.

In Table 1 the statistics of the response, of the control force and actuation power are shown. The response of the main structure provided with ATMD1 and ATMD2 compares to that of the main structure provided with AMD1 and AMD2, respectively. In the case of ATMD1, however, the control force is almost 20 times lower, and the power almost 12 times lower than they would be if an active device (AMD1) were used. The drawback is an increase of the device stroke of about 40%. In the case of ATMD2, the control force and control power are about six and four times lower than for the corresponding active device (AMD2), indicating that when the control force of the ATMD is increased, the advantage of the hybrid device with respect to the active device reduces. The active force in an ATMD has therefore to be small, and designed as a minor contribution to the passive reaction.

In Figs. 6, 7 and 8 the spectra of the passive reaction, active control force and total control force are plotted, for the ATMDs and for the TMD. In all the cases most of the control force is provided by the spring and the dashpot. In the case of ATMD3, the increase in the active control force is accompanied by an increase in the passive reaction of the same order of magnitude, and by a much lower increase in the total control force. This indicates that an overdesigned active control force tends to counteract, through larger strokes, the spring and dashpot actions.

In Figs. 9 and 10 the spectra of the main structure displacements and of the auxiliary mass relative displacements are plotted. As for the AMD, an increase in the control gains allows the device to be operated in a broader range of frequencies, which reduces the effectiveness of the control system.

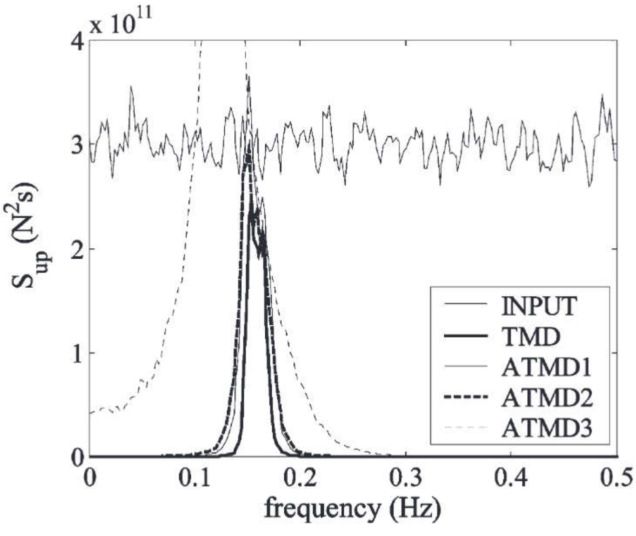

Fig. 6. Spectra of the ATMDs passive reaction for the 1 + 1 DoF model.

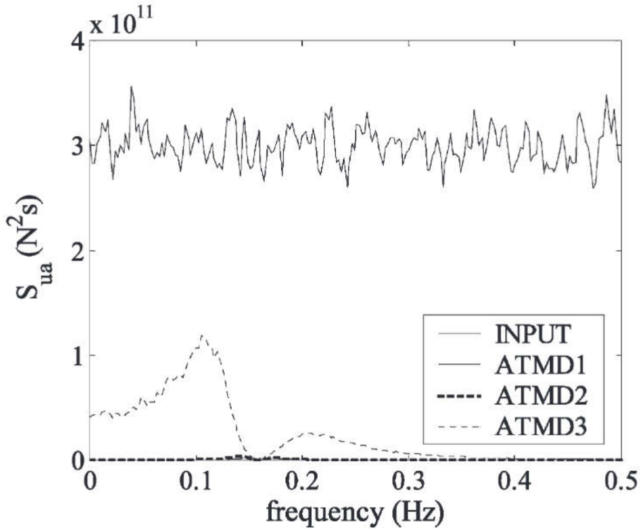

Fig. 7. Spectra of the ATMDs active control force for the 1 + 1 DoF model.

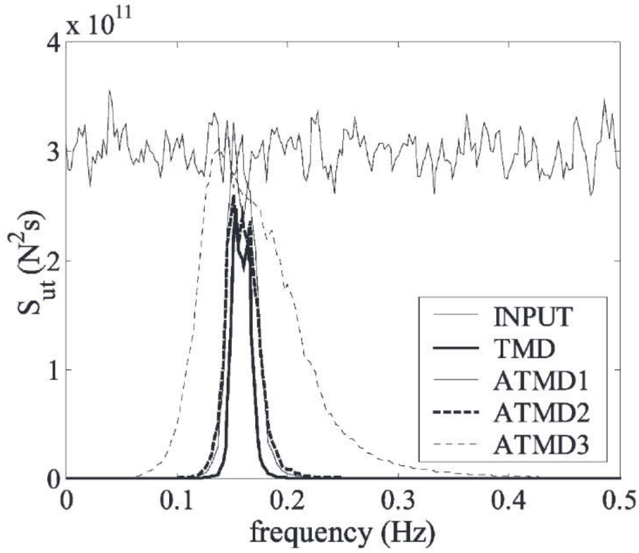

Fig. 8. Spectra of the ATMDs total control force for the 1 + 1 DoF model.

4. Effectiveness of passive and active mass damper control: tall buildings

Literature optimisation criteria allow the components of the TMD to be easily sized, but are based on a number of simplifying hypotheses. In Ricciardelli [11] it was shown how the shape of the spectrum of the excitation and the contribution of the higher modes of vibration

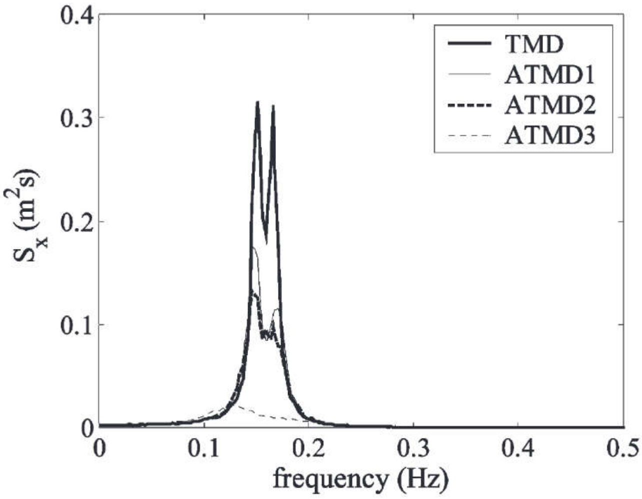

Fig. 9. Spectra of the main system displacement evaluated on the 1 + 1 DoF model.

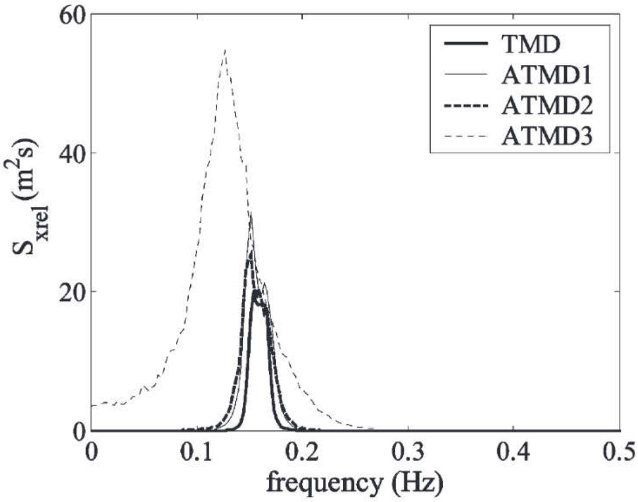

Fig. 10. Spectra of the device relative displacement evaluated on the 1 + 1 DoF model.

affect the performance of TMDs. In addition, in many cases the practical design is based on a trade-off solution between the reduction of the structural response and the constraints imposed to the motion of the auxiliary mass.

On the other hand, the use of an AMD allows simultaneous control of more than one mode of vibration, which may prove advantageous when acceleration reduction is of concern. The use of an ATMD also allows control of more modes of vibration, as the actuators can provide control forces over a broad frequency range. However, as was shown in Section 3, for an ATMD to work properly the control forces have to be in a narrow frequency range, around the natural frequency of the device.

To discuss these issues further, the tall building described in Section 3 will again be investigated, accounting for more than one mode of vibration and for a more realistic atmospheric turbulence input. The performance of the TMDs and of the AMDs and ATMDs will be again compared.

5. Comparison of the performance of TMDs, AMDs and ATMDs: MDoF model under coloured noise excitation

The 64-storey steel building shown in Fig. 11 was considered. The building is 256 m tall and with a square plan of 26.50 m. A reduced four DoF model was used, including the building first four longitudinal modes of vibration, having natural frequencies of 0.16, 0.64, 1.27 and 1.87 Hz, respectively, and a damping of 1% in all modes.

The mean wind profile was modelled with a power law with an exponent α = 0.15 and a reference mean wind velocity U_{10} = 28 m/s. The buffeting modal force was calculated using the Kaimal turbulence spectrum and the Vickery coherence function, with a vertical decay coefficient C_z = 10 (see [12]). A uniform intensity of turbulence was considered of 8%. A drag coefficient of 1.4 was used for the calculations, for a wind direction orthogonal to a building face.

Sixteen, 54-min long correlated time histories of the wind velocities at elevations of 16, 32…256 m were gen-

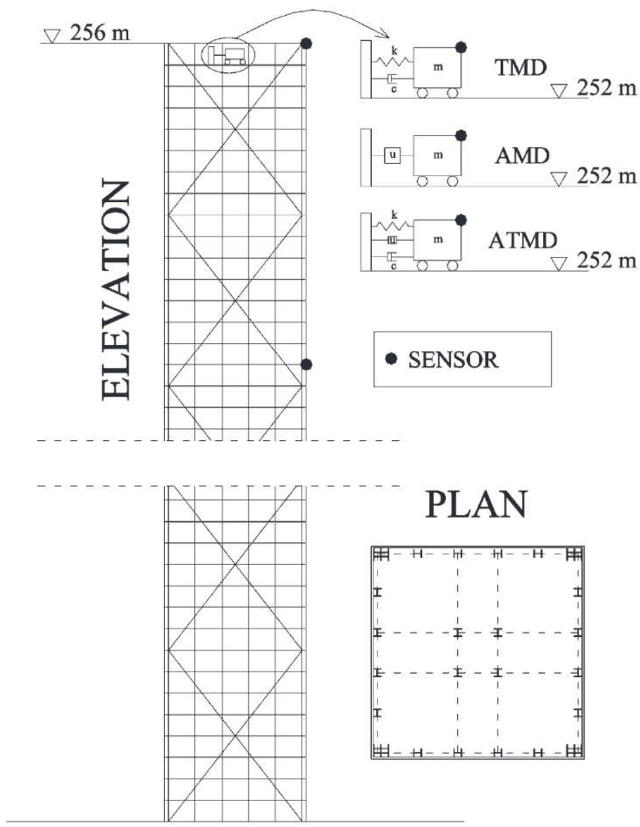

Fig. 11. Schematic of the 64-storey building.

erate using a second order AR filter [13], which were used to calculate the aerodynamic forces. The mean and the nil-mean fluctuating forces were separated, and used to evaluate the mean and fluctuating response, respectively.

The three different MD control strategies were compared, all with an auxiliary mass of 1% of the building first modal mass (m_d = 109 t), located at an elevation of 252 m.

The stiffness and damping of the TMD and of the ATMD were chosen following the criterion of minimisation of the first mode RMS displacement. As was done for the 1 + 1 DoF system, the active control law was designed following a LQR approach. In the case of the tall building modelled as a 4 + 1 DoF system, however, the state was not available for immediate feedback, and had to be estimated from the building 64th and 50th floor accelerations and from the auxiliary mass acceleration, using a state observer. As a first step, from the measured accelerations the corresponding velocities and displacements were evaluated by integration. A Kalman filter was then used for the stochastic estimation of the system state.

In a state space formulation, the equations of the system are:

$$\dot{\mathbf{x}}(t) = \mathbf{A}\mathbf{x}(t) + \mathbf{B}_u u(t) + \mathbf{B}_f \mathbf{f}(t) \tag{10}$$
$$\mathbf{y}(t) = \mathbf{C}\mathbf{x}(t) + \mathbf{D}_{yu} u(t) + \mathbf{D}_{yf}\mathbf{f}(t) + \boldsymbol{\nu}$$

where $\mathbf{y}(t)$ is the measured vector, $\mathbf{f}(t)$ and $\boldsymbol{\nu}$ are the external excitation and the measurement noise, and where $\mathbf{C}$, $\mathbf{D}_{yu}$ and $\mathbf{D}_{yf}$ are appropriate matrices, depending on the characteristics of the control device (either AMD or ATMD), and on the measured output [14].

As in the 1 + 1 DoF model, three AMDs and three ATMDs were considered. The choice of the terms of the 10 · 10 $\mathbf{Q}$ matrix and the scalar R was done following criteria similar to those described in Section 3. For each case examined the velocity weight q_v (the same value for all modes), was chosen and the weight on the i-th mode displacement q_{di} was set equal to $\omega_i^2 \cdot q_v$.

The equation of the Kalman filter used to estimate the state vector $\mathbf{x}(t)$ in Eq. (10) is:

$$\dot{\tilde{\mathbf{x}}}(t) = \mathbf{A}\tilde{\mathbf{x}}(t) + \mathbf{B}_u u(t) + \mathbf{L}(\mathbf{y}(t) - \mathbf{C}\tilde{\mathbf{x}}(t) - \mathbf{D}_{yu} u(t)) \tag{11}$$

where $\tilde{\mathbf{x}}(t)$ is the estimated state vector. The observer gain matrix $\mathbf{L}$ [15] is obtained as:

$$\mathbf{L} = \mathbf{P}\mathbf{C}^{\mathrm{T}}\mathbf{V}^{-1} \tag{12}$$

where $\mathbf{V}$ is the covariance matrix of the measurement noise ($\mathbf{V} = 10^{-5} \cdot diag[1\ 1\ 1]$ was used in the calculations), and $\mathbf{P}$ is the solution of Riccati equation: state vector $\mathbf{x}(t)$ in Eq. (10) is:

$\mathbf{W}$ being the covariance matrix of the external excitation $\mathbf{f}(t)$.

In Table 2 the statistics of the response of the building provided with AMDs and ATMDs are given, respectively, as obtained on the 4 + 1 DoF model. Comparison of Tables 1 and 2 shows how the effectiveness of the TMD and of the AMDs in terms of the reduction of the main structure displacement tends to be overestimated when a 1 + 1 DoF model is used, combined with a white noise excitation. As an example, using the 1 + 1 DoF model, AMD1 was calculated to reduce the main structure displacement by 43%, while on the 4 + 1 DoF model the main structure displacement is reduced by 25%. An opposite effect is obtained when the reduction of the main structure accelerations is of concern. The use of a model including more than one mode of vibration allows control of the response in the higher modes, which in the case of the accelerations significantly contribute to the total response. A reduction of the accelerations of 34% was predicted on the 1 + 1 DoF model using AMD1, while a reduction of 40% was calculated using the 4 + 1 DoF model. It is also noted that the 1 + 1 DoF model tends to underestimate the device stroke. Also in the case in which ATMDs are used, the predictions of the response carried out using a 1 + 1 DoF model turn out to be inaccurate. In this case, however, the reduction of the accelerations is also overestimated by the 1 + 1 DoF model, as the device acts only on the first mode. However, controlling the higher modes of vibration with a ATMD would require extraordinarily high control forces.

In Figs. 12–17 the smoothed spectra of the control force and of the response of the building provided with AMDs and ATMDs are plotted in a semi-logarithmic scale, as evaluated on the 4 + 1 DoF model.

In Fig. 12 the spectra of the AMDs control force are plotted. In agreement with what was pointed out in Section 3 for the 1 + 1 DoF case, the control force for AMD3 is a wide band process as the control force tends to directly counteract the external excitation. For AMD1 and AMD2 the spectrum of the control force lays below that of AMD3, except at the four resonant frequencies. This suggests that AMD1 and AMD2 are effectively controlling the resonant response of the structure.

The spectra of the building tip displacements are shown in Fig. 13. The control action of AMD3 eliminates all four resonant peaks. However, the RMS tip displacement of the building, evaluated on the 4 + 1 DoF model is larger than that evaluated on the 1 + 1 DoF model of Section 3. This is due to the use of a more realistic wind excitation, including a background action, which would require a large control effort to obtain a reduction of the background response. In the case of the accelerations there is no background action, and the use of an AMD allows control of the system response in the whole frequency range (Fig. 14).

Table 2
Response of the 64-storey building with AMDs and ATMDs (4+1DoF model)

	Uncontrolled	TMD	AMD1 $\frac{q_{d1}}{q_{dm}} = 50$	ATMD1 $\frac{q_{d1}}{R} = 10^{10}$	AMD2 $\frac{q_{d1}}{q_{dm}} = 120$	ATMD2 $\frac{q_{d1}}{R} = 10^{11}$	AMD3 $\frac{q_{d1}}{q_{dm}} = 50000$	ATMD3 $\frac{q_{d1}}{R} = 10^{13}$
$\tilde{x}_1[m]$	0.109	0.082	0.080	0.079	0.074	0.074	0.063	0.064
$\hat{x}_{rel}[m]$	–	1.27	1.52	1.40	2.53	2.12	42.7	9.44
$\tilde{\ddot{x}}_1[ms^{-2}]$	0.105	0.075	0.063	0.072	0.054	0.065	0.028	0.036
$F_{max}[kN]$	–	137	139	13	192	69	668	843
$P_{max}[kW]$	–	–	77	14	155	100	3600	2200

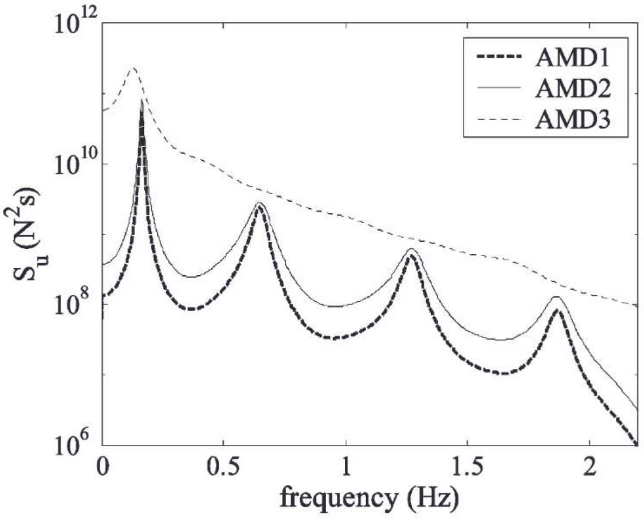

Fig. 12. Spectra of the AMDs control force for the 4 + 1 DoF model.

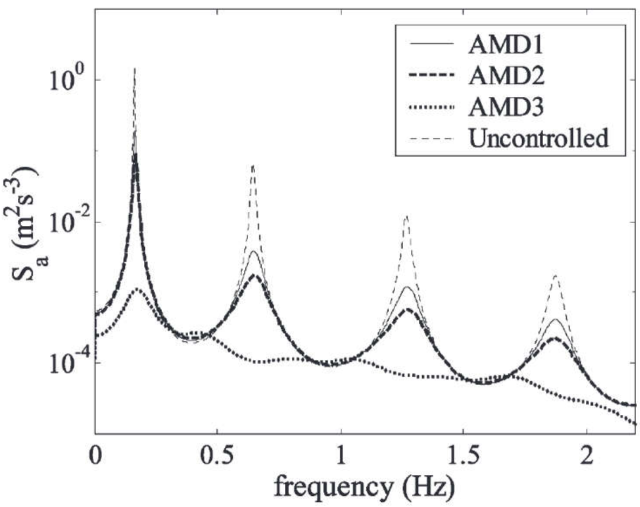

Fig. 14. Spectra of building tip accelerations evaluated on the 4 + 1 DoF model.

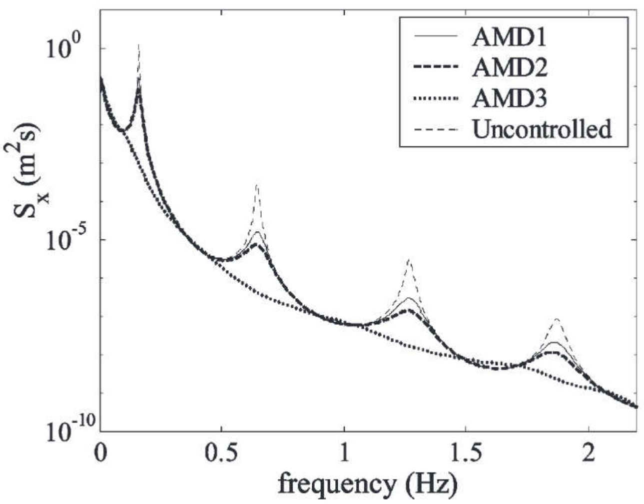

Fig. 13. Spectra of building tip displacements evaluated on the 4 + 1 DoF model.

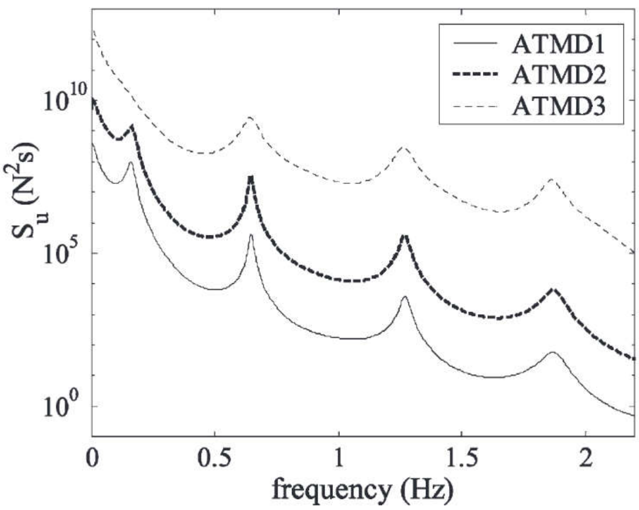

Fig. 15. Spectra of the ATMDs active control force for the 4 + 1 DoF model.

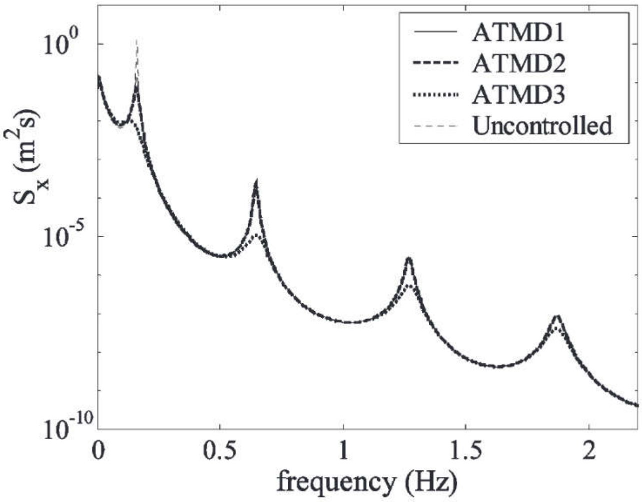

Fig. 16. Spectra of building tip displacements evaluated on the 4 + 1 DoF model.

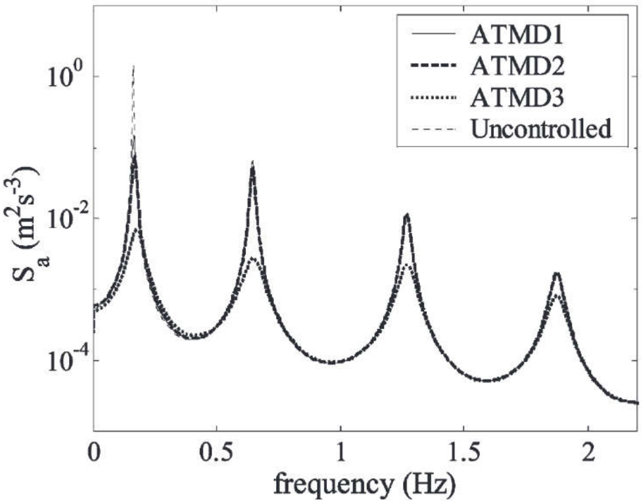

Fig. 17. Spectra of building tip accelerations evaluated on the 4 + 1 DoF model.

In Figs. 15–17 the spectra of the active control force and building tip displacements and accelerations are shown, respectively, for the cases in which ATMDs are added to the building. From comparison of Fig. 16 with Fig. 13 it can be observed that, while increasing the control effort on an AMD allows better control of all the modes, increasing the control effort on an ATMD allows better control of only the first mode. At the first natural frequency, in fact, the active control force interacts with the spring and dashpot reactions increasing the damping capacity, while at the higher natural frequencies the control force tends to be resisted by the reactions of the spring and dashpot. Comparison of Figs. 16 and 17 with Figs. 13 and 14 confirms this finding, showing how in the case in which an ATMD is used, even for large values of the active control force, no total mitigation of the resonant response peaks can be reached.

6. Conclusions

The performance of passive (TMD), active (AMD) and hybrid (ATMD) mass dampers for the reduction of the buffeting response of tall buildings has been compared in this paper.

A first comparison has been carried out on a simple 1 + 1 DoF model, using a white noise input. It has been shown how, even though theoretically unbounded, the performance of AMDs is in real life constrained by the large strokes and the large control forces and power required by the actuation system. On the other hand, the use of ATMDs required much lower control forces and power, and for this device to work properly, the active force has to be designed as a minor modification to the passive reaction. This allows ATMDs to improve the damping capacity of TMDs, with very reasonable control effort.

A second comparison was carried out using a 4 + 1 DoF model, and accounting for a more realistic wind excitation. Comparison with the results obtained on the 1 + 1 DoF model showed that the latter can provide misleading information about the damping capacities of the devices, for two main reasons. First because neglecting the higher modes of vibration can bring an overestimation or an underestimation of the performance of the damping device, depending on whether the response parameters to be mitigated are displacements or accelerations. Second because the use of a simplified white noise input does not allow account to be made of the background excitation, indeed significantly contributing to the displacement response, which cannot be effectively controlled.

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