Employing hybrid tuned mass damper to solve off-tuning problems for controlling human induced vibration in stadia

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ABSTRACT

A key objective in the design of any sports stadium is to include maximum number of spectators with minimum obstruction in the visual cone. This functional requirement often results in employing one or more cantilevered tiers, which in turn culminates in more slender grandstands often with relatively low natural frequencies and modal damping ratios. These natural frequencies may sometimes fall in the range of frequencies of human movement which can possibly excite the structure in resonance resulting in vibration serviceability issues. One of the available techniques to reduce excessive responses is to use passive vibration control techniques such as Tuned Mass Dampers (TMD). However, the off-tuning problem is a substantial drawback of this technique, whereby changes in natural frequencies caused by crowd-structure interaction may detune the TMDs. This paper presents a study into the possibility of using hybrid (combination of active and passive control) technology to augment the vibration serviceability of sports stadia. It shows a comparative analysis of vibration mitigation performances that are likely to be attained by utilising a passive TMD and the proposed HTMD. An appropriate control scheme is utilised with the proposed HTMD to deal with the off-tuning issues in TMDs caused by crowd loading, and is shown to be effective.

Keyword: Human Induced Vibration, Hybrid Control, Active/Passive Control, Hybrid Tuned Mass Damper, HTMD

INTRODUCTION

It is well known that there is a possibility for concert arenas and grandstands to be vulnerable to human activities such as jumping and bobbing. This is evident mainly in relatively slender structures as a result of expansion in material technologies and structural design skills, rising and increasing utilisation of sport stadia for live music performances and also more energetic and active audiences who have more synchronised movements [1–5].

In order to deal with this problem and satisfy vibration serviceability criteria, various kinds of control methods have been proposed and executed including passive, active, semi-active and hybrid vibration control.

Passive vibration control is an established method to improve the performance of a structure by dissipating vibration energy by applying additional dissipating materials or equipment to the structure and, as a result, the damping and occasionally stiffness of the structure increase. They have relatively simple design and there is no requisite for external power sources [6]. However, they might not be completely engaged especially at low levels of vibrations [7, 8]. In addition, they also have the possible drawback of off-tuning [9–11], which is mostly due to the change of natural frequency of the structure in the presence of human occupants [12-16].

Hybrid control contains an integration of passive and active control systems. It is created by the combination of active and passive segments (also known as composite active-passive controllers) to reduce structural response mostly by energy dissipation through the passive part, whereas the active part is included to improve its performance. In hybrid control systems the active part is smaller and less power is required than for a fully active system [8, 17].

The work presented here demonstrates the effect of employing a passive control method (Tuned Mass Damper - TMD) and a proposed hybrid control strategy (TMD with active element), which aims to improve the vibration performance of a typical stadium structure. A suitable control algorithm is developed with the proposed HTMD to deal with off-tuning issues in TMDs produced particularly by crowd loading. The dynamic properties of the employed stadium here are obtained from past stadia modal testing [12]. Also, in order to have more accurate outcome from both active and passive spectators in stadium, the suggested model in [13] was occupied.

MODEL OF GRANDSTAND

The structure considered in this study is a 7m cantilevered upper seating tier situated in the corner of a football stadium located in United Kingdom (Fig.1) [12]. In-service monitoring during a lively music performance confirmed that the highlighted area was quite lively and had a maximum acceleration higher than the guidance recommended at the time.

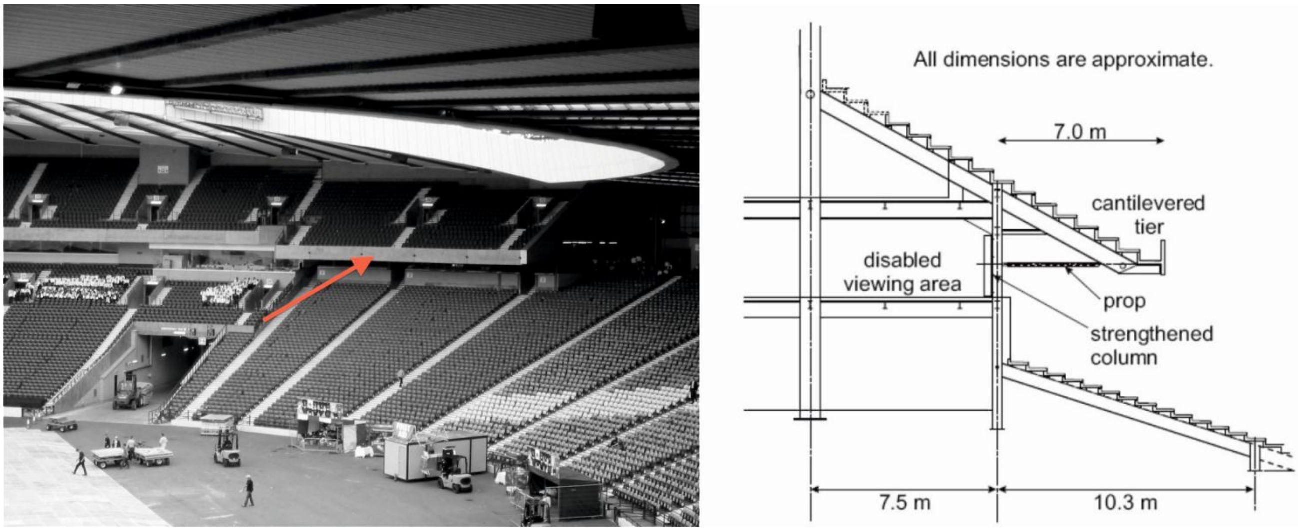

Fig.1: View of the modeled seating deck (left) and cross section of the tier (right) [12]

The first vertical natural frequency of the structure is 4.34 Hz derived from both ambient test and updated finite element (FE) model (Table 1) [12, 13]. However, due to the measured human-structure interaction phenomenon and also from the model of the stadium in the presence of both active and passive spectators, the frequency of the structure dropped from 4.34 Hz to 3.2 Hz [13, 19]. This placed the structure in the second harmonic of the music's frequency which has the frequency of 1.6 Hz. It should be noticed that in [12, 13] the frequency of the full structure is 3.80 Hz. This difference is due to the assumption of choosing the number of occupied people in this paper, which are only the first 7 rows of the tier (i.e. only the cantilever part).

Table 1: Dynamic properties of the stadium

Structure	Frequency(Hz)	Damping Ratio(%)	Modal Mass(Kg)	Modal Damping (Ns/m)	Modal Stiffness (N/m)
Empty	4.34	3.70	82,811	167,105	61,578,233
Full	3.20	11.00	108,019	567,396	61,578,233

A state space approach is employed [15] in order to use the three-degree-of-freedom (3DOF) model (Fig.2) that was proposed in [13]. The states of the system (Eq.(1)) are [15] :

$$\left\{\begin{array}{l} X_1 = x_s \\ X_2 = \dot{x}_s \\ X_5 = x_{as} \\ X_6 = \dot{x}_{as} \\ X_7 = x_{ps} \\ X_8 = \dot{x}_{ps} \end{array}\right\} \quad \therefore \quad \left\{\begin{array}{l} \dot{X}_1 = X_2 \\ \dot{X}_5 = X_6 \\ \dot{X}_7 = X_8 \end{array}\right\} \qquad (1)$$

In Fig.2, m_s is the mass of the empty structure, m_{as} is the total mass of active spectators and m_{ps} is the total mass of passive spectators. Also c_s, c_{as}, c_{ps} and k_s, k_{as}, k_{ps} are the respective damping coefficients and stiffnesses of the empty structure, active and passive spectators. P_{as} is the motion induced force produced within the body unit [3]. $\ddot{x}_s, \ddot{x}_{as}, \ddot{x}_{ps}$ are the accelerations associated with the masses of the structure, active and passive spectators, respectively. Further, $\dot{x}_s, \dot{x}_{as}, \dot{x}_{ps}$ and x_s, x_{as}, x_{ps} are velocity and displacement of the structure, active and passive spectators, respectively.

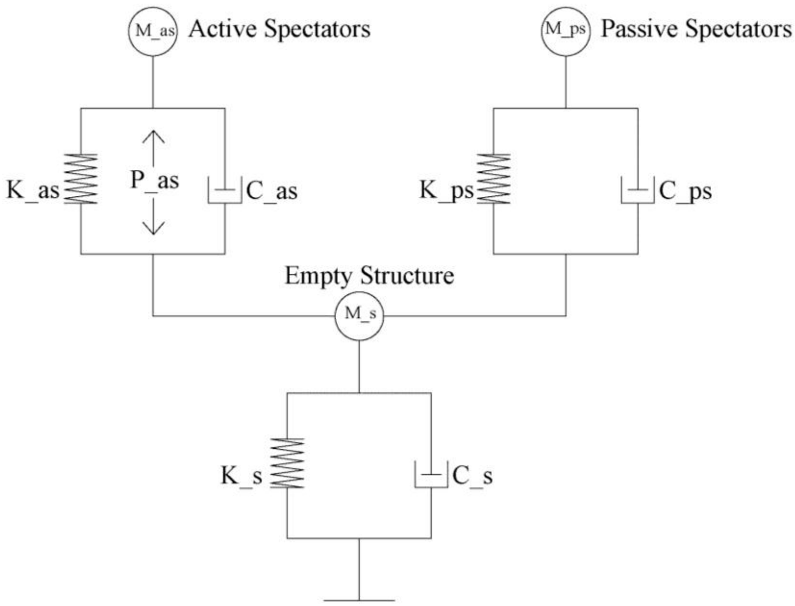

Fig.2: 3DOF model of the structure [15]

The state space representation of Fig.2 is given in Eq.(2) where Y_1, Y_2 and Y_3 are the displacement, velocity and acceleration of the main structure, respectively.

$$\begin{Bmatrix} \dot{X}_1 \\ \dot{X}_2 \\ \dot{X}_5 \\ \dot{X}_6 \\ \dot{X}_7 \\ \dot{X}_8 \end{Bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\frac{(k_s+k_{as}+k_{ps})}{m_s} & -\frac{(c_s+c_{as}+c_{ps})}{m_s} & \frac{k_{as}}{m_s} & \frac{c_{as}}{m_s} & \frac{k_{ps}}{m_s} & \frac{c_{ps}}{m_s} \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_{as}}{m_{as}} & \frac{c_{as}}{m_{as}} & -\frac{k_{as}}{m_{as}} & -\frac{c_{as}}{m_{as}} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ \frac{k_{ps}}{m_{ps}} & \frac{c_{ps}}{m_{ps}} & 0 & 0 & -\frac{k_{ps}}{m_{ps}} & -\frac{c_{ps}}{m_{ps}} \end{bmatrix} * \begin{Bmatrix} X_1 \\ X_2 \\ X_5 \\ X_6 \\ X_7 \\ X_8 \end{Bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{m_s} \\ 0 \\ -\frac{1}{m_{as}} \\ 0 \\ 0 \end{bmatrix} * \{P_{as}\}$$

$$\begin{Bmatrix} Y_1 \\ Y_2 \\ Y_3 \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ -\frac{(k_s+k_{as}+k_{ps})}{m_s} & -\frac{(c_s+c_{as}+c_{ps})}{m_s} & \frac{k_{as}}{m_s} & \frac{c_{as}}{m_s} & \frac{k_{ps}}{m_s} & \frac{c_{ps}}{m_s} \end{bmatrix} * \begin{Bmatrix} X_1 \\ X_2 \\ X_5 \\ X_6 \\ X_7 \\ X_8 \end{Bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{1}{m_s} \end{bmatrix} * \{P_{as}\} \qquad (2)$$

HYBRID TUNED MASS DAMPER (HTMD)

Fig.3 illustrates the model of a HTMD attached to the stadium structure with both active and passive people. This HTMD (Fig.4) consists of a passive TMD integrated with an active element (i.e. actuator). The vibration energy dissipation is achieved by the passive part whereas the active part helps the system to improve its performance by dealing with the off-tuning problem and low-level vibration issues.

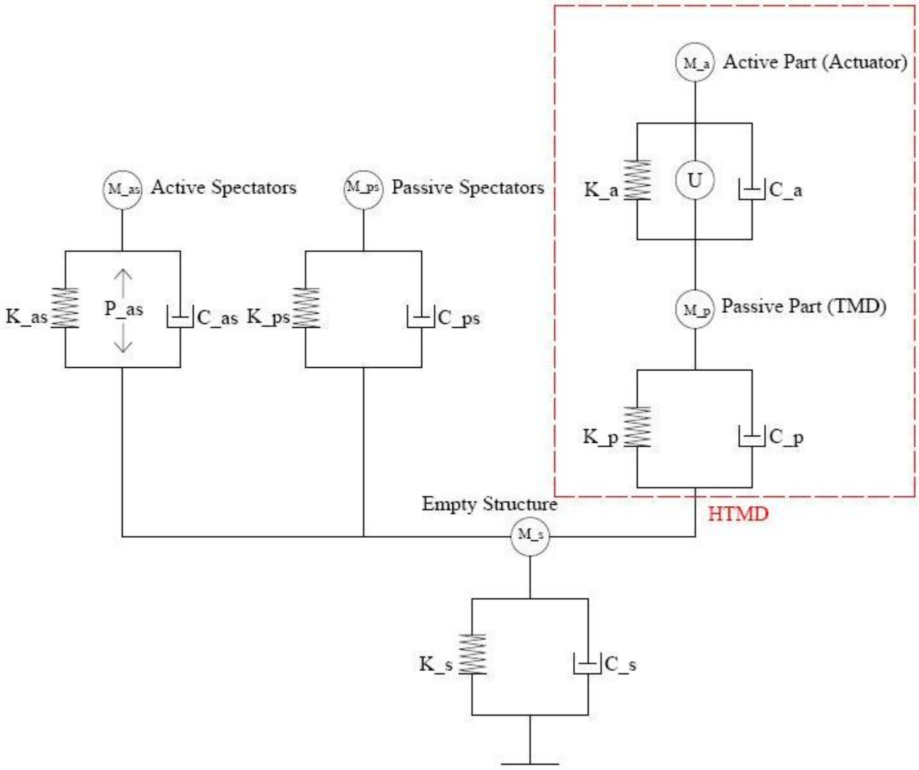

Fig.3: HTMD attached to the stadium cantilever

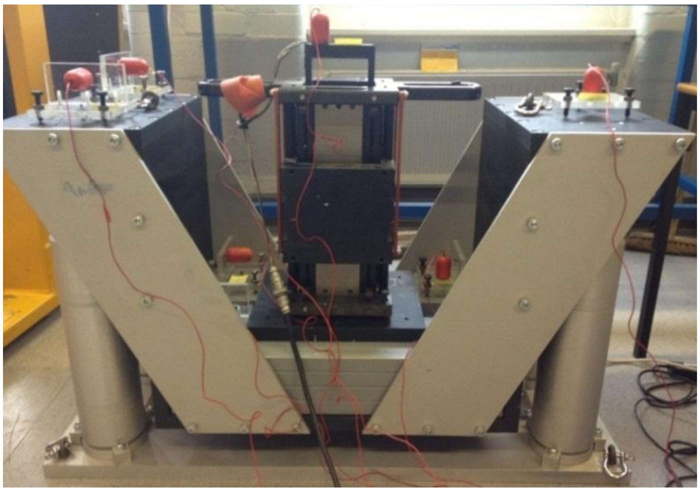

Fig.4: Laboratory HTMD

Using the equations of motion derived by the authors in [15], the state space approach of the HTMD system is shown in Eq. (3). It should be noticed that as shown in [15], the DOF of the active part of the HTMD can be replaced by the inertia force of the actuator (i.e. $F_{I,a}$) in the equations of motion of the system. This force is derived using a transfer function between the actuator's input voltage (V_{in}) and its inertia force ($F_{I,a}$) [15]. This transfer function is included in the state space matrix in Eq. (3).

$$
\begin{Bmatrix} \dot{X}_1 \\ \dot{X}_2 \\ \dot{X}_3 \\ \dot{X}_4 \\ \dot{X}_5 \\ \dot{X}_6 \\ \dot{X}_7 \\ \dot{X}_8 \\ \dot{X}_9 \\ \dot{X}_{10} \\ \dot{X}_{11} \end{Bmatrix} =
\begin{bmatrix}
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
-\frac{(k_s+k_p+k_{as}+k_{ps})}{m_s} & -\frac{(c_s+c_p+c_{as}+c_{ps})}{m_s} & \frac{k_p}{m_s} & \frac{c_p}{m_s} & \frac{k_{as}}{m_s} & \frac{c_{as}}{m_s} & \frac{k_{ps}}{m_s} & \frac{c_{ps}}{m_s} & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\frac{k_p}{m_p} & \frac{c_p}{m_p} & -\frac{k_p}{m_p} & -\frac{c_p}{m_p} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\
\frac{k_{as}}{m_{as}} & \frac{c_{as}}{m_{as}} & 0 & 0 & -\frac{k_{as}}{m_{as}} & -\frac{c_{as}}{m_{as}} & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\
\frac{k_{ps}}{m_{ps}} & \frac{c_{ps}}{m_{ps}} & 0 & 0 & 0 & 0 & -\frac{k_{ps}}{m_{ps}} & -\frac{c_{ps}}{m_{ps}} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{(k_a*\varepsilon)}{m_a} & -\frac{(c_a*\varepsilon+k_a)}{m_a} & -\frac{(m_a*\varepsilon+c_a)}{m_a}
\end{bmatrix}
$$

$$
\begin{Bmatrix} X_1 \\ X_2 \\ X_3 \\ X_4 \\ X_5 \\ X_6 \\ X_7 \\ X_8 \\ X_9 \\ X_{10} \\ X_{11} \end{Bmatrix} +
\begin{bmatrix}
0 & 0 & 0 \\
\frac{1}{m_s} & 0 & 0 \\
0 & 0 & 0 \\
0 & \frac{1}{m_p} & 0 \\
0 & 0 & 0 \\
-\frac{1}{m_{as}} & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & w_a
\end{bmatrix}
\begin{Bmatrix} P_{as} \\ F_{1,a} \\ V_{in} \end{Bmatrix}
$$

$$
\begin{Bmatrix} Y_1 \\ Y_2 \\ Y_3 \\ Y_4 \\ Y_5 \\ Y_6 \\ Y_{13} \\ Y_{14} \\ Y_{15} \end{Bmatrix} =
\begin{bmatrix}
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
-\frac{(k_s+k_p+k_{as}+k_{ps})}{m_s} & -\frac{(c_s+c_p+c_{as}+c_{ps})}{m_s} & \frac{k_p}{m_s} & \frac{c_p}{m_s} & \frac{k_{as}}{m_s} & \frac{c_{as}}{m_s} & \frac{k_{ps}}{m_s} & \frac{c_{ps}}{m_s} & 0 & 0 & 0 \\
0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\frac{k_p}{m_p} & \frac{c_p}{m_p} & -\frac{k_p}{m_p} & -\frac{c_p}{m_p} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & m_a
\end{bmatrix}
\begin{Bmatrix} X_1 \\ X_2 \\ X_3 \\ X_4 \\ X_5 \\ X_6 \\ X_7 \\ X_8 \\ X_9 \\ X_{10} \\ X_{11} \end{Bmatrix}
$$

$$
+\begin{bmatrix}
0 & 0 & 0 \\
0 & 0 & 0 \\
 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
0 & & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0
\end{bmatrix}
\begin{Bmatrix} P_{as} \\ F_{1,a} \\ v_{in} \end{Bmatrix} \tag{3}
$$

Where m_p, c_p, k_p and m_a, c_a, k_a are mass, stiffness and damping of the passive and active part of the HTMD respectively. ε and w_a are respective actuator's low pass filter element and force-voltage characteristic. Y_4, Y_5, Y_6 are displacement, velocity and acceleration of the passive part of HTMD respectively. In addition, Y_{13}, Y_{14}, Y_{15} are associated with displacement, velocity and inertia force of the active part of HTMD.

As was discussed in [15], for the proposed HTMD, two feedback gains including G_1 and G_2 are employed. The role of these are to produce a 'Driving Force' which enhances the TMD's inertia force and also provides an 'Active Damping Force' which regulates the damping force of the HTMD respectively. Herein, in addition to these previous two gains, another two extra additional feedback gains are introduced (G_3 and G_4), which are displacement and acceleration feedback of the passive part of the TMD respectively. The role of these two is to change the dynamic properties of the TMD which leads to tune it to the new frequency. In another words, these two gains deal with off-tuning of the HTMD.

$$\begin{cases} \text{Driving Force} = G_1 * \ddot{x}_s \\ \text{Active Damping} = G_2 * \dot{x}_p \\ \text{Tuning Force} = G_3 * x_p \\ \text{Tuning Force} = G_4 * \ddot{x}_p \end{cases} \tag{4}$$

ANALYTICAL STUDY

In order to compare the performance of the HTMD with passive TMD under the off-tuning problem, an analytical simulation was implemented in the MATLAB/Simulink software. The same random white noise signal was applied as an external excitation to the uncontrolled structure, the structure with a passive TMD and finally to the structure with an HTMD. As an evaluation method, frequency response function (FRF) plots were produced. The occupied TMD and HTMD's properties (i.e. mass, stiffness and damping ratios) are in Table 2. The passive TMD in [15] was tuned to the situation where the ratio of active/passive people in the stadium was 40:60. Also, the frequency of the structure in this case was 3.20 Hz. **Error! Reference source not found.** shows the FRF of the system when the passive TMD is tuned to 3.20 Hz. As was shown in [15], the HTMD performs better compared to an equivalent passive TMD, even when it is properly tuned to the structural frequency.

Table 2: Dynamic Properties of the HTMD and TMD

	Mass Ratio	Frequency Ratio	Frequency (Hz)	Damping Ratio (%)	Mass (Kg)	Damping (Ns/m)	Stiffness (N/m)
TMD	2.6%	98.40%	3.15	5.1	2,174	4,412	850,786
HTMD	2.6%	98.40%	3.15	5.1	2,174	4,412	850,786

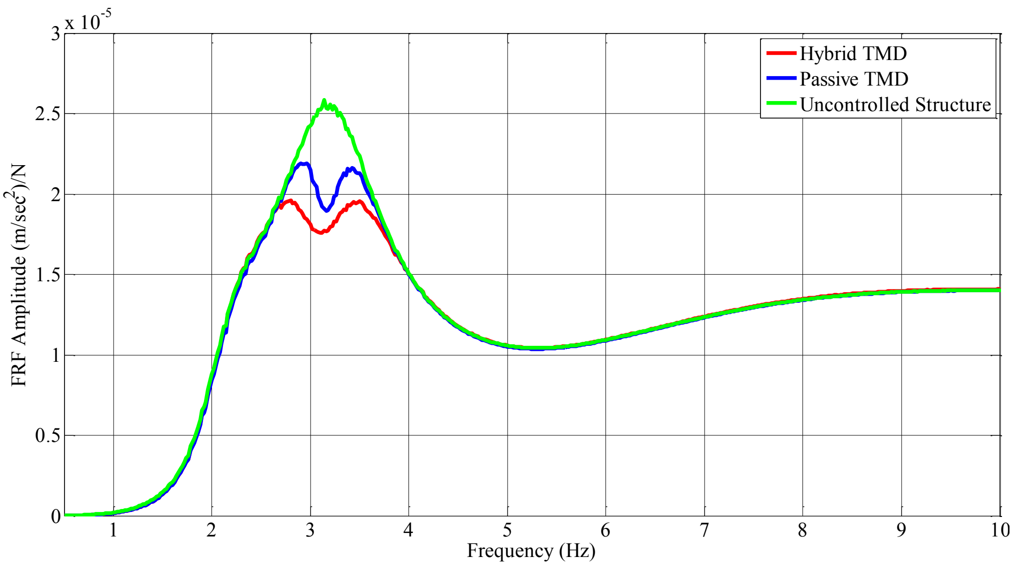

Fig.5: FRF of the structure with natural frequency of 3.20 Hz

To investigate the effect of off-tuning due the changes in the frequency of the structure, the percentage of the active spectators in stadium was changed to various ratios from 1% to 99%, as indicated in Table 3. This resulted in a change of the total mass of the DOF of active spectators (i.e. m_{as}), which leads to a variation in the frequency of the main structure. As is shown in Table 3, the frequency of the structure varies from 2.68 Hz to 5.18 Hz whilst the passive TMD is tuned to 3.20 Hz.

Table 3: Properties of the structure with different ratios of active people

Scenario	Active people (%)	Mass of Active people m_{as} (kg)	Mass of Passive people m_{ps} (kg)	Frequency of the structure (Hz)	Changing of the frequency of the main structure (%)
1	1%	970	96005	2.68	-15%
2	5%	4849	92126	2.68	-15%
3	10%	9698	87278	2.78	-12%
4	20%	19395	77580	2.86	-9%
5	30%	29093	67883	3.04	-3%
6	40%	38790	58185	3.20	0%
7	60%	58185	38790	3.52	+12%
8	80%	77580	19395	4.14	+31%
9	99%	96005	970	5.18	+64%

As the result of the changing frequency of the main structure, the TMD becomes detuned and its effectiveness is reduced. However, the proposed HTMD can deal with this off-tuning by employing the introduced feedback gains. In order to select the appropriate gain, firstly root locus analyses were performed for individual gains separately to achieve the range of the gains to guarantee stability of the system. Following this, a manual sensitivity approach (considering the effect of changing the gains on the response) was applied by combining the gains and achieving the minimum peaks in the calculated FRFs.

Table 4 shows the selected HTMD gains and the percentage of the response reduction compared with the uncontrolled structure and the structure with passive TMD. For the comparison between TMD and HTMD, both peaks of the FRF and also the response of FRF at the structural resonant frequency have been considered. As is noted in Table 4, below the frequency of the tuning, since the differences in the frequency (e.g. 15%) are not high compared to those above 3.20 Hz, it is sufficient to just employ one of the tuning gains, G_4 (i.e. acceleration of the TMD), in addition to HTMD damping force gain, G_2, to deal with off-tuning and to reduce the response over the frequency band encompassing the structural mode.. However, above the tuning frequency (3.20 Hz), due to larger differences in the frequencies (up to 64 %), in addition to off-tuning, the HTMD needs to expend more effort to enhance its performance which is the reduction in the response. Hence, another off-tuning gain G_3 in addition to driving force gain (i.e. G_1) is applied to the HTMD to drive the HTMD more compare to the less sever scenario (less than 3.20 Hz).

Table 4: Comparison of the TMD and HTMD performance in off-tuning

Scenario	G_1	G_3	G_2	G_4	TMD peak reduction (%)	HTMD peak reduction (%)	TMD reduction at resonance (%)	HTMD reduction at resonance (%)
1	0	0	-44	-2	4	40	4	44
2	0	0	-41	-2	1	33	1	36
3	0	0	-36	-2	5	28	9	34
4	0	0	-28	-1	7	22	9	27
5	0	0	-17	-1	10	19	22	22
6	-40	-4125	-61	-11	15	24	26	31
7	-40	-1375	-22	-2	7	24	8	36
8	-10	-4400	-8	-2	3	10	3	17
9	-5	-8250	-55	-1	1	8	1	11

In addition to this, Fig.6 shows plots of the FRF amplitudes for different scenarios. It demonstrates that the performance of TMD is reduced when the structural frequency changes, whereas the HTMD is able to compensate for the detuning.

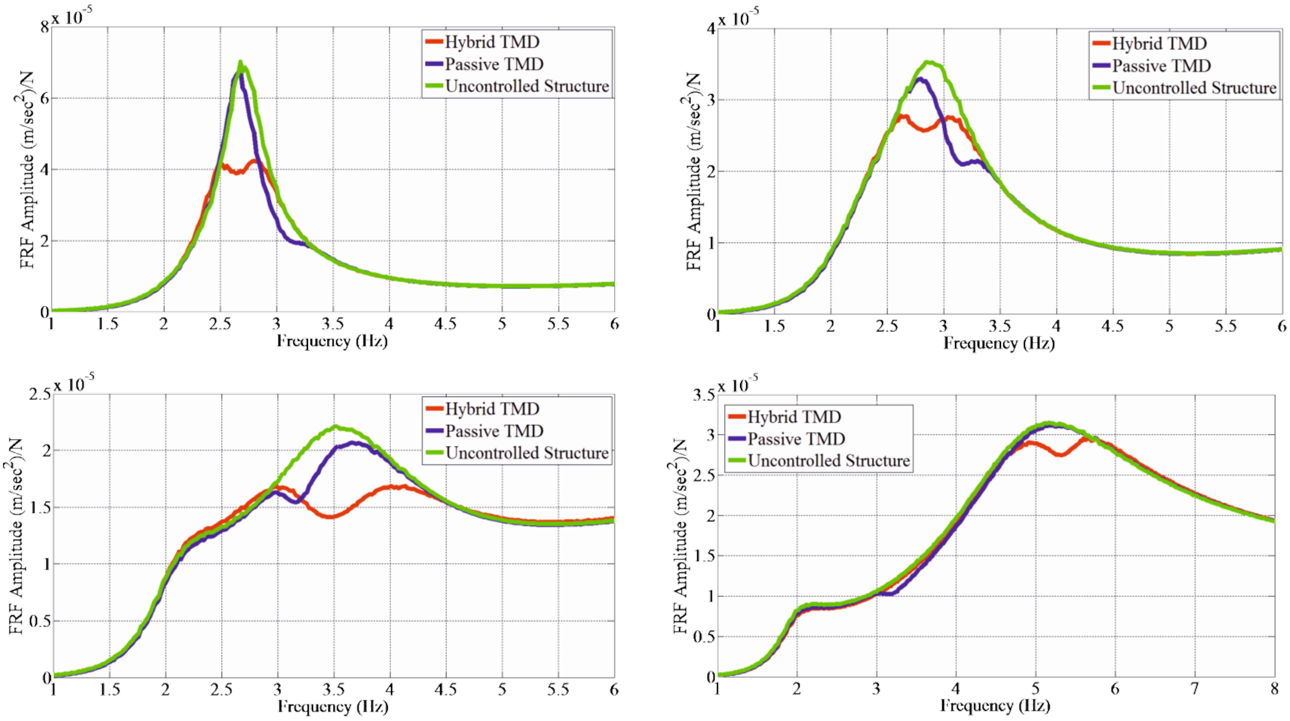

Fig.6: FRF scenario 1 (top-left), scenario 4 (top-right), scenario 7 (bottom-left), scenario 9 (bottom -right)

CONCLUSIONS

Dealing with off-tuning stadium structures is an important concern for vibration control of stadia. This might occur due to changing number of spectators and even by changes in the number of active and passive people during a single event. Passive tuned mass dampers as a conventional method for vibration control have reduced effectiveness in the presence of off-tuning, since a change in the frequency of the primary structure leads to a detuned TMD.

Employing an HTMD in a grandstand has been investigated here by altering the percentage of the active and passive people which leads to modification of the resonant frequency of the primary structure. It has been shown that the proposed HTMD has the capability to deal with off-tuning when the frequency of the primary structure changes.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the financial assistance provided by the UK Engineering and Physical Sciences Research Council (EPSRC) through Leadership Fellowship grant "Advanced Technologies for Mitigation of Human-Induced Vibration" (Ref. EP/J004081/1) and platform grant "Dynamic Performance of Large Civil Engineering Structures: An Integrated Approach to Management, Design and Assessment" (Ref. EP/G061130/1).

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