Observer based adaptive tuned mass damper with a controllable MR damper

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ABSTRACT: This paper presents an observer based linearised control of semi-active tuned mass damper (OSTMD) with a controllable magnetorheological (MR) damper. The proposed model replaces a passive damping element with a controllable semi-active damper to emulate controllable stiffness and controllable damping, which distinguishes it from the classical passive tuned mass damper (TMD). As a result, the frequency of the tuned mass damper could be tuned in real time to changing frequencies of the main structure. Many semi-active control algorithms measured the damper force with load-cells, and the piston displacement of the damper, e.g. with a linear variable differential transformer (LVDT). However, the implementation of force transducers and LVDT sensors are difficult, and expensive in practice. They also increase the complexity of system. In this study, by proposing a non-linear observer, these data are estimated using two simple, low cost accelerometers which are placed on the structural mass and on the tuned mass. The observer gain matrix is evaluated by using a pole placement method based on Luenberger theory. Both the damping and the natural frequency are tuned according to classical theory. The proposed theory is validated by numerical and experimental testing under the broadband excitation and the results are compared to a passive tuned mass damper which is precisely tuned to the main structure. The comparison shows that, the observer based linearised semi-active tuned mass damper with controlled MR damper is able to reduce the vibration amplitude as much as the optimal passive tuned mass damper.

KEY WORDS: Semi-active tuned mass damper; Observer.

1 INTRODUCTION

Excessive vibration has been a common problem throughout engineering history which can result from a variety of sources; human body motions, rotating, oscillating and impacting machines, wind flows, earthquake induced vibration, road traffic, railway traffic, construction works etc. [1]. Often the most effective and economic way to reduce vibration is to apply an additional dynamic system at a discrete point on the existing structure, to change the system dynamics in a desired way [2]. Simple mass-spring-damper systems attached to a selected point of the vibrating structure (as seen in Figure 1-(a)), are one example. They are called tuned mass dampers (TMD), tuned vibration absorbers [3], or vibration neutralisers [2]. Historically, the first time tuned mass dampers were used to reduce the rolling motion of ships by Frahm in 1911 [4]. Later, TMDs were implemented to reduce the vibration amplitude of single degree of freedom systems by Ormondroyd and Den Hartog [5], and Brock [6]. Den Hartog developed closed form expressions of optimum damper parameters which are frequency ratio and damping ratio of the TMD [7]. These expressions are only for undamped main systems with a single degree of freedom. Later, damping in the main system was included by several researchers [8-9]. To summarise, simple optimum solutions to excessive vibration problems using passive tuned mass damper have been widely investigated and implemented on real structures [10].

However, all these proposed tuning methods for optimal passive TMD system assumed that the natural frequency and structural mass are known and do not change. If the modal properties of the main structure are changed (e.g. due to environmental effects) after the TMD has been installed, the performance can be significantly reduced [12-13]. These environmental effects could be the time-varying pay loads, estimation errors or temperature changes. All these problems will result in the de-tuning of the pre-tuned mass damper. As a solution to this generic de-tuning problem, many adaptive TMD control concepts have been proposed to develop new
designs and concepts by controlling the properties of the TMD in real time to match the changing properties of structure. Such concepts and their control were summarised by Fisco and Adeli [14]. Due to high power requirements, expense and fail-safety problems of the active system, adaptive tuned mass damper systems are designed mostly based on semi-active approaches [15]. After seven decades of first passive TMD damper implementation, Hrovat et al. introduced the concept of a semi-active TMD for wind-induced vibrations in a high rise building [16].

Semi-active dampers require simpler hardware and have lower power requirements than active systems, thus reducing operational costs. These devices can provide variable damping and/or stiffness and have been used for vibration reduction of mechanical and civil engineering structures. Semi-active tuned mass damper concepts have considered a tuned mass, a tuned passive spring, or a controlled semi-active damper. A variety of semi-active dampers have been proposed, such as piezo stack [17], active smart materials (shape memory alloy) [18], piezoelectric materials [19], controllable friction devices [20], and magnetorheological (MR) damper [21-22]. Weber et al. presented a semi-active approach where a rotational MR damper was used to emulate both controllable damping and stiffness force under harmonic excitation [15]. In addition to this, the same authors extended this numerical work by experimental implementation of proposed theory on a laboratory footbridge, using accelerometers and a force transducer to implement force tracking control [11]. They were able to reduce the vibration amplitude between 38% to 63% relative to a passive TMD system, but they did not investigate the performance of proposed system under random or broadband excitation which is more likely to be realistic case. In addition the use of a force transducer also increases the complexity of the system. In this study, these two drawbacks have been explored using an observer based linearised semi-active tuned mass damper system. The lumped parameters/block diagram model of this system is shown in Figure 2.

This paper offers a description of the proposed observer based semi-active tuned mass damper (OSTMD), along with a dynamic analysis of the optimal passive tuned mass damper (OPTMD) that is based upon numerical simulations (optimal tuning). Before introducing OSTMD, this paper describes a classical passive tuned mass damper to help distinguish the two systems. Lastly, this paper summarizes the dynamic performance of the observer base tuned systems and suggests the best control method for the proposed observer based semi-active system, based upon the numerical results.

2 NUMERICAL MODELLING AND OPTIMAL TUNING

A classical model of force-excited \( F_{ex} \) passive TMD configuration is shown in Figure 1-(a). The main structure is coupled with a classical passive vibration absorber, and the mass of the structure and absorber are defined by \( m_s \) and \( m_a \) with their corresponding displacements as \( x_s \) and \( x_a \), respectively. The absorber’s spring \( (k_a) \) and damper \( (c_a) \) are mounted on the structure. The stiffness and damping of the structure are represented by \( k_s \) and \( c_s \), respectively.

The equations of motion of the force-excited passive TMD in matrix form is:

\[
\begin{bmatrix}
\dot{x}_s \\
\dot{x}_a
\end{bmatrix} =
\begin{bmatrix}
0 & \frac{c_s + c_a}{m_a} \\
\frac{k_s + k_a}{m_a} & -\frac{k_a}{m_a} - k_s
\end{bmatrix}
\begin{bmatrix}
x_s \\
x_a
\end{bmatrix} + \begin{bmatrix}
F_{ex}
\end{bmatrix}
\]

The steady state solution of equation 1 can be obtained;

\[
\begin{bmatrix}
x_s \\
x_a
\end{bmatrix} =
\begin{bmatrix}
\frac{k_s}{\omega_{ss}^2 + \omega_{sa}^2} & -\frac{k_a}{\omega_{ss}^2 + \omega_{sa}^2} \\
-k_{sa} & \frac{k_a}{\omega_{ss}^2 + \omega_{sa}^2}
\end{bmatrix}
\begin{bmatrix}
\frac{m_s}{\omega_{ss}^2 + \omega_{sa}^2} \\
\frac{m_a}{\omega_{ss}^2 + \omega_{sa}^2}
\end{bmatrix}
\begin{bmatrix}
F_{ex}
\end{bmatrix}
\]

By defining the following parameters:

\[
\begin{align*}
\omega_{ss} &= \sqrt{\frac{k_s}{m_s}} \quad \text{Natural frequency of the main structure} \\
\omega_{sa} &= \sqrt{\frac{k_s}{m_a} + \frac{k_a}{m_a}} \quad \text{Natural frequency of the absorber} \\
\zeta_s &= \frac{c_s}{2m_s \omega_s} \quad \text{Damping ratio of main structure} \\
\zeta_a &= \frac{c_a}{2m_a \omega_a} \quad \text{Damping ratio of absorber} \\
g &= \omega_s \omega_a \quad \text{Ratio of natural frequencies} \\
r &= \omega_s / \omega_a \quad \text{Forced frequency ratio}
\end{align*}
\]

The final transmissibility equations for the forced-excited system become:

\[
\begin{align*}
X_s &= \frac{\omega_s^2 + 2\zeta_s g^2}{\omega_s^2 + \omega_r^2 + \omega_s^2 g^2 + \omega_r^2 g^2 + \omega_s^2 g^2 + \omega_r^2 g^2 + \omega_s^2 g^2} \\
X_a &= \frac{\omega_s^2 + 2\zeta_s g^2}{\omega_s^2 + \omega_r^2 + \omega_s^2 g^2 + \omega_r^2 g^2 + \omega_s^2 g^2 + \omega_r^2 g^2 + \omega_s^2 g^2}
\end{align*}
\]

The transmissibility Equation 5 provide the means of tuning TMDs using Mead’s [6] description. The target of the techniques is to minimize the maximum transmissibility.

3 SEMI-ACTIVE TMD DESIGN

The proposed observer base semi-active tuned mass damper (TMD) model replaces a passive damping element with a controllable semi-active damper to emulate controllable stiffness and controllable damping, which distinguishes it
from the classical passive system. Figure 1-(a) shows a conventional passive TMD model, and the proposed semi-active tuned mass damper model is shown in Figure 1-(b). A controllable damper, such as an MR damper, is the key element for the proposed system. It can provide a wide dynamic force range, can offer a real-time control environment at low power, and can be quite cost-effective. The unified model of MR damper [23] is chosen for this numerical study, due to not only it is flexibility in the choice of the model’s damping function, but also the performance of the unified model has been proved by experimental testing of Lord Corporation’s RD-1005-3 short stroke MR damper by Batterbee and Sims [24] under the broadband mechanical excitation.

Incorporating this versatile damper into the proposed model will significantly enhance its performance, combining the benefits of both passive and active systems. The new system is anticipated to surpass the performance of classical passive TMD in reducing the maximum vibration levels of the primary structure and robustly adapt to the primary system’s parameter changes.

Figure 3 was used to derive the dynamic equations of motion for the semi-active model:

\[
\begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2 \\
\dot{x}_3 \\
\dot{x}_4
\end{bmatrix} =
\begin{bmatrix}
0 & 0 & 1 & -1 \\
0 & 0 & 1 & 0 \\
k_s & 0 & 0 & 0 \\
-k_s & 0 & 0 & 0
\end{bmatrix}
\begin{bmatrix}
x_1 \\
x_2 \\
x_3 \\
x_4
\end{bmatrix} +
\begin{bmatrix}
0 \\
0 \\
1 \\
1
\end{bmatrix} u +
\begin{bmatrix}
0 \\
0 \\
0 \\
0
\end{bmatrix} w(t)
\]

(6)

Where \( F_{MR} \) is the force produced by the semi-active magnetorheological damper. Equation 6 will be used in the development of the numerical model of the semi-active TMD.

\[\begin{align*}
\dot{x}_1 &= \begin{bmatrix} 0 & 0 & 1 & -1 \end{bmatrix} x_1 + \begin{bmatrix} 0 \\ 0 \\ 1 \\ 1 \end{bmatrix} u + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} w(t) \\
\dot{x}_2 &= \begin{bmatrix} k_s & 0 & 0 & 0 \end{bmatrix} x_2 + \begin{bmatrix} 0 \\ 0 \\ 1 \\ 1 \end{bmatrix} u + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} w(t) \\
\dot{x}_3 &= \begin{bmatrix} k_s & 0 & 0 & 0 \end{bmatrix} x_3 + \begin{bmatrix} 0 \\ 0 \\ 1 \\ 1 \end{bmatrix} u + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} w(t) \\
\dot{x}_4 &= \begin{bmatrix} k_s & 0 & 0 & 0 \end{bmatrix} x_4 + \begin{bmatrix} 0 \\ 0 \\ 1 \\ 1 \end{bmatrix} u + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} w(t)
\end{align*}\]

(8)

The following dynamical system 9 is considered as an observer.

\[\begin{align*}
\dot{\hat{x}} &= A\hat{x} + Bu + Lz + G\hat{F}_{ex} \\
\hat{y} &= C\hat{x} + Du + H\hat{F}_{ex}
\end{align*}\]

(9)

Where \( z = y - \hat{y} \), \( \hat{x} \) is the observer state, \( \hat{y} \) is the observer output, \( \hat{F}_{ex} \) is the estimated disturbance force signal (Equation 10) and \( L \) is the observer gain matrix. The error between the actual state \( x \) and the observed state \( \hat{x} \) is defined as

\[ e = x - \hat{x} \]

(11)

Where the dynamics of the state estimation error is then given by;

\[ \dot{e} = (A - LC)e + (G - LH)(F_{ex} - \hat{F}_{ex}) \]

(12)

Here \( L \) is the 4x2 observer gain matrix. Several numerical test results showed that with a proper observer matrix the estimated disturbance force and the actual disturbance force matched each other. The pole placement method was used to evaluate the observer gain matrix [25].

5 MAIN PRINCIPLES AND CONTROL CONCEPTS

The proposed approach uses the feedback linearisation control of an MR damper which has been proposed by The University of Sheffield [26]. But the goal of this paper is to emulate controllable viscous damping and/or controllable spring stiffness of the equivalent optimal passive system. As seen in Figure 2, this desired damper force is produced by control concept part, which uses the estimated states of the system. Two types of control concept are proposed, which are explained as;
5.1 Control concept 1
The first control concept is intended to simply linearise the non-linear dynamics of the MR damper, so it behaves as a linear viscous damper with varying linear damping. With reference to Figure 4 the estimated desired set-point force is:
\[
\hat{F}_{\text{desired}} = c_a (\hat{x}_a - \hat{x}_a)
\]

![Figure 4. Lumped parameter models of tuned mass damper configurations. (a) passive tuned vibration absorber, (b) Semi-active MR based tuned vibration absorber (concept 1).](image)

Where \(\hat{F}_{\text{desired}}\) is the estimated desired damper force, \(c_a\) is the optimal absorber damping, and \((\hat{x}_a - \hat{x}_a)\) is the relative piston velocity. In this concept, the MR damper just emulates the optimal viscous damping of the passive tuned mass damper system.

5.2 Control concept 2
In the second control concept, the MR damper tries to emulate the optimal viscous damping and optimal spring stiffness of the passive tuned mass damper system. In this concept, optimal controllable spring stiffness is the sum of the emulated controllable stiffness and passive spring stiffness of the proposed system (as seen in Figure 1-(b)) due to parallel installation of the passive spring and controllable MR damper. This will enable frequency tuning of the TMD system by attempting to emulate the controllable positive or negative spring stiffness, and controllable damping force. Referring to Figure 1 -(b), the estimated desired set-point force is given by:
\[
\hat{F}_{\text{desired}} = c_{a,\text{optimal}} (\hat{x}_a - \hat{x}_a) + k_{\text{added}} (\hat{x}_a - \hat{x}_a)
\]

Where \((\hat{x}_a - \hat{x}_a)\) is the relative piston displacement, and \(k_{\text{added}}\) is the controllable stiffness which was assumed to be zero for control concept 1. But, for the control concept 2,
\[
k_{\text{added}} = k_{a,\text{opt}} - k_a
\]

These two control concepts are investigated by numerical and experimental testing. Tests are carried out in four steps with assumption that the absorber mass \((m_a)\), absorber stiffness \((k_a)\), structural spring stiffness \((k_s)\) and structural damping \((c_s)\) do not change at any step of testing. Then, \(k_{a,\text{opt}}\) and \(c_{a,\text{opt}}\) can be adjusted to the actual frequency of the main structure according to Mead’s formulae without any constraints [2]. The testing steps are:
- Step 1; tuning of the absorber, where the spring stiffness and damping level of the absorber are optimally tuned, and the desired force is driven by the control concept 1.
- Step 2; de-tuning of the main structure. In this step, the service load of main structure is changed by adding or removing masses to/from main structure. The new structural mass becomes, \(m = m + m_d\), where \(m_d\) represents change of the service load (which could be negative or positive). Assuming the structural stiffness does not change, the system frequency ratio changes, and this leads to the detuned system, as shown in Figure 5.

![Figure 5. Detuned lumped parameter models of detuned mass damper configurations. (a) passive detuned vibration absorber, (b) Semi-active MR based detuned vibration absorber (step 2).](image)

In this de-tuned case, the adaptive TMD with controllable MR damper is linearised to emulate the pre-tuned viscous damping of the optimal passive pre-tuned damper system as in concept 1.
- Step 3; retuning of the damping. The analytical frequency retuning of the detuned absorber is done according to Mead’s optimisation theory [2] by using the de-tuned structural mass \((m_d)\) and unchanged structural stiffness \((k_s)\) as seen in Figure 5.

![Figure 5. Detuned lumped parameter models of retuned mass damper configurations. (a) passive retuned vibration absorber, (b) Semi-active MR based retuned vibration absorber (step 3).](image)

In this study, it is assumed that, structural mass of this new detuned could be estimated by empirical mode decomposition and Hilbert transform (EMD/HT) [27], or
modified cross-correlation (MCC) method [13] in the real-time control process. So that, the frequency of the absorber system could be tuned to the actual frequency of the main structure by adjusting $k_a$ to $k_a,\text{opt}$ and $c_a$ to $c_a,\text{opt}$. This retuned passive TMD system is called as ideal adaptive TMD and expected to equalize the response peaks according to theory. In this step, just viscous damping was retuned so that MR damper only emulates the viscous damping of the passive TMD and estimated desired force was chosen by control concept 1.

- Step 4; retuning of damping and stiffness. In this step retuning of the stiffness element of the passive TMD was done according to theory. The estimated desired force was chosen by the control concept 2 so that the MR damper is forced to emulate the retuned viscous damping and retuned spring stiffness of the new retuned ideal adaptive TMD as seen in Figure 6.

$$\begin{bmatrix}
-969.97 & 5.5247 \\
25.3155 & -951.3691 \\
0 & 0.0001 & 0.0002 \\
0 & 0.0003 & -0.0001 \\
951.3691 & -25.3155 & 5.5240 \\
0 & 969.9747 & -25.3155
\end{bmatrix}$$

The main structure with optimal passive TMD and a observer based adaptive TMD with controllable MR damper is simulated for varied de-tuned structural masses $m_s = 5.0m, 5.6m, ..., 1.5m$ and for each mass the structure is excited with high-pass filtered random white noise force signal with bandwidth of 0-15 Hz of duration 300 seconds, for each of the four steps. The first ten seconds of this excitation force signal is shown in Figure 7.

![Figure 7. First ten seconds of the excitation force signal.](image)

The test results will indicate the damping potential of the proposed theory. The entire system is solved in MATLAB/Simulink using the ode4 (Runge Kutta) solver with fixed sampling frequency of 0.0002 Hz. The absorber viscous damper is replaced by the numerical unified model of MR damper where depending on the control concept; the MR damper will track the estimated desired force.

7 TEST RESULTS

Frequency response curves are used to analyse the performance of the each control concepts for numerical and experimental testing, which is evaluated by using tfestimate method of Simulink software. The Figure 8 indicates that the passive TMD is able to equalise the peaks at resonance amplitude, according the Mead’s theory (solid line, step 1 tuning). But, the damping performance dramatically decreases with detuning of the system (dashed and dash-dotted lines, step 2).

![Figure 8. Frequency responses of the passive TMD, for optimally tuned case (step 1, solid line) and detuned cases (step 2, dashed and dash-dotted lines).](image)

Table 1. Parameters of passive tuned mass damper.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of absorber</td>
<td>$m_a$</td>
<td>kg</td>
<td>448</td>
</tr>
<tr>
<td>Mass of structure</td>
<td>$m_s$</td>
<td>kg</td>
<td>18425</td>
</tr>
<tr>
<td>Absorber stiffness</td>
<td>$k_a$</td>
<td>Nm$^{-1}$</td>
<td>250760</td>
</tr>
<tr>
<td>Structural stiffness</td>
<td>$k_s$</td>
<td>Nm$^{-1}$</td>
<td>10821000</td>
</tr>
<tr>
<td>Absorber damping</td>
<td>$c_a$</td>
<td>Nsm$^{-1}$</td>
<td>2000</td>
</tr>
<tr>
<td>Structural damping</td>
<td>$c_s$</td>
<td>Nsm$^{-1}$</td>
<td>50</td>
</tr>
</tbody>
</table>

The observer gain matrix is than evaluated by using pole placement method such as that;
Also it is clear that from the Figure 8, if the detuning is done with increased structural mass, the host structure is over-damped with an almost un-damped absorber resonance frequency; whereas if the detuning process is done with decreased structural mass, the resonance frequency amplitude of the absorber is over-damped and the structural resonance is un-damped. As expected, Figure 9 illustrates that the amplitude reduction of the ideal adaptive TMD (with retuned damping and stiffness), depends on the frequency of the main structure.

![Figure 9. Frequency responses of the ideal adaptive TMD, for step 1 (solid line), and step 3 (dashed and dash-dotted lines).](image)

The frequency response of the observer based adaptive TMD with linearised MR damper controlled according to concept 1 is shown in Figure 10. It is clear that if the desired force gain is chosen as the viscous damping $c_a$ of the passive tuned mass system, then it is expected to achieve the same amplitude reduction for tuned and detuned cases of passive TMD. Comparing Figure 8 and Figure 10 clearly proves that for the step 1, the observer based adaptive TMD with linearised MR damper is able to equalize the peak for tuned values (solid line). In addition, also for the step 2 (detuned case), it is also able to follow the performance of the passive TMD (dashed and dash-dotted lines).

![Figure 10. Frequency responses of the adaptive TMD with linearised MR damper, for optimally tuned case (step 1,solid line), and de-tuned cases (step 2, dashed and dash-dotted lines).](image)

If the detuned system is retuned as in step 3 with control concept 1, then the observer based adaptive TMD with linearised MR damper improves the amplitude reduction compared to the passive TMD as seen in Figure 11. But, if the system is detuned with increased structural mass the frequency of the detuned structure get closer to the frequency of the absorber, which has greater effect on the system response. On the other hand, if the system is detuned with decreased mass, frequencies of host structure and the absorbers move away from each other.

![Figure 11. Frequency responses of the adaptive TMD with linearised MR damper, for optimally tuned case (step 1,solid line), and re-tuned cases (step 3, dashed and dash-dotted lines).](image)

However, the aim of this study to investigate that whether the observer based linearised MR damper can emulate the controllable stiffness and viscous damping or not. To find this out, the last numerical tests step 4 and control concept 2 are implemented, where the linearised MR damper will emulate both damping and stiffness forces. The results are shown in Figure 12, where up to several 100% amplitude reduction is achieved for the observer based adaptive TMD with linearised MR damper.

![Figure 12. Frequency responses of the adaptive TMD with linearised MR damper, for optimally tuned case (step 1, solid line) and re-tuned with controllable viscous damping and controllable stiffness cases (step 4, dashed and dash-dotted lines).](image)

In conclusion, these numerical test results show that the observer based linearisation of an adaptive TMD is able to emulate the positive or negative stiffness in addition to providing energy dissipation in the TMD.

REFERENCES


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