Development of a multi-passive tuned mass damper, theory and experiments

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ABSTRACT: In this paper, a bi-directional multi-passive tuned mass damper is presented. The application for the damper is on vertical hangers of an existing steel arch railway bridge. The hangers have been found susceptible to resonance and the resulting stresses results in a reduced service life due to fatigue. Due to different boundary conditions, the natural frequencies of the hangers are different in the longitudinal and the transverse direction. In addition, the natural frequencies increase during train passage, due to increased tensile force in the hangers. A prototype of the damper has been developed, consisting of two suspended masses coupled in series. Different lateral suspensions are used to obtain different natural frequencies in the longitudinal and the transverse direction. One mass is tuned to the conditions of the fully loaded bridge and the other mass to the unloaded bridge. The performance of the damper is verified using controlled loading under laboratory conditions and the results are compared with a finite element model. The damper is shown to perform as expected and the motion of the two masses is near uncoupled. Finally, the performance of the damper is verified by in-situ testing on the case study bridge.

KEY WORDS: Tuned Mass Damper; Railway Bridge Dynamics; Resonance; Frequency Response Function.

1 INTRODUCTION

There is constant demand on railway authorities to increase both the allowable axle loads and allowable train speeds on existing railway lines. For bridges on non-high speed railway lines, dynamic effects are often accounted for by increasing the static response with dynamic amplification factors, not accounting for the risk of resonance. For bridges with low mass, low natural frequencies and low damping, resonance may however occur even for moderate train speeds. In combination with long and heavy trains, resulting stresses may cause a significant reduction in the fatigue service life.

Previous studies of a steel arch railway bridge [1] and [2] have shown that some of the vertical hangers are susceptible to resonance during train passage, which results in a reduced service life due to fatigue. A system of passive pendulum dampers were installed [3], tuned to the natural frequencies of the unloaded bridge. During train passage however, the increased tensile force in the hangers show in a significant increase in natural frequencies. This may cause a detuning to a passive damper system. In addition, the boundary conditions of the hanger to main beam connection, Figure 1, results in different frequencies in the longitudinal and transverse direction.

Simulations of different adaptive and semi-active systems showed great potential in improving the performance [4], accounting for both the loaded and unloaded bridge conditions. Implementations of such systems are however a challenge, especially when based on variable stiffness control. A proposed compromise is instead a system of passive dampers with several natural frequencies to account for both the loaded and unloaded conditions as well as the difference in the longitudinal and transverse direction.

In this paper, the development of a bi-direction multi-passive tuned mass damper (bi-MTMD) is presented. The performance of the damper is verified under controlled loading and compared with numerical models. From free vibration tests of the unloaded bridge hanger, the natural frequencies \( f_{1x} = 4.3 \) Hz and \( f_{1y} = 3.6 \) Hz and appertaining damping ratios \( \zeta_{1x} = 0.15 \) % and \( \zeta_{1y} = 0.30 \) % was estimated. During train passages, the natural frequencies increased to \( f_{1xF} = 6.3 \) Hz and \( f_{1yF} = 5.5 \) Hz. The modal mass of the hanger is calculated to 135 kg.

![Figure 1. Details of a bridge hanger.](image)

2 NUMERICAL MODEL OF THE DAMPER

The initial design of the damper is based on the model illustrated in Figure 2. It works as a combination of a tuned mass damper and a pendulum damper. The mass consists of two steel rings that are suspended in wires and coupled in series. The length \( L \) is chosen sufficiently long to avoid
significant coupling between the upper and lower ring. Each ring is also connected to the hanger by horizontal springs and dashpots. The upper ring is tuned to the natural frequencies of the loaded hanger and the lower ring to the natural frequencies of the unloaded hanger.

During experimental testing, the prototype damper is connected to a sliding base and excited using a load shaker. The mass \( m_b \) of the sliding base is lumped to a single point and the properties of the load shaker estimated by a stiffness \( k_b \) and damping \( c_b \). The rings and the wires are modelled with Euler-Bernoulli beam elements. From an eigenvalue analysis, the modes in Figure 3 are obtained. The modes show that the upper and lower rings are near uncoupled.

![Figure 2. FE-model of the bi-MTMD.](image)

![Figure 3. Modes of vibration.](image)

3 EXPERIMENTAL TESTING

The prototype of the bi-MTMD is depicted in Figure 4. Each steel ring has a mass of 2.3 kg. The wires consists of 2 mm nylon strings and has a length \( L = 110 \text{ mm} \). The mass of the base \( m_b = 5.3 \text{ kg} \) and from a separate test of only the base and the load shaker, a natural frequency of 9 Hz and damping of 32% is estimated.

The horizontal connection between the ring and the hanger consists of coil springs and blocks of foam material, illustrated in Figure 5. The damper is primary tuned by modifying the coil springs. The foam is primary used to prevent excessive vibrations and to add additional damping, but also contributes to the stiffness.

![Figure 4. Prototype damper during load shaker tests.](image)

![Figure 5. Illustration of the components of the bi-MTMD.](image)

3.1 Setup and instrumentation

The damper is calibrated under controlled laboratory conditions using the setup illustrated in Figure 4. The system is instrumented with uni-axial accelerometers \( a_1, a_2 \) and \( a_b \). The base displacement \( d_b \) is measured with an LVDT and the input force \( F \) is measured with a load cell. The sensor specifications are provided in Table 1.

<table>
<thead>
<tr>
<th>sensor</th>
<th>make</th>
<th>range</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a_1, a_2 )</td>
<td>Entran EGCS-A2</td>
<td>( \pm 2 \text{ g} )</td>
</tr>
<tr>
<td>( a_b )</td>
<td>Sensotec 060-F482-02</td>
<td>( \pm 5 \text{ g} )</td>
</tr>
<tr>
<td>( F )</td>
<td>Vishay, Model 615</td>
<td>( \pm 50 \text{ kg} )</td>
</tr>
<tr>
<td>( d_b )</td>
<td>Sangamo DC25 LVDT</td>
<td>( \pm 25 \text{ mm} )</td>
</tr>
</tbody>
</table>

The laboratory equipment consists of the following:

- LDS V455 Electro-dynamic shaker.
- LDS PA1000L Linear Power Amplifier.
- Tektronix TDS 210 Oscilloscope.
- TGA1241 waveform generator.
- Vishay 7000-32-SM data acquisition system (24 bit).
The effective data resolution for the system is about 18 bit, corresponding to about $1 \times 10^3$ m/s$^2$ for the accelerometers, $0.4 \times 10^3$ mm for the LVDT and 0.2 N for the load cell. The performance of the sensors is studied in a separate test, by mounting all sensors onto the sliding base and removing the damper. The difference in peak acceleration between $a_1$ and $a_2$ is about 4%. Integrating the acceleration two times give similar results as measured with the LVDT and differentiating the measured displacement two times give similar results as the acceleration. Also, dividing the measured force with the known mass of the base results in similar acceleration.

### 3.2 Frequency Response Function

The dynamic characteristics of the damper can be studied by the frequency response function (FRF) of the measured response. An input load with linearly variable frequency and constant amplitude is sent to the load shaker via the oscilloscope and the linear power amplifier. A 10 min sweep is performed with a 3 V input gain and a frequency range from 2 to 10 Hz and then back to 2 Hz. All results are normalized with respect to a unit base displacement. The corresponding transfer function is calculated as the time domain response of the single sided Fourier transform of the measured base displacement.

The displacement of the damper is estimated based on time integration of the corresponding acceleration. The FRF is obtained directly from the peaks of the resulting time displacement. An input load with linearly variable frequency and constant amplitude is sent to the load shaker via the oscilloscope and the linear power amplifier. A 10 min sweep is performed with a 3 V input gain and a frequency range from 2 to 10 Hz and then back to 2 Hz. All results are normalized with respect to a unit base displacement. The corresponding transfer function is calculated as the time domain response of the single sided Fourier transform of the measured base displacement.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>$\zeta$</th>
<th>$d$</th>
<th>$F$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f_{1x}$</td>
<td>4.3</td>
<td>9.2</td>
<td>$F_{1x}$</td>
</tr>
<tr>
<td>$f_{1y}$</td>
<td>3.9</td>
<td>7.4</td>
<td>$F_{1y}$</td>
</tr>
<tr>
<td>$f_{2x}$</td>
<td>6.6</td>
<td>8.4</td>
<td>$F_{2x}$</td>
</tr>
<tr>
<td>$f_{2y}$</td>
<td>5.6</td>
<td>5.5</td>
<td>$F_{2y}$</td>
</tr>
</tbody>
</table>

To study the influence of the foam material, a separate test of the damper is performed with the foam removed. The results are presented in Table 3.

### Table 3. Estimated natural frequencies and damping ratio of the prototype damper without foam material, based on 10 min sine sweep load.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>$\zeta$</th>
<th>$d$</th>
<th>$F$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f_{1x}$</td>
<td>3.2</td>
<td>16.5</td>
<td>$F_{1x}$</td>
</tr>
<tr>
<td>$f_{1y}$</td>
<td>3.4</td>
<td>15.5</td>
<td>$F_{1y}$</td>
</tr>
<tr>
<td>$f_{2x}$</td>
<td>4.9</td>
<td>23.1</td>
<td>$F_{2x}$</td>
</tr>
<tr>
<td>$f_{2y}$</td>
<td>4.9</td>
<td>14.8</td>
<td>$F_{2y}$</td>
</tr>
</tbody>
</table>

The FE-model is calibrated based on the experimental data, with the natural frequency and steady-state displacement as the primary objective functions. All calibrated input parameters are given in Table 4, both with and without the foam material. The foam results in a stiffness increase of about a factor of 2 in the x-direction and about 40% in the y-direction. As illustrated in Figure 5, the foam is only present about a factor of 2 in the x-direction and about 40% in the y-direction due to the shear. The increase in viscous damping is about a factor 2 for both directions of the lower ring, a factor 3 for the upper ring in the y- and x-direction correspondingly. The large ratio of increase is due to the inherent low damping of the setup with only coil springs.

### 3.3 Response to harmonic load blocks

To simulate the conditions on the bridge, the damper is subjected to a harmonic load block. The base displacement is about 1.5 mm and the frequency change from 6.3 Hz to 4.2 Hz, Figure 7. This corresponds to the loaded and unloaded hanger conditions. The frequency in Figure 7b is estimated by a Short Time Fourier Transform with a 2 second window and 95% overlap.

The response of the damper is presented in Figure 8. The displacement is estimated by time integration of the measured acceleration. As a verification, the integrated base acceleration shows good agreement with the measured base displacement presented in Figure 7a. The measured base displacement is also used as input to the FE-model.

During loading, the upper ring is first activated, Figure 8a. The amplitude of the lower ring is about 2 mm, slightly higher than the base displacement of 1.5 mm. During the change in frequency, the displacement of the lower ring is rapidly attenuated and the lower ring activated. The FE-model is shown to reproduce a similar response.
Figure 7. Input for harmonic load block, a) base plate displacement, b) forced frequency.

Figure 8. Damper displacement due to the forced vibrations, a) upper ring, b) lower ring.

4 NUMERICAL MODEL OF THE BRIDGE

A 3D FE-model of the bridge has been developed, illustrated in Figure 9. The properties of the bridge are described in detail in [1] and [2]. It is a simply supported tied arch steel railway bridge, carrying a single unballasted track. The span length is 45 m. The bridge deck consists of main girders, secondary stringer beams and cross beams.

The whole bridge is modelled with 3D Euler-Bernoulli beam elements. To simulate the detail depicted in Figure 1, the lower end of the hanger is assumed pinned in the transverse direction and clamped in the longitudinal direction. The upper end of the hanger is assumed pinned in both directions. The natural frequencies of FE-model shows good agreement with previous field measurements reported in [1].

The FE-model of the damper, Figure 2, is assembled into hanger 5 in the global FE-model of the bridge, Figure 9. The length of hanger 5 is 6.95 m, the upper ring of the damper is located at 2.1 m from the lower end of the hanger. This is not the most optimal location, but is the same as used during later full-scale testing.

Passing trains are simulated by moving point loads directly onto the stringer beams. The dynamic simulations are performed using direct time integration and geometrical nonlinearity, to account for the increase in axial force of the hangers during train passage. A typical freight train is studied, consisting of 19 wagons and an axle load of 225 kN. Simulations are performed at different speeds, to investigate resonance in different hangers. The resulting displacement of hanger 5 when running at 90 km/h is shown in Figure 10. For the current speed, resonance is obtained for longitudinal bending. The damper is shown effective in mitigating both the forced vibrations during the train passage and the following free vibrations.

The frequency content from the displacement of hanger 5 is shown in Figure 11. In the x-direction, the unloaded frequency is 4.3 Hz and the loaded frequency 7 Hz. The damper is able to mitigate both these frequencies. In the y-direction, the unloaded frequency is 3.5 Hz and the loaded frequency 5.3 Hz. The damper is mainly able to mitigate the unloaded frequency and may be slightly detuned for the loaded frequency.
5 FULL-SCALE TESTING
The performance of the damper was further studied during in-situ tests on the case study bridge. Measurements of passing trains were recorded both with and without the damper mounted on the hanger. The installed damper is shown in Figure 12.

Figure 12. Photo of the installed damper on hanger 5.

The displacements of hanger 5 during train passages were estimated based on measured accelerations. The time response is shown in Figure 13. The displacements in the x-direction is of similar magnitude as the FE-model, but larger in the y-direction. One reason may be different train speeds or axle distances in the FE-model and the real train.

A clear tendency of vibration mitigation is found for all except the forced vibration in the x-direction. Studying the corresponding frequency content in Figure 14, it is found that the damper is slightly detuned at that frequency.

6 CONCLUSIONS
From the development of the bi-MTMD, the following conclusions are drawn.
- The developed damper was found to be relatively uncoupled, although the two masses were connected in series.
- A combination of coil springs and foam material was used to achieve the target tuning frequencies and sufficient damping. The foam contributed to both stiffness and damping.
- Good agreement was obtained when comparing the experimental results with the numerical model under various loadings.
- Simulations using a 3D FE-model of the bridge showed significant vibration mitigation when implementing the damper.
- In-situ measurements verified that the damper is able to mitigate vibrations during both forced and free vibrations in two directions simultaneously. It may however be rather sensitive to detuning.
ACKNOWLEDGMENTS

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REFERENCES