Design and Experimental Implementation of a Passive Dynamic Vibration Absorber to Attenuate Broadband Vibration

C S Chai, Y H Ko 1, *
1Faculty of Engineering and Technology (FOET), Tunku Abdul Rahman University College (TARUC), Kuala Lumpur, Malaysia
koyh@tarc.edu.my

Abstract. The challenge of designing an effective vibration absorber for any mechanical or mechatronic system is formidable. A good vibration absorber must provide broadband vibration reduction yet still keep in lightweight and size to minimize space requirements and maximize equipment accessibility. Broadband vibration is usually attenuated by using an active or semi-active vibration absorber. However, these methods are relatively costly and complicated compared with a passive vibration absorber. Moreover, passive vibration absorber often limited to narrowband frequency reduction since, in the past, it is usually tuned to the targeted frequency or optimum damping and optimum tuning ratio. This paper demonstrated that a passive vibration absorber could be implemented to reduce the broadband vibration of a system, considering the effect of the parameter of passive vibration absorber, the location attachment of the absorber, and the targeted mode shape. The result shows that a properly tuned passive vibration absorber can reduce the root mean square amplitude vibration by 60.91%, covering the first four vibration modes (0 to 300Hz).

1. Introduction

Vibration is a common cause that can seriously degrade the operation, lessen the working life, and, in some cases, lead to failure of a mechanical or mechatronics or electrical or computer system. The challenge of designing an effective vibration absorber for any mechanical or mechatronic system is formidable because a good vibration absorber has to provide broadband vibration reduction.

The solution for broadband vibration is always an active or semi-active vibration absorber. The relevant study has been well studied [1]–[4]. However, these methods are more complicated and costly than passive vibration absorber and required maintenance, and there is a higher possibility of failure. Passive vibration absorber without external sources, so it does not require a power source. In most cases, it offers the most straightforward and reasonable solution to the vibration problem. It requires little or no maintenance, and its installation is relatively simple. The typical example is a tuned mass damper consisting of a mass-spring-damper system hosted on the problematic system. However, the performance of a passive vibration absorber is limited by a specific range of frequency or narrow band. Its parameters are unable to be altered accordingly to the system [5].

Only a few researchers, such as Dayou and Brennan [6], studied broadband vibration's passive control. Their study shows that an accurately located dynamic vibration absorber (DVA) can contribute to an
excellent global vibration control for broadband with a broad frequency range. However, they used a vibration absorber that tuned to a different frequency. MTMD (Multiple Tuned Mass Dampers) is always the solution to attenuating broadband vibration by passive control. Therefore, the adoption of MTMD can improve the effectiveness of attenuation on broadband vibration. Many researchers have extensively studied the design parameters of MTMD [7]–[10]. However, passive control of broadband vibration by a single passive DVA is not well studied and explored. Nevertheless, in practice, the use of MTMD is constrained by the maximum weight installed to the primary structure.

Hence this paper demonstrated that a passive vibration absorber could be implemented to reduce broadband vibration of a system by considering the effect of the parameters of the passive vibration absorber, the location attachment of the absorber, and the targeted frequency/mode shape.

2. Methodology

2.1. Experimental Modal Analysis Setup

In order to evaluate the effectiveness of a passive vibration absorber to attenuate broadband vibration, a cantilever beam is selected as the primary system since it is a continuous system and has an infinite degree of freedom system (DOF). A 1.3kg cantilever beam with a dimension of 700mm (length) X 50mm (width) X 4mm (thickness) was used. Experimental modal analysis of cantilever (Figure 1) is carried out to identify the first four natural frequencies, and these frequencies were selected as target frequency. The experimental modal analysis is done using the impact testing method, and the measurement from each attachment point (total 14 points) is analyzed using the LMS Test Lab.

![Figure 1. Experiment Modal Analysis setup.](image)

2.2. Design of Passive Dynamic Vibration Absorber

Figure 2 shows the passive DVA made of aluminium with a weight of 0.13 kg. The DVA has a unique design mounted with a movable mass. The movable mass was bolted at the centre of the hole on the beam-like structure. As the DVA is a beam-like structure, the movable mass at a different location on the beam can change its stiffness and hence, the natural frequency can be easily tuned to the desired value.
2.3. **Tuning Parameters of DVA**

After performing the experimental modal analysis (0 to 300Hz), the first six modes of the cantilever beam is obtained. The predicted responses of a vibration absorber attached to the cantilever beam are obtained from the LMS Test Lab Modal Analysis Modification Prediction. The design parameters of the vibration absorber are the attachment point (14 nodes), the targeted frequency (first four bending modes), mass, stiffness and damping value. The mass of the vibration absorber is set to be 10% of the cantilever beam. The optimum parameters are determined by comparing the reduction of overall vibration level over the first four bending natural frequencies.

3. **Results And Discussion**

3.1. **Experiment Modal Analysis of Cantilever Beam**

The mode shape of the cantilever beam are obtained from experimental modal analysis (Figure 3). The mode shapes for the 1st, 2nd, 4th and 6th natural frequency are the bending modes of the cantilever beam, whereas the 3rd and 5th natural frequency are torsional modes (Table 1). The objective of the DVA is only focusing on attenuating bending moment natural frequencies of the cantilever beam at the Z direction of the cantilever beam. Therefore, the attenuation will only focus on the 1st, 2nd, 4th and 6th mode of the natural frequency.

| Mod e | Natural frequencie s (Hz) | Damping coefficien t (%) | Mode type |
|-------|--------------------------|--------------------------|-----------|
| 1     | 6.838                    | 0.38                     | Bending   |
| 2     | 42.668                   | 0.69                     | Bending   |
| 3     | 86.197                   | 1.95                     | Bending   |
| 4     | 125.791                  | 0.23                     | Torsion   |
| 5     | 184.845                  | 0.29                     | Torsion   |
| 6     | 242.283                  | 0.26                     | Bending   |

**Table 1.** Natural frequencies of cantilever beam obtained from impact testing.
3.2. Vibration Analysis

The Root Mean Square (RMS) is used to measure the overall broadband vibration reduction. The RMS amplitude of vibration indicates the overall energy of the vibration. The higher the vibration energy, the greater the RMS amplitude and vice versa. From Figure 3, it is calculated that the original RMS vibration of the cantilever beam was 3.2208 g/N. The LMS Test Lab Modal Analysis Modification Prediction had carried out to analyze the effect of DVA on a different natural frequency, damping coefficient and attached at different locations on the cantilever beam. It is interesting to discover which condition of the DVA can produce optimal vibration attenuation to the natural frequencies of the cantilever beam. The RMS value of the cantilever beam after the DVA was attached to it is presented in Table 2.

![Figure 4. Fourteen attachment points of the cantilever beam.](image)

| Point | Natural Frequency of DVA | Mode Shape 1 (6.838 Hz) | Mode Shape 2 (42.741 Hz) | Mode Shape 3 (86.236 Hz) | Mode Shape 4 (125.838) |
|-------|--------------------------|-------------------------|--------------------------|--------------------------|-------------------------|
| 2     |                          | 6.7105                  | 4.1174                   | 3.5923                   | 4.0770                  |
| 5     |                          | 1.0543                  | 1.0884                   | 1.4572                   | 1.4395                  |
| 8     |                          | 1.4648                  | 1.2986                   | 1.3725                   | 0.6032                  |
| 11    |                          | 3.0797                  | 4.0362                   | 1.4570                   | 1.0874                  |
| 14    |                          | 2.3175                  | 1.8976                   | 1.1085                   | 7.6460                  |
| 17    |                          | 1.3248                  | 1.8460                   | 1.5516                   | 6.6766                  |
| 20    |                          | 2.0439                  | 1.3835                   | 1.1825                   | 0.6142                  |
| 23    |                          | 3.5198                  | 1.9254                   | 8.9903                   | 0.6510                  |
| 26    |                          | 2.0646                  | 1.3121                   | 0.9368                   | 0.8150                  |
| 29    |                          | 2.0345                  | 1.2679                   | 0.5661                   | 6.1606                  |
| 32    |                          | 3.0412                  | 1.6293                   | 0.9439                   | 0.8983                  |
| 35    |                          | 4.7779                  | 2.4859                   | 1.2130                   | 0.7000                  |
| 38    |                          | 3.0826                  | 2.0684                   | 1.0325                   | 0.4993                  |
| 41    |                          | 1.3738                  | 1.3667                   | 1.3803                   | 0.9478                  |

Green cells indicate that the RMS amplitude of the cantilever beam was reduced, whereas red cells indicate that the RMS amplitude of the cantilever beam was amplified as it is well known that for a
continuous structure such as a cantilever beam, suppressing the vibration amplitude at a point may increase the amplitude at some other points [11].

Figure 5 shows the RMS amplitude of the cantilever beam for the different condition of DVA. It can notice that the different targeted mode shape of the DVA will have a different effect on the broadband vibration control. In a specific condition, the cantilever beam’s RMS amplitude can be higher than the original RMS amplitude of the cantilever beam. By comparing the predicted overall RMS responses, it is found that the optimum targeted mode is the second mode (42.741Hz) and the best location is point 5. Hence, a DVA is experimentally designed to mount at location point 5 with a target frequency of 43Hz (k=240 N/m) to validate the effect on the broadband attenuation of the cantilever beam.

Figure 6 showed the experimental FRF of the cantilever beam when DVA was attached to point 5 with the target frequency of 42.741Hz. From Figure 6, it can be visually compared the effect of mounting a DVA to the cantilever beam. It shows that the original FRF of the cantilever beam was reduced significantly compared to the experimental data and the predicted data. It should be noted that the predicted FRF of the cantilever beam is used as a benchmark to evaluate the performance of the designed DVA in attenuate the broadband vibration of the cantilever beam. It is noticed that the vibration amplitude at the other mode was also decreased. This show that the DVA has absorbed the vibration at the other frequencies as well.

The first four bending natural frequency of the cantilever beam with the amplitude of 4.7420 g/N, 6.0968 g/N, 23.9034 g/N and 26.9985 g/N are attenuated by 44.27%, 67.87%, 71.86% and 83.31%, respectively. It can be noticed that when the DVA has attached to the cantilever beam, the natural frequencies of the cantilever beam were shifted away from the original position. The RMS amplitude for the original FRF, predicted FRF and the experiment FRF were presented in Table 3.

Table 3. RMS amplitude without DVA and with DVA on the cantilever beam.

| Data               | RMS amplitude (g/N) | Reduction (%) |
|--------------------|---------------------|---------------|
| Without DVA        | 3.2209              | -             |
| With DVA (Predicted) | 0.8235              | 74.43%        |
| With DVA (Experiment) | 1.2592              | 60.91%        |

Table 3 shows that the RMS amplitude of the cantilever beam without DVA is 3.2209 g/N. It can be seen that the RMS amplitude of the cantilever beam was reduced significantly after the DVA attached to the cantilever beam at point 5. The RMS amplitude of the cantilever beam with DVA (predicted) is 0.8235 g/N, whereas the RMS amplitude of the cantilever beam with DVA (Experiment) is 1.2592 g/N, which shows a reduction in amplitude of 74.43% and 60.91%, respectively.

By comparing the experiment to the predicted, it can note that the RMS amplitude of the experiment is slightly higher. Although there are slight differences compared to the predictions, the results are still in reasonable agreement. Figure 7 shows a better insight into the experimental data and the predicted data.
From Figure 7, the peak amplitude of the natural frequencies of the cantilever beam with DVA (experiment) is higher than the peak amplitude of the cantilever beam with DVA (Predicted). Besides that, the parameters of the fabricated DVA do not perfectly match the desired parameter in the prediction. Therefore, the result of the FRF of the cantilever beam was slightly different. However, the result of the experimental DVA is still in close agreement with the predicted result.

Figure 6 and Figure 7 show that the passive DVA can attenuate broadband vibration as effective as an active DVA if the DVA is properly tuned. It has been proven that it is possible to get the optimum tuning of a DVA for broadband vibration of a continuous structure.

4. Conclusion

Experiments showed that the DVA with optimal tuning could attenuate the broadband vibration of a continuous structure (first four modes), which covers from 0-300Hz. The RMS amplitude of the cantilever beam was attenuated from 3.2209 g/N to 1.2592 g/N (60.91%) when the DVA was attached to the cantilever beam.
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**References**

[1] Ladipo I L and Muthalif A G A 2013 Active Dynamic Vibration Absorber for Broadband Control of a Multi-Mode System: Simulation and Experimental Verification *J. Low Freq. Noise, Vib. Act. Control*, vol. 31, no. 3

[2] Casadei F, Ruzzene M, Dozio L, and Cunefare K A 2010 Broadband vibration control through periodic arrays of resonant shunts: Experimental investigation on plates *Smart Mater. Struct.*, vol. 19

[3] Komatsuzaki T, Inoue T and Terashima O 2016 Broadband vibration control of a structure by using a magnetorheological elastomer-based tuned dynamic absorber *Mechatronics, vol. 40*

[4] Xu Z, Gong X, and Chen X 2011 Development of a mechanical semi-active vibration absorber *Adv. Vib. Eng.*, vol. 10, no. 3

[5] Elias S and Matsagar V 2017 Research developments in vibration control of structures using passive tuned mass dampers *Annu. Rev. Control*, vol. 44

[6] Dayou J and Brennan M J 2003 Experimental verification of the optimal tuning of a tunable vibration neutralizer for global vibration control, *Appl. Acoust.*, vol. 64, no. 3

[7] Rade D A and Steffen V 2000 Optimization of dynamic vibration absorbers over a frequency band *Mech. Syst. Signal Process.*, vol. 14, no. 5

[8] Lee C L, Chen Y T, Chung L L, and Wang L P 2006 Optimal design theories and applications of tuned mass dampers *Eng. Struct.*, vol. 28, no. 1

[9] Jacquot R G 2001 Suppression of random vibration in plates using vibration absorbers *J. Sound Vib.*, vol. 248, no. 4

[10] Li C 2000 Performance of multiple tuned mass dampers for attenuating undesirable oscillations of structures under the ground acceleration *Earthq. Eng. Struct. Dyn.*, vol. 29, no. 9

[11] Dayou J 1999 Global control of flexural vibration of a one dimensional structure using tunable vibration neutralizers *Thesis University of Southampton*