VOL. 11, NO. 8, APRIL 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

NEW DESIGN OF TUNED VIBRATION ABSORBER FOR WIDE FREQUENCY RANGE APPLICATION

Mohd Hafiz Ghazali[1] and Izzuddin Zaman[1, 2]
[1]Structural Integrity and Monitoring Research Group
[2]Faculty of Mechanical and Manufacturing Engineering, Universiti Tun Hussein Onn Malaysia, Batu Pahat, Johor, Malaysia
E-Mail: hafizghazali88@gmail.com

ABSTRACT

Uncontrolled vibrations can leave a bad impression to the machine, structure, and human. For example, vibration on machine can damage the equipment, decrease the machine lifetime and also causing the safety factor problems. Therefore, a vibration absorber is obliged to reduce these vibrations. The present paper investigated a new design of tuned vibration absorber (TVA). The proposed TVA is: (1) light in weight and small in scale, which suitable for mobility purpose, and (2) can addressed a broad frequency range of application. However in this paper, the effectiveness of new design of TVA to reduce the vibration is not covered since the TVA has been proved in the previous study being able to reduce the vibration significantly. The frequency range of absorber was determined through finite element analysis (FEA) and validated with the experimental result. The aim is to enhance the range of frequency that TVA can tune from 0 to 1000 Hz. In order to generate the result, SolidWorks® software was used in the finite element analysis and DEWEsoft equipment was used in experiment. The results in the finite element analysis showed that the maximum frequency that TVA can tune is 800 Hz while from experiment is 980 Hz in the experimental analysis. Although there is an error about 18% between FEA and experiment, the findings are still significant because the maximum frequency of the new design TVA can reach is approximate to 1000 Hz. This study concludes that the new design of TVA which is small in scale and light in weight is able to reduce the structural vibration extensively if it is tuned correctly to the targeted frequency range of 0 to 980 Hz.

Keywords: finite element method, tuned vibration absorber, frequency, mode shape.

INTRODUCTION

The vibrations are of much importance branches of engineering and a major concern in most engineering fields. Super harmonic resonance may turn to be a hazard that reduces the fatigue life of the structure. Excessive vibration also can lead to structural failure (Zaman *et al*., 2013a) and induce uncomfortable noise which ultimately can cause human discomfort and muscular skeletal pain (Bosco *et al*., 1999). This is the intention why a proper vibration control method is required to address the structural vibration problems.

The idea of dynamic vibration absorber (DVA) emerges due to a common vibration problem existed in engineering structural applications (Sun *et al*., 1995). The dynamic vibration absorber (DVA) have been widely used to attenuate harmonic excitation structure (Carneal *et al*., 2004), (Acar and Yilmaz, 2013), (Zaman *et al.,* 2014a), (Zaman *et al*., 2014b). The DVA commonly consist of a mechanical appendages comprising mass, stiffness, damping elements system which attached to the vibrating structure. Applications of these devices can be found in bridges, civil structures, machine tools and others engineering systems (Wright and Kidner, 2004), (Wong and Cheung, 2008), (Weber and Feltrin, 2010), (Beltrán-Carbajal and Silva-Navarro, 2014).

The DVA normally used to address harmonic excitation with a single frequency targeted harmonic frequency (Brennan, 2006). Advantages of DVA are found to be an efficient, reliable, and low-cost suppression device for vibrations caused by harmonic or narrow-band excitations. However, it produces drawbacks such as the obvious weight increase, large in size, and improperly placement and frequency tuning of vibration absorber may result in large increase of vibration level (Sun *et al*., 1995). Therefore, in this paper a new design of TVA is developed by addressing previous drawbacks of TVA. The new design of TVA will include these criteria, such as: (1) light in weight and small in scale, which suitable for mobility purpose, and (2) can addressed a broad frequency range of application. A finite element method and experimental testing were carried out in the study in order to compare the frequency range obtained by a new design of TVA.

METHODOLOGY

SolidWorks® 2014 was used to model the TVA. It is also used to perform the simulation analysis. SolidWorks® is a powerful finite element modelling package for numerically solving a large variety of mechanical problem. In this study, frequency analysis was selected in order to determine the natural frequency and the mode shape of design tuned vibration absorber (Zaman *et al*., 2013b).

During the literature review stage, all the information about the previous studies were collected such as related mechanism, patent search, available product and researchers study. Thereafter, the design concept was developed by drawing through computer aided design (CAD) 3D modelling software by using SolidWorks® 2014. The proposed software is then simulated in CAD software to determine the design workability. Furthermore the suitable materials are selected for every component of the vibration absorber component with recommendation of manufacturing process and analyzing the safety factor.

VOL. 11, NO. 8, APRIL 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

After the design process completed, the author proceed to the fabrication phase. In this phase, the components of the prototype are fabricated using the dimension and material that had been determined earlier in the project. Then, the project undergoes simulation approach using finite element method. After finished, the projects went to the experimental phase, in which the vibration test was performed. The TVA placed with a horizontal manner on the structure and fastened by using bolt and nut. An accelerometer sensor attached at each of TVA flyers in order to obtain the data while conducting the experiment. The TVA undergoes several testing to ensure it passed the minimum requirement that author had determined in the design process. All these experiment are done several times to get the average result. The result gained has been analyzed and proceed with the validation process.

Since all the validation process has been done, the data are then analyzed to measure the characteristics of TVA, such as its frequency and mode shape.

Design of vibration absorber

There are many types of vibration absorber that are used to reduce the vibration such as spring-mass-damper, damped dynamic vibration absorber, passive vibration absorber and many others. Figure-1 shows the new design of tuned vibration absorber. For this project, one design model has been created to reduce the vibration.

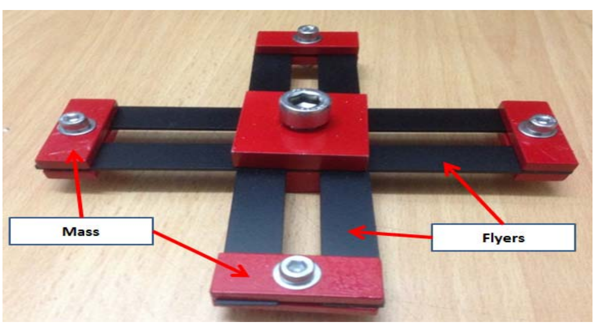

Figure-1. New design of tuned vibration absorber.

The concept design of the vibration absorber model consists of T-shape with four mass attached each on the wing. All the mass are adjustable and it can be only tuned along one direction to get the desired frequency. The function of the flyers is to support all the masses. The design of TVA has included the criteria needed as mentioned before, such as light in weight and small in scale. Based on these criteria, the dimensions of the vibration absorber are 160 x 160 mm. Material employed for the model are steel AISI 1075, in which subsequently identify the mass of TVA as 620 g.

FEM simulation results

In this section, FEM was performed by using frequency analysis mode in order to find its natural frequency and mode shape. The model was fixed at the bottom of the absorber. The frequency analysis of the vibration absorber was carried out using Solidworks® in the frequency range of 1-1000 Hz.

Figure-2 shows the meshed model of vibration absorber produced in Solidworks®. This model is meshed using standard solid mesh, which generates total nodes and elements of 18582 and 10473, respectively. From the analysis, it shows that the maximum range of natural frequency obtained was about 800 Hz as can be seen in Figure 3. The mode shape of TVA during this frequency is found to be in 1st bending mode of flyers. It is important to determine these frequencies in order to acquire the position of side masses for matching later with the host structure's resonance frequency.

Figure-2. FEM meshed model of TVA.

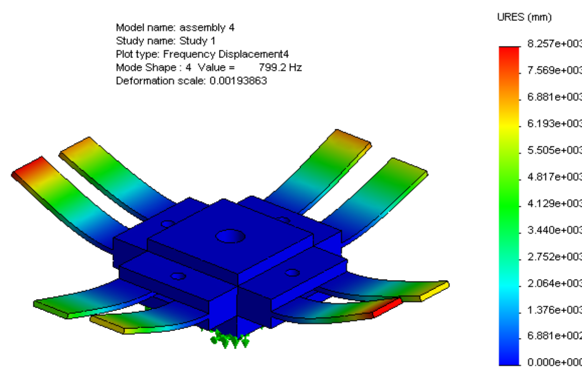

Figure-3. FEM simulation analysis of TVA.

Table-1 shows the natural frequency of the vibration absorber when all the masses were tuned simultaneously from 0 mm to 40 distance from the center of absorber to the edge of flyer. From this result, it shows that the natural frequency increases when the masses moving near to the center. The farthest the masses, the lowest frequency range of TVA can address as been proved in our previous study (Zaman *et al.*, 2014c). Figure-4 plots the frequency trend of TVA when the masses is moved from the center of absorber to edge.

ARPN Journal of Engineering and Applied Sciences

Table-1. FEA frequency analysis of TVA.

Mass distance (mm)	**FEA Frequency** (Hz)
0	800
10	509
20	470
30	280
40	218

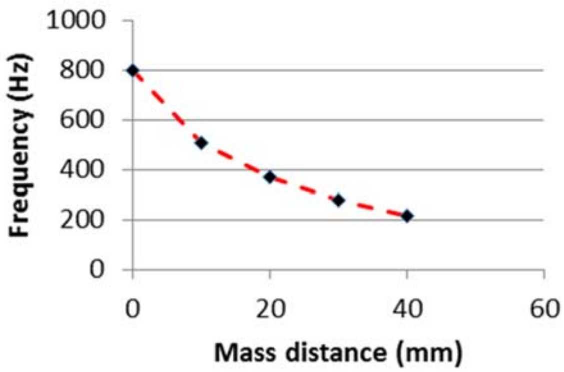

Figure-4. Frequency trend of TVA against the mass distance.

Experimental results

The analysis result from the experimental works was conducted to determine the vibration characteristic of the absorber, such as natural frequency. Similarly as FEM, the tunable mass can be shifted along the axis and the experimental works done several times with a different mass distance. However, in this experimental work, the TVA was attached to the structure and fastened by using bolt and nuts. Figure-5 shows the TVA apparatus setup for the experimental work. There were four accelerometer sensors used, where two were placed at absorber flyers and another two at the plate structure using a wax. The use of this accelerometer sensor is to read the vibration and converting the result into graph.

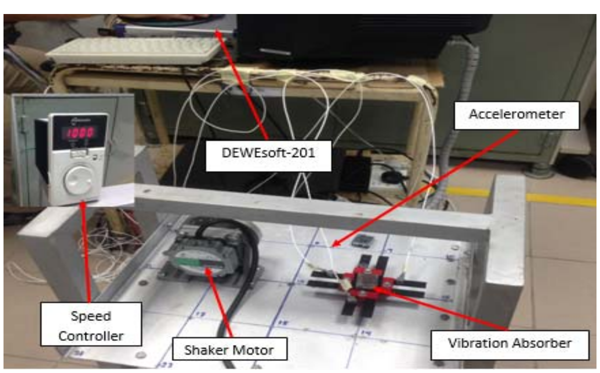

Figure-5. TVA's experimental process.

In this study, there were five distances used to identify the frequency and each distance gave a different frequency value. Figures 6 to 10 show the experimental analysis results. The analysis is started from 0 mm (mass distance from center of TVA) to 40 mm and it was done in sequent. These TVA experimental results data are then tabulated in Table-2.

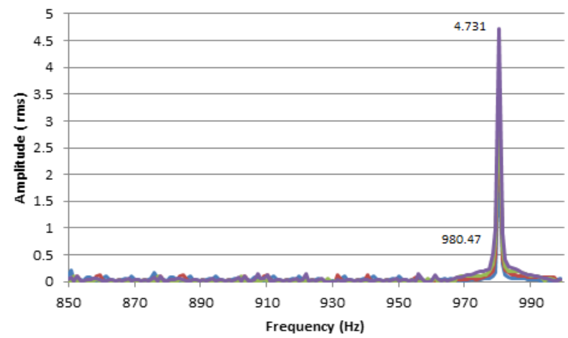

Figure-6. 0 mm mass distance.

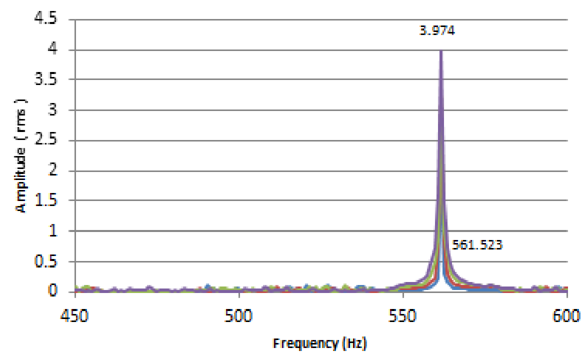

Figure-7. 10 mm mass distance

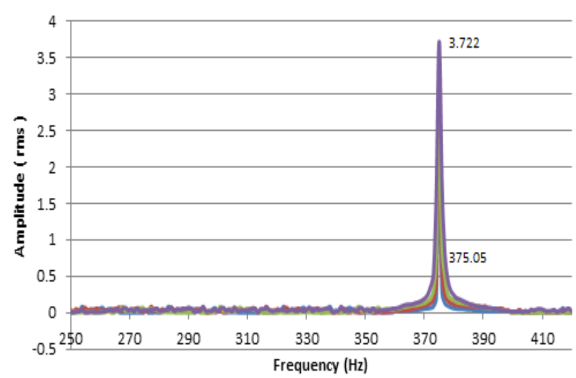

Figure-8. 20 mm mass distance.

VOL. 11, NO. 8, APRIL 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

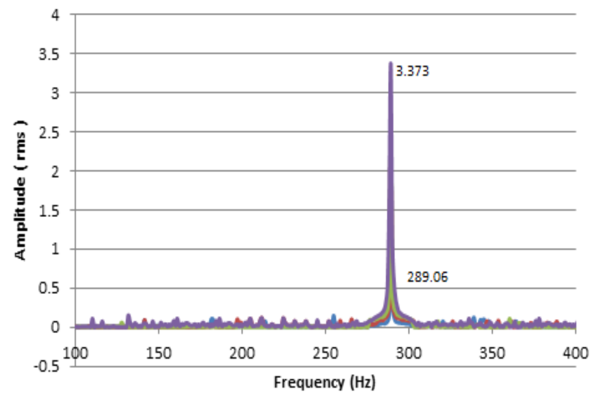

Figure-9. 30 mm mass distance.

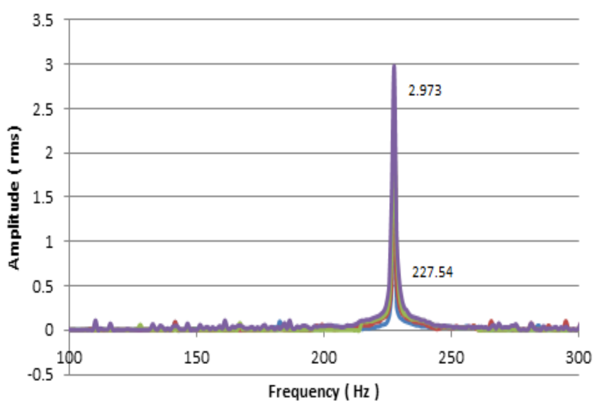

Figure-10. 40 mm mass distance.

Table-2. Experimental frequency analysis of TVA.

Mass distance (mm)	**Experimental Frequency** (Hz)
0	980
10	562
20	375
30	289
40	228

Theoretically, the experimental data results should match with one of the finite element analysis results (Zaman *et al*., 2007). In this case, all the frequency data obtained from the experiment are almost similar or even higher than finite element analysis result in particular at mass distance of 0 and 10 mm. Table-3 shows the comparison of finite element analysis and experimental results with the average error difference ± 7%.

Table-3. FEA and experimental result comparison.

Mass distance (mm)	**FEA Frequency** (Hz)	**Experimental Frequency** (Hz)	**Error** (%)
0	800	980	18.4
10	509	562	9.4
20	470	375	1.3
30	280	289	3.1
40	218	228	4.5

CONCLUSIONS

In this paper, finite element simulation analysis has been used to study the vibration characteristic of the new tuned vibration absorber, such as its natural frequency and mode shape. A new design of vibration absorber, namely the tuned vibration absorber (TVA), has been proposed to reduced the vibration on a vibrating structure by take consideration of light in weight and small in size scale. The design are then fabricated and has been validated with the experimental results. From the results, the maximum frequency that the TVA can tune in finite element simulation analysis is 800 Hz and compared to experiment is 980 Hz. Overall, the finite element simulation analysis shows slightly lower range of natural frequency compared to the experimental analysis value and it was ± 7% average of error difference. Although there is a slight discrepancy between FEA and experiment, the findings are still significant. The result revealed that the newly design of TVA is able to tackle the broad range of vibration frequency from 200 to 980 Hz by not compromising its size and weight.

ACKNOWLEDGEMENT

The authors thank Office for Research, Innovation, Commercialization and Consultancy Management and Universiti Tun Hussein Onn Malaysia for the financial support under Incentive Grant Scheme for Publication (IGSP) vote U247.

REFERENCES

Acar, M. A. and Yilmaz, C. 2013. Design of an adaptive–passive dynamic vibration absorber composed of a string–mass system equipped with negative stiffness tension adjusting mechanism, Journal of Sound and Vibration, 332, pp.231-245.

Beltrán-Carbajal, F. and Silva-Navarro, G. 2014. Active vibration control in Duffing mechanical systems using dynamic vibration absorbers. Journal of Sound and Vibration, 333(14), pp.3019–3030.

Bosco, C., Cardinale, M., Colli, R., Tihanyi, J., von Duvillard, S. P. and Viru, A. 1999. The influence of whole body vibration on the mechanical behavior of skeletal muscle, Clinical Physiology, 19, pp.183-187.

ARPN Journal of Engineering and Applied Sciences

Brennan, M. J. 2006. Some recent developments in adaptive tuned vibration absorbers/neutraliser, Shock and Vibration, 13, pp.531-543.

Carneal, J. P., Charette, F. and Fuller, C. R. 2004. Minimization of sound radiation from plates using adaptive tuned vibration absorbers, Journal of Sound and Vibration, 270, pp.781-792.

Sun, J. Q., Jolly, M. R. and Norris, M. A. 1995. Passive, adaptive and active tuned vibration absorbers - a survey, Transactions of the American Society of Mechanical Engineers, 117, pp.234-242.

Weber, B. and Feltrin, G. 2010. Assessment of long-term behavior of tuned mass dampers by system identification. Engineering Structures, 32(11), pp. 3670-3682.

Wong, W.O. and Cheung, Y.L. 2008. Optimal design of a damped dynamic vibration absorber for vibration control of structure excited by ground motion. Engineering Structures, 30, pp.282–286.

Wright, R. I. and Kidner, M. R. F. 2004. Vibration absorbers: a review of applications in interior noise control of propeller aircraft, Journal of Vibration and Control, 10, pp. 1221-1237.

Zaman, I., Rahman, R. A., Khalid, H. 2007. Dynamic analysis on off-road vchicle chassis using 3-d finite element model, Journal of Science and Technology, 4, pp.1-16.

Zaman, I., Salleh, M. M., Manshoor, B., Khalid, A. and Araby, S. 2014. The application of multiple vibration neutralizers for vibration control in aircraft, Applied Mechanics and Materials, 629, pp.191-196.

Zaman, I., Khalid, A., Manshoor, B., Araby, S. and Ghazali, M. I. 2013. The effects of bolted joints on dynamic response of structures, IOP Conference Series: Materials Science and Engineering, 50(1), pp. 012018.

Zaman, I., Manshoor, B., Khalid, A., Araby, S. and Ghazali, M. I. 2013. Vibration characteristics of composite plate embedded with shape memory alloy at elevated temperature, Applied Mechanics and Materials, 393(1), pp.655-660.

Zaman, I., Salleh, M. M., Ismon, M., Manshoor, B., Khalid, A., Sani, M. S. M. and Araby, S. 2014. Vibration attenuation of plate using multiple vibration absorbers, MATEC Web of Conferences, 13, pp.03003.

Zaman, I., Salleh, M. M., Ismon, M., Manshoor, B., Khalid, A., Sani, M. S. M. and Araby, S. 2014. Study of passive vibration absorbers attached on beam structure, Applied Mechanics and Materials, 660, pp. 511-515.