ASM Sc. J., 12, Special Issue 3, 2019 for ICST2018, 48 - 52

Design of a Passive Vibration Absorber to Attenuate Wide Band Vibration

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Active dynamic vibration absorber or semi-active vibration absorber is widely used to control beam vibration (wide band vibration). The application of passive vibration absorber to control beam vibration is limited. This paper aims to design a passive vibration absorber to attenuate the first four modes of vibration by tuning the absorber's mass, damping, stiffness, targeted frequency and attachment location. In this research, experimental modal analysis of a continuous system (cantilever beam) is carried out and the optimum parameters of a passive vibration absorber attached to the primary system is determined through LMS Test Lab. The optimally tuned passive vibration absorber had successfully reduced the peak amplitude for 62% at first mode, 87% at second mode, 92% at third mode and 76% at fourth mode. It showed the possibility of a passive vibration absorber to attenuate beam vibration by optimally tuned the parameters as suggested by the authors.

Keywords: passive vibration absorber; beam vibration; wide band vibration control

I. INTRODUCTION

Passive vibration absorber is found to be effective to reduce narrow band vibration (Yang *et al.*, 2010). Narrow-band vibration control can be done by varying basic parameters (damping value, stiffness and mass) of a passive vibration absorber. However wide band vibration control can only be achieved by using dynamic vibration absorber (DVA) which is active or semi-active control. Bertran and Montoro (1998) stated that an active control vibration absorber to reduce the vibrations consist of generating a signal that must produce in the machine, the signal is known as counteracting signal is used to stabilize the vibrating machine. For the semi active control, its properties or parameters, such as stiffness, damping and etc. can be altered in real time with a low power input. The reason of naming them semi-active vibration absorber is because they do have any feedback path which makes them unable to destabilize the system. Sun *et al.*, (2016) designed an absorber which adopt the multilayered magnetorheological elastomer (MRE) structure to expand the effective bandwidth and reduced multi-frequency vibration. MRE is a smart material whose stiffness can be controlled by external magnetic field, has been widely used to develop adaptive tuned vibration absorber. Moreover, according to the work of (Flatau *et al.,* 2000), the tunable vibration absorber that achieved wide band vibration control is consists of Terfenol-D core which have the magneto elastic properties that able to make changes in its elastic modulus with magnetic bias which the modulus can vary by over 160%.

There are also some researches which were study on the passive control in wide band vibration control. Howard et al. (2004) used a large number of DVAs and optimized

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their parameters to perform wide band vibration control. Using the simplex algorithm to minimize the peak of the frequency response over a certain frequency range, Rice (1993) also obtained the optimal parameters (stiffness and damping) of two single degree of freedom system vibration absorber for a cantilever beam. Yang *et al.* (2010) achieved a broadband vibration reduction by using multiple DVAs. They determined the optimal numbers of DVAs and found that the interaction between the plate and the absorber by means of the reaction force from the absorber plays a dominant role in a narrow band control, while in a relatively broadband control the dissipation by the absorber damping governs the control performance.

Other than that, Mahadevaswamy & Suresh (2016) stated the importance of the optimal mass ratio for a rectangular plate toward wide band vibration control. Although in their study, the best optimal mass ratio for DVA is 7%, the mass ratio of the DVA seems to be a constraint for the perspective of the industrial used. High mass of DVA may help to reduce further vibration but it might affect the system.

Passive vibration absorber is used to control narrow band vibration only as it is not effective in a wide band vibration control. To control wide band vibration, multiple passive vibration absorbers is always the solution. However, no study has been carried out on the application of single passive vibration control to reduce the wide band vibration. Hence, this study is looking for the possibilities in reducing wide band vibration by using a single passive vibration absorber.

II. MATERIAL AND METHODS

A. *Experimental modal analysis of continuous system*

In order to evaluate the effectiveness of a passive vibration absorber to attenuate wide band vibration, a cantilever beam is selected as primary system since it is a continuous system and has infinite number of degree of freedom system (DOF). The dimension of the cantilever beam is 0.5m (Length) x 0.05m (Width) x 0.004m (Height). Experimental modal analysis of this cantilever (Figure 1) is carried out to identify the first four natural frequencies and these frequencies will be selected as target frequency. The experimental modal analysis is done by using impact testing method, and the measurement from each attachment points (total 22 nodes) is analyzed by using LMS Test Lab.

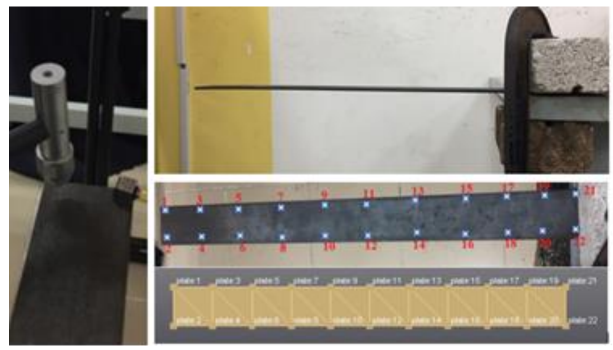

Figure 1. Cantilever Beam Experimental Modal Analysis Setup (impact testing and the absorber's

B. Tuning Parameters of DVA

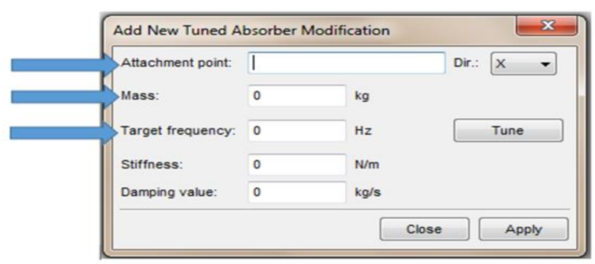

Figure 2. LMS Test Lab Modal Analysis Modification Prediction to simulate the attachment of vibration absorber on experimental modal analysis results

After performing the experimental modal analysis, the first four modes of cantilever beam can be calculated. Each mode has a mass and stiffness matrix that can be modified. The predicted responses of a vibration absorber attached to the cantilever beam are obtained from the LMS Test Lab Modal Analysis Modification Prediction (Figure 2). The design parameters of vibration absorber are the attachment point (22 nodes), the targeted frequency (mode 1 to mode 4), mass, stiffness and damping value. The mass of the vibration absorber is set to be 5% of the cantilever beam. After creating a group of modifications, a new set of mode shapes and modal frequencies is calculated that incorporates the changes. The optimum parameter is determined by comparing the reduction of overall vibration level over the first four natural frequencies.

IV. RESULTS AND DISCUSSION

The first five natural frequencies (modes) of cantilever beam are identified (Figure 3) which is 12.417 Hz (1st bending mode) with the damping of 0.57%, 83.434 Hz (2nd bending mode) with the damping of 0.63%, 226.776 Hz (3rd bending mode) with the damping of 0.72% and 444.819 Hz (4th bending mode) with the damping of 0.50%. While 257.776 Hz is identified as torsional mode and hence it is excluded in this analysis.

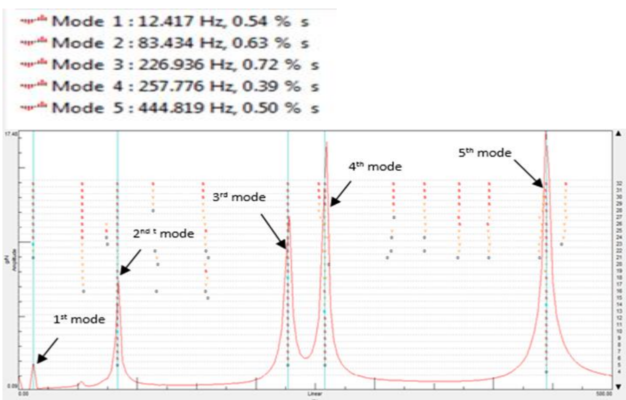

Figure 3. First five modes of vibration obtained from experimental modal analysis

The effectiveness of the passive vibration absorber attached to the cantilever beam is evaluated by comparing the predicted response produced LMS Test Lab Modal Analysis Modification Prediction and Root Mean Square (RMS). The relationship between the RMS and the vibration reduction is inversely proportional, the smaller the RMS value, the greater the reduction of the overall amplitude of the cantilever beams after attaching the tuned passive vibration absorber.

Results (Figure 4) showed that DVA attachment points and tuned mode play an important role in the reduction of vibration. Some attachment locations were unable to reduce vibration and caused overshoot at the targeted mode. For example, A DVA tuned to fourth mode (444.819 Hz) and attached at Point 16 amplified the response at mode 2 (83.434Hz). The same DVA attached at Point 17 and Point 19 amplified the response at mode 3 (226.936Hz). It shows not all the location that DVA attached at is able to reduce vibration. Some of the location which DVA attached might produce overshoot at the target frequencies or at other frequencies.

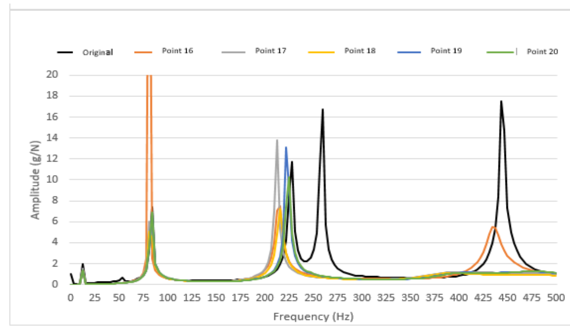

Figure 4. The responses of DVA tuned at fourth mode and different attachment point (point 16 to point 20)

Attachment point 1 or point 2 (near the extreme free end of the beam) is the best location according to the result. Other than that, the results also showed that the 3rd and 4th mode are the best target frequencies to tune a passive vibration absorber for better vibration control. The reason of the 3rd and 4th mode are the best frequency for the tuning purpose is because the stiffness and the damping value after tuned will be relatively higher which is more suitable to suppress the vibration. At the 3rd mode, it will be the optimum target frequency because it lying between 2nd and 4th mode which when tuning according to the 3rd mode. The amplitude peaks around the 3rd mode will be suppressed or the natural frequencies changed to other frequencies.

After the comparison between the all the parameters of passive vibration absorber, absorber that attached at Point 1 and tuned at targeted 3rd mode was the best parameters for improving the bandwidth of the vibration control (Table 1).

Table 2 showed that the optimum tuned passive vibration absorber able to achieve more that 60% of the reduction for all the targeted modes of vibration. Besides, from the predicted responses produced by LMS Test Lab Modal Analysis Modification Prediction (Figure 5), it showed that the optimum tuned vibration absorber has successfully to reduce wide band vibration.

Table 1. Specifications of the optimum tuned passive vibration absorber

Parameters	Value
Attachment Point of DVA (Location)	1 (extreme free end of the beam)
Target Frequency	3rd mode shape (226.936 Hz)
Mass	40.25 g
Stiffness	62432.3 N/m
Damping Value	19.0826 kg/s

Table 2. The amplitudes generated after attaching optimum tuned passive vibration absorber and its vibration reduction at the four modes of vibration

	Original Amplitude (g/N)	With optimum tuned passive vibration absorber (g/N)	Vibration reduction (%)
1st Mode (12.417 Hz)	1.920644	0.713453	62.85
2nd Mode (83.434 Hz)	7.416936	0.964901	86.99
3rd Mode (226.936 Hz)	8.417986	0.595885	92.92
4th Mode (444.819 Hz)	17.47523	4.195079	75.99

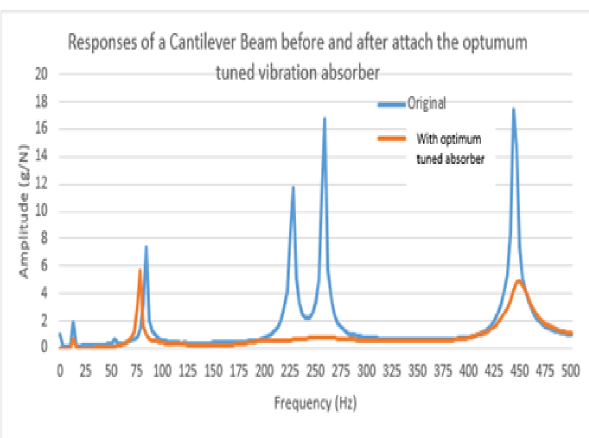

Figure 5. The responses of cantilever beam before and after attachment of optimum tuned vibration absorber

V. CONCLUSION

Wide band vibration was reduced by using a passive vibration absorber. The passive vibration absorber had successfully reduced the peak amplitude for 62% at first mode, 87% at second mode, 92% at third mode and 76% at fourth mode. The overall reduction of vibration also can be reached up to 60% by comparing the RMS values (g/N) of the original response and the predicted response. The parameters of passive vibration absorber, such as location of attachment and the target frequency were identified as the important factors for wide band vibration design and the effect of the parameters toward the bandwidth of vibration control had been analyzed. The best attachment location of vibration absorber in this project is in Point 1 and 2 and the best target frequency to tune the vibration absorber are the 3rd and 4th mode shape frequencies. The parameters of passive vibration absorber that provide the best vibration reduction response (wide band vibration) was identified.

VI. ACKNOWLEDGEMENTS

The support from Tunku Abdul Rahman University College (TAR UC) is highly appreciated.

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