VOL. 11, NO. 4, FEBRUARY 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

FINITE ELEMENT MODELLING OF FIXED-FIXED END PLATE ATTACHED WITH VIBRATION ABSORBER

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ABSTRACT
The present paper investigated the effect of the lightweight dynamic vibration absorber (LDVA) to reduce vibration of thin walled structure. The free and forced vibration response of a rectangular thin plate were performed using finite element method. Subsequently, the effects of attached single and dual LDVA were analysed in depth by using Ansys workbench 14.5. Results demonstrated that single LDVA attached at the centre of the plate succesfully attenuate vibration over the frequency range of 0- 600 Hz. By contrast, attached with dual LDVA only suppresses the resonance of the first second and fourth modes but not for third and fifth modes of thin walled structure. It was found that by simply increasing the weight of mass does not improve the vibration absorption over the entire frequency range. The study conclude that attached single LDVA are better than dual LDVA for vibration absorption of thin walled structure over the entire frequency range.

Keywords: finite element method, lightweight dynamic vibration absorber.

INTRODUCTION
The vibration are of much importance branches of engineering and a major concern in most engineering fields (Ross, 1996). Super harmonic resonance may turned to be a hazard that reduces the fatigue life of the structure. Excessive vibration also can lead structural failure (Zaman *et al.* 2013) and induce uncomfortable noise which ultimately can cause human discomfort and muscular skeletal pain (Bosco *et al.* 1998). 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 (Den Hartog, 1956), (Thomson 1981). The dynamic vibration absorber (DVA) have been widely used to attenuate harmonic excitation structure (Sun *et al.* 2010), (Kojima and Saito, 1983), (Acar and Yilmaz, 2013), (Zaman *et al.* 2014). The DVA commonly consist of a mechanical appendages comprising mass, stiffness, damping elements system which attached to the vibrating structure (Steffen Jr V, 2001), (Lee *et al.* 2006). Applications of these devices can be found in bridges, civil structures, machine tools and others engineering systems (Behady, 2012), (Beltrán-Carbajal and Silva-Navarro, 2014), (Weber and Feltrin, 2010), (Yang *et al.* 2010).

The DVA normally used to address harmonic excitation with a single frequency targeted harmonic frequency (Sun *et al.* 2008), (Beltrán-Carbajal and Silva-Navarro, 2014). Advantages of DVA is found to be an efficient, reliable, and low-cost suppression device for vibrations caused by harmonic or narrow-band excitations (Wong & Cheung 2008; Wong *et al.* 2007).

In this paper, a new design of LDVA will be analyzed in term of its ability to reduce the vibration of thin walled structure. Finite element method was carried out to compare vibration response of a single and two LDVAs applied to a plate structure. This paper also aims to tackle multi frequency harmonic band when control LDVA attached to the thin walled structure.

FINITE ELEMENT MODELS
Solidwork 2012 was used to model the plate and DVA. With the Solidwork 2012 in hand, simulation using ANSYS Workbench 14.5 were performed. ANSYS is a very powerfully built finite element modelling package for numerically solving a large variety of mechanical problems (Aw *et al.* 2007). ANSYS 14.5 workbench is selected for modal analysis and the load is selected by program automatically. The calculation of the vibrational characteristics of the structure with the FEM, the following equation of motion is performed using modal FE code-ANSYS 14.5.

The dimensions of plate and vibration absorber are 450 x 450 x 1 mm and 280 x 30 x 30 mm, respectively. Figure-1 presents the thin walled structure attached with dual LDVA. Material employed for both systems are plain carbon steel, in which subsequently identify the mass of LDVA is 516 g. In the finite element modelling of plate, the analysis was performed plate with boundary condition of fixed-fixed end subjected to point load, F. The equation of motion of plate can be written in Equation. 1 (Zaman *et al.* 2014b), (Zaman *et al.* 2015).

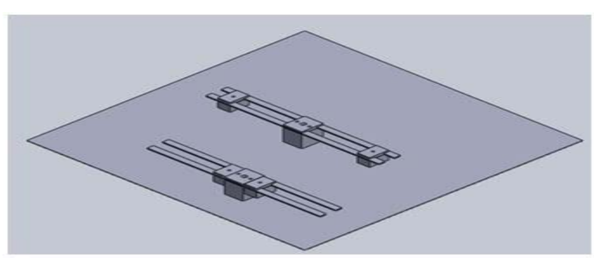

Figure-1. Isometric view of plate with dual DVA.

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$$EI\left(\frac{\partial^4}{\partial x^4} + 2\frac{\partial^2 \omega}{\partial x^2 \partial y^2} + \frac{\partial^4 \omega}{\partial y^4} + \rho h\frac{\partial^2 \omega}{\partial t^2} = -F(x,y,t)\right. \quad (1)$$

$$I = \frac{h^2}{12(1-v^2)} \quad (2)$$

Where ρ is density, and I is the area moment of inertia which defined in Equation. 2.

Where [M] is a mass matrix, [K] is a real valued stiffness matrix, and {u} is the displacement vector. Natural frequency and modes shape can be determined by performing a numerical modal analysis with ANSYS. For the case of simulating harmonic analysis, the damping matrix [C] is used. In its most general form, it is showed in Equation. 3. The detail of this part can be found in reference.

$$[M]\{\ddot{x}(t)\} + [C]\{x(t)\} + [K]\{x(t)\} = \{F(t)\} \quad (3)$$

Where-[M], [C], [K] are the global mass, Damping and Stiffness Matrix of the model {x (t)}-Displacement vector. For undamped free vibration analysis the damping and external excitation force is zero ([c] =0, [F] =0). So the Equation. 3 can be represented as undamped free vibration.

$$[M]\{\ddot{x}(t)\} + [K]\{x(t)\} = 0 \quad (4)$$

Equation above can be written as:

$$\{x\} = (X)e^{-i\omega t} \quad (5)$$

where {X} represents the amplitudes of vibration of all the masses (mode shape or eigenvector's), ω natural frequency, so the Equation. 4 reduces in-

$$([K] - \omega^2 [M])\{X\} = 0 \quad (6)$$

If we replace ω^2 by λ the Equation. 6 become a linear problem in matrix algebra. $\{X^i\}$ has nonzero solution, then coefficient matrix must be equal zero. Each eigenvector{X} and corresponding eigenvalues $\{\omega i^2\}$ is solved using ANSYS.

Computer Simulation Result and Discussion

In this section FEM performed in plate structure to attenuate harmonic response using the single and dual LDVA. The complete model was meshed to 3016 numbers of elements and 7804 numbers of nodes. Figure-1 shows the models of LDVA coupled to a plate structure with single excitation response. While, the dominant frequencies of plate was important to predict the dynamic characteristic of plate structure. Table-1 tabulates the results of first five natural frequencies of a free vibration response of plate structure.

Table-1. Natural frequencies of fixed-fixed plate.

Mode (nth)	Natural frequency (Hz)
1	43
2	162
3	281
4	387
5	519

The concept of the DVA is to match the frequency of vibration neutralizer with the plate's fundamental frequency (Zaman *et al.* 2014c). The resonance frequency of LDVA decreases when both masses moving away (along the axis) the centre mass of neutralizer. Figure-2 shows the comparison of simulated displacement response of the plate attached with single LDVA. The single LDVA resonance frequency is tuned to the first fundamental frequency of the plate at 43 Hz in order to suppress the vibration. The single vibration neutralizer was attached at the centre of the plate in order to address the first frequency mode of the plate. From the result obtain, can observe that after placing the LDVA on the plate, the vibration at the targeted resonance 43 Hz has been greatly attenuated. LDVA suppresses the resonance to which it is tuned. Surprisingly, by attaching single LDVA, the displacement response of a thin plate structure was decreased at all frequency bands.

Figure-3 illustrated the comparison frequency response of plate without attached with dual DVA. The current results found that when dual LDVA installed on the plate, it suppresses the resonance for the first second and fourth mode of the entire frequency band. The 3rd and 5th frequencies band were identified that the displacement of specified modes was drawback after attaching with dual LDVA towards plate structure. Prior studies that have noted the two main factor influence of drawback when the obvious weight of the DVA and the position of DVA (Brennan & Dayou 2000; Ranjan & Ghosh 2005). The result of average percentage reduction of vibration for plate structure without LDVA, with single LDVA and dual LDVA was demonstrated in Table-2.

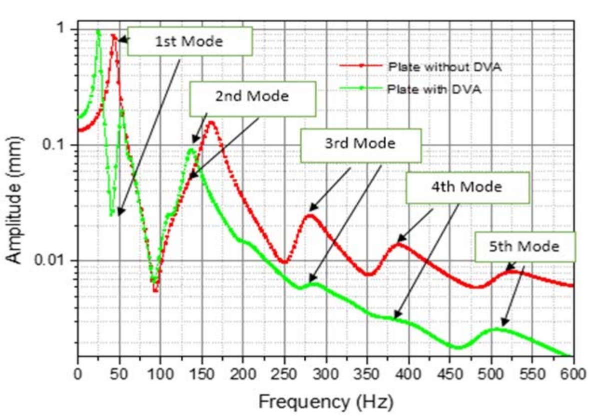

Figure-2. Frequency response plate with single DVA.

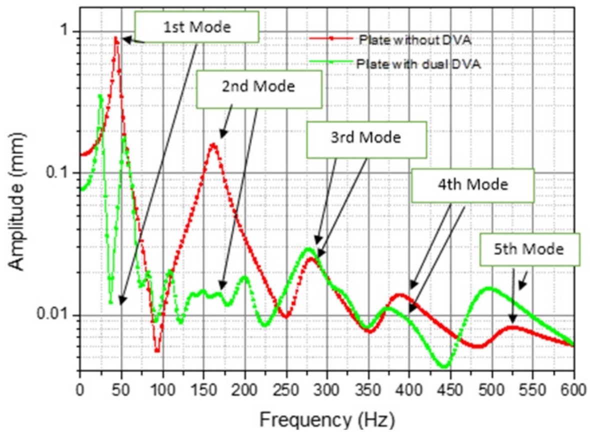

Figure-3. Comparison between frequency response plate with dual absorber.

Table-2. Vibration amplitude of plate before and after attached with single and dual DVA.

Mode nth	Without DVA	Single DVA		Dual DVA	
	Amp (g)	Amp (g)	Reduction (%)	Amp (g)	Reduction (%)
1	0.8930	0.0251	97.18	0.0406	95.45
2	0.1588	0.0375	76.38	0.0131	91.75
3	0.0240	0.0061	1.79	0.0281	-17.08
4	0.0138	0.0031	77.53	0.0105	23.91
5	0.0066	0.0025	62.12	0.0153	-131.8
		Average	63.28	Average	31.192

CONCLUSIONS

In this paper, we have described the finite element simulation analysis of a mechanism for reducing the vibration caused by vibrating thin walled structure. A novel passive vibration control configuration, namely the lightweight dynamic vibration absorber, has been proposed to suppress the oscillatory motion of excited linear plate structure. The result revealed that the plate attached with single LDVA provide a larger suppression of resonant vibration amplitude at the targeted mode about 97.8% and reduced vibration amplitude along frequency band of the primary system excited by excite force motion. However, when the plate attached with dual absorber, the vibration amplitude at targeted frequency was found reduce to 93.4% and while the vibration amplitude at 3rd and 5th mode was highly compared with plate without LDVA. However, the additional weight of vibration absorbers need to be optimized in order to compromise the weight limitations.

ACKNOWLEDGEMENTS

The authors thank Office for Research, Innovation, Commercialization, and Consultancy Management (ORICC) and Universiti Tun Hussein Onn Malaysia for the financial support under Postgraduate Incentive Research Grant (GIPS) vote U040 and Incentive Grant Scheme for Publication (IGSP) vote U247.

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