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MODIFICATION OF VIBRATING THIN WALLED STRUCTURE USING DYNAMIC VIBRATION ABSORBER

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ABSTRACT

Vibration control have widely used to suppress the vibration of thin walled structure. There are two different types of thin walled properties such as a thin plate with flexural rigidity and without flexural rigidity. To improve modification of thin walled structure, dynamic vibration absorber (DVA) was introduced to reduce vibration on thin structure. This paper demonstrated experimental force vibration by using exciter motor shaker to investigate the dynamical behaviour of a thin walled structure attached with DVA. Experimental results, in terms of modal parameters and vibration reduction performance, are compared with numerical approaches which obtained through a finite element model. The measurement result found that when DVA attached to plate structure the vibration amplitude of the controlled mode are significantly reduced by almost 85% and 94% for single and dual DVA attachment, respectively. It concludes that adding more DVA attached on the structure produce better result in vibration reduction.

Keywords: Force Vibration・Finite Element Method・Vibration Absorber

INTRODUCTION

Over the past two decades, there are numerous research works concerning on excessive vibration through the thin walled structures such as plate (Howard, 2007), (Zaman et al., 2013), (Zaman et al., 2014). Researcher investigated dynamic characteristic on plate structure because the are various type of thin walled structural components are commonly found in spacecraft, missiles, aircraft, land based vehicles, under-water vessel, several machine structures, civil engineering structure, boards in electronic equipment and structure including chemical processing equipments, computer peripherals and modern housing (Ranjan and Ghosh, 2005) (Cook and Thompson, 1990) (Yang et al., 2011). Many studies of the vibration response of plate structures either employed analytical (Lin 2012; Cook 2001) or numerical approaches (Dozio & Ricciardi 2009; Holopainen 1995)

Experimental modal analysis (EMA), one of the common method to identify dynamic characteristic of structure. Modal vibration testing has the potential to provide the basis for rapid, inexpensive characterization of both elastic and viscoelastic properties of composites for design and manufacturing (Gibson 2000), (Zaman et al., 2009). Frequency response functions are important measurement functions in experimental vibration analysis. Experimental modal analysis divide two common technique, impact excitation test and shaker test (He and Fu 2001).

Vibration control has been used to maintaining a high performance level of thin walled structure. In recent years, researches have devoted to studying passive vibration absorber to attenuate vibration of the structure. Tarng et al. (2000) applied tuned vibration absorber to suppress of chatter in turning operation. Nakano et al. (2013) investigated the effect of the multiple dynamic absorber to reduce chatter on milling machine. Anderson (2014) developed dynamic vibration absorber to reduce beaming vibration of highway tractor. Yang et al. (2011) proposed analytical solution when number of the vibration absorber attached to the structure. Rubio et al. (2013) studying of a passive vibration absorber attached to a boring bar and analyzed using Euler-Bernouli method. Hao et al. (2011) proposed simple approach to supress hand arm vibration in electric grass trimmer using tuned vibration absorber. Modal analysis experiment was executed to achive this studying.

This paper focuses on investigating the forced vibrations of a thin walled structure with attached dynamic vibration absorber. The plate used is fixed-fixed ends and subjected to the exciter force. The restoring force of the dynamic vibration absorber is represented by a mass and damping element. In order to evaluate the vibrational characteristics, the natural frequencies and the vibration amplitude of the plate are identified experimentally through modal tests by adopting the exciter force technique.

Formulation of Dynamic vibration absorber

The concept of the DVA is to match the frequency of vibration neutralizer with the plate's fundamental frequency. The resonance frequency of DVA decreases when both masses moving away (along the axis) the centre mass of neutralizer (Bonello, 1997), (Zaman et al., 2014b). The equation used in DVA attached to the structure is:

$$r_{AA}(\omega) = \frac{Y_A}{F} = \frac{1-\alpha^2}{m_h\,\omega^2} \quad (1)$$

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$$\left[\frac{\left[\frac{1}{\tilde{\omega}^2}+\beta_{AA}(\tilde{\omega})\right]\left[1-\sigma-\sigma\tilde{\omega}^2\beta_{BB}(\tilde{\omega})\right]+\left[-\frac{1}{\tilde{\omega}^2}+\beta_{BA}(\tilde{\omega})\right]\left[-\sigma+\sigma\tilde{\omega}^2\beta_{AB}(\tilde{\omega})\right]}{[1-\tilde{\omega}^2]\left[1+\sigma-\sigma\tilde{\omega}^2\sum_{k=1,3,5,\ldots}^{K}\frac{\{\phi_B^{(k)}\}^2}{\tilde{\omega}_k^2-\tilde{\omega}^2}\right]-\sigma\tilde{\omega}^2\{\phi_B^{(k)}\}^2}\right] \quad (2)$$

Where:

$$\sigma=\frac{m_{att}}{m_b},\quad \tilde{\omega}=\frac{\omega}{\omega_1},\quad \tilde{\omega}_k=\frac{\omega_k}{\omega_1},\quad \beta_{QS}(\tilde{\omega})=\sum_{k=1,3,5,\ldots}^{K}\frac{\phi_Q^{(k)}\phi_S^{(k)}}{\tilde{\omega}_k^2\tilde{\omega}^2} \quad (3)$$

In the above equation m_{att} is the sum of the attached masses at B (for moveable masses) and m_b is the mass of the cantilever beam alone. $\phi_Q^{(k)}$, $\phi_S^{(k)}$ are the k^{th} flexural non-dimensional mode-shape functions evaluated at arbitrary points Q, S and ω_k is the k^{th} flexural natural frequency. The expression in eq. 1 is exact for K→ ∞ but, for a given upper limit of the non-dimensional excitation frequency $\tilde{\omega}$, will be virtually exact for a sufficiently large finite value of K (corresponding to a total of (K+1)/2 symmetric modes). For the moveable masses absorber, $\phi_B^{(k)}$ is fixed and $\phi_A^{(k)}$ is variable. Figure-1 shows moveable mass for dynamic vibration absorber.

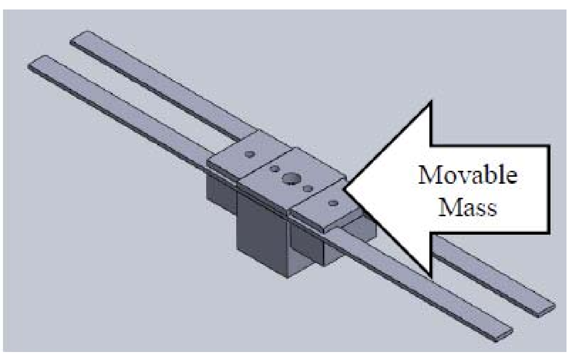

Figure-1. Dynamic vibration absorber

Experimental Setup

Experimental force vibration was conducted to evaluate dynamic response of the thin walled plate structure. Result was obtained result using DEWE with 16-channel analyzer. The thin walled structure was in condition fixed-fixed end of the testing rig. The motor force was used to provide an impulsive force on the structure to ensure the thin walled structure excite in the frequency range between 0-600 Hz.

Single-input multi-output (SIMO) is a common method to analyze force vibration response of structure (Jin 2011). Before begin the investigation accelerometer was calibrated using microphone calibrator. Calibration is important to ensure measurements of the experimental force vibration is accurate. Five accelerometers were used to measure the absolute accelerations on the plate structure for all 25 points. The excite shaker is placed at the left side of plate structure are shown in Figure-2. Multi response is captured and the data collected is imported to the DEWE software.

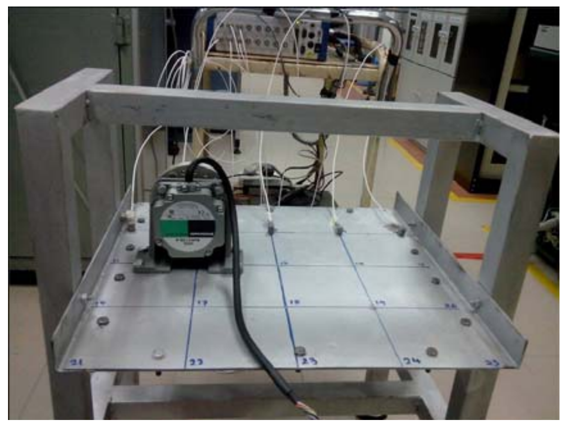

Figure-2. Excite force attached to plate structure

RESULTS AND DISCUSSION

The current study describes the frequency response of a fixed-fixed end of plate without DVA in the frequency range 0–600 Hz are illustrated in Table-1. From the graph frequency response indicate the dominant 5 peaks that represent each dynamic characteristic of plate structure as shown in Figure-3. The highest amplitude of displacement is at the first peak which is 1.76147 g. This is because the natural frequency for exciter approximately same with the natural frequency of the first peak.

Table-1. Vibration on thin walled structure without DVA

Mode	Natural Frequency (Hz)	Amplitude (g)
1	60	1.761476
2	160	1.27148
3	286	1.098021
4	359	0.75670
5	500	0.801391

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Investigation was done for plate structure attached with DVA to improve the dynamic behaviour of the plate structure. Experimentation of single and dual DVA attached to plate structure were continued to compared performance of vibration reduction. The first DVA was tuned to 60 Hz in order to suppress the vibration at the first frequency band. Vibration neutralizer was attached at the centre of the plate. The second DVA was tuned to the second harmonic response at 160 Hz to tackle second harmonic response.

Result below shows the three highest vibration response at point 8, 13 and 19 to compare the resonance amplitude of plate after attached single and dual DVA. Figures 3 to 5 comparee the effect of vibration amplitude when adding neutralizers on thin plate. Obviously by attaching single and dual vibration neutralizer, the vibration of a thin plate structure was decreased, especially in the first and second mode predominant frequency. This was in agreement with our previous studies (Zaman et al., 2014c), (Zaman et al., 2015), (Salleh and Zaman, 2015). In this case, the neutralizers was executed in order to counter the first bending motion of the thin plate which excited by a motor shaker.

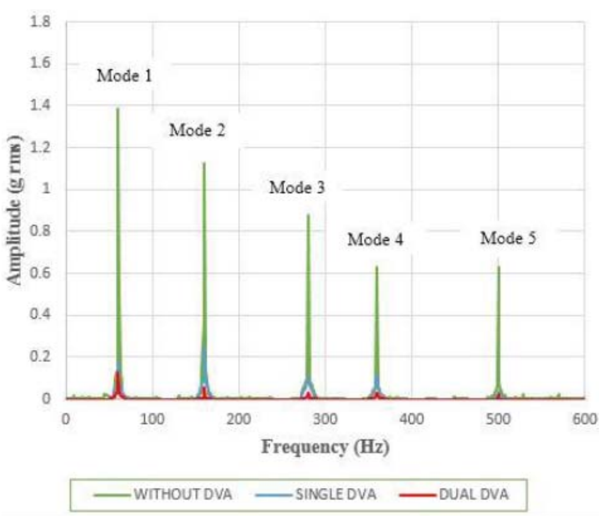

Figure-3. Vibration amplitude versus frequency at point 8

The most interesting finding by attaching single and dual DVA, the vibrations of a thin walled structure in Figures 3 to 5 were found decreasing at all points of frequency modes. The average percentage reduction of single and dual DVA attaching to plate structure are shown in Table 1 about 85.18 and 93.73 percent.

The time response of the thin walled structure also investigated to ensure results for EMA more robustness. Dynamic response such displacement as a function of time as due to excitation external forces. The displacement of the thin walled structure single, dual and without two DVAs is illustrated in Figures 6 to 8 show good reductions of the structural responses has been achieved as the structure is implemented with dual DVAs.

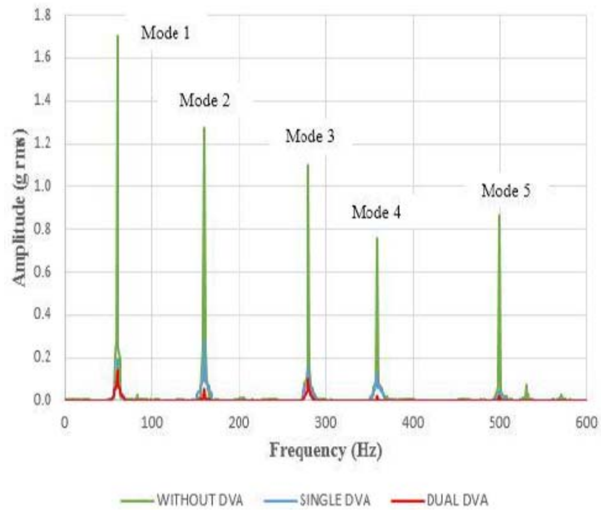

Figure-4. Vibration response versus frequency at point 13

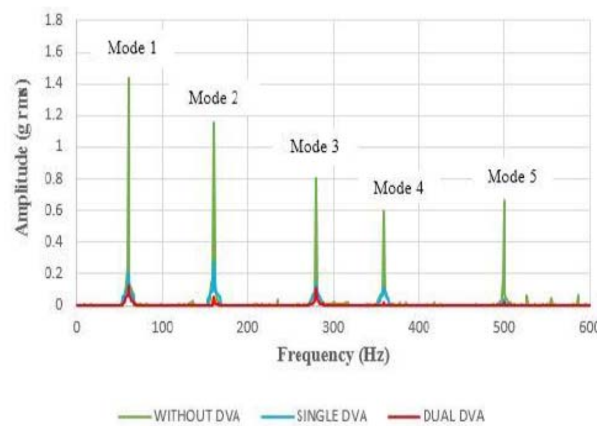

Figure-5. Vibration response versus frequency at point 19

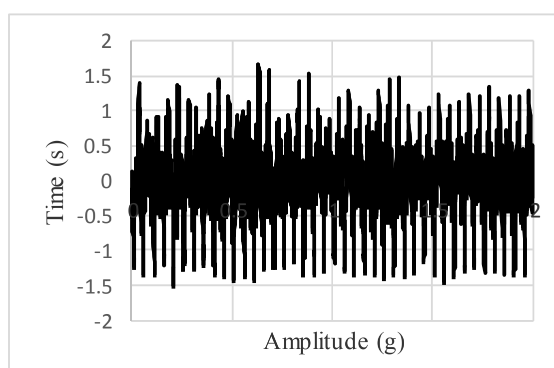

Figure-6. Time response of plate without attached DVA

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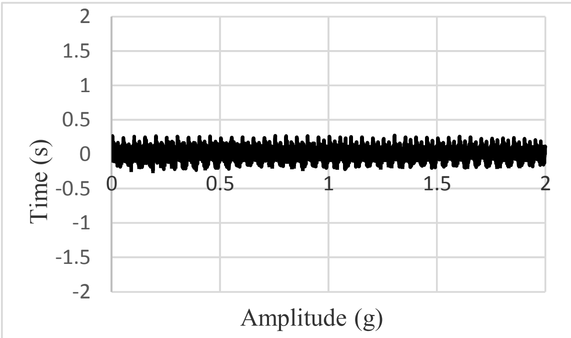

Figure-7. Time response of plate with attached single DVA

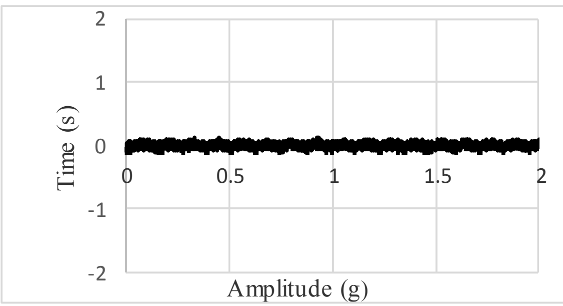

Figure-8. Time response of plate with attached dual DVA

The performance of the DVA on the suppression of the super-harmonic resonance response is also evident in the vibrational signals of the nonlinear primary oscillator with and without absorber, as shown in Figures 6 to 8. The amplitude of vibration of the nonlinear primary oscillator with DVA is smaller than that of the nonlinear primary oscillator without the absorber. It is also noted that the super-harmonic resonance is eliminated in the response of the nonlinear primary oscillator with absorber. The elimination of the super-harmonic resonance corresponds to the disappearance of the free-oscillation term.

Comparison between experimental and simulation

This section presents data for EMA and FEM to validated in term of modal parameter and vibration reduction of plate structure attached with DVA. This study was done by using ANSYS software and exciter force experiment. The five reliable modes of vibration in the frequency range up to 600 Hz were identified in Table-2.

Results of free vibration response are presented in Table-2 between numerical analysis and exciter force. From the result found that a very good correlation value is observed between for 2nd, 3rd, 4th and 5th mode frequency band. Meanwhile, remarks that for both cases the difference percentage error increases for first modes with the difference more than 28.33%.

Table-2. Compares natural frequency for different method

Mode	Exciter Force(Hz)	FEM(Hz)	Percentage error (%)
1	60	43	28.33
2	160	162	1.25
3	280	281	0.035
4	359	387	7.79
5	500	519	3.80

In Figure-7(a) and (b) compares the percentage vibration reduction of plate structure attached with single DVA and dual DVA used two different approaches . The results indicate that the percentage reduction of displacement by both approaches after modification with attached single absorber more than 88%. Suprisingly, when dual absorbers were attached onto plate shows the average global vibration reduction at targeted mode is more than 91% for both method.

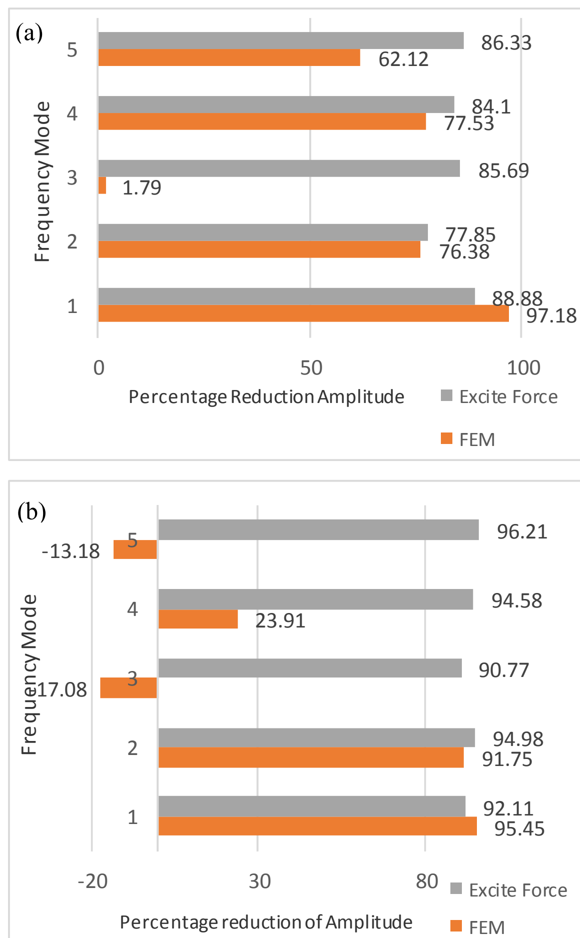

Figure-7. a) Percentage response of single DVA b) Percentage error of dual DVA.

CONCLUSIONS

In this paper forced vibrations of fixed-fixed end plate attached with DVA was analyzed using an experimetal and numerical study to investigate the natural frequencies and the displacement amplitude. The passive vibration utilization to modifying the vibration response of the plate structure transmitted to the DVA. From the results, it was shown that the DVA acts as a vibration attenuator of the plate structure. From the result, the following observations were made:

- Using single DVA the vibration response using both methods reduce on targeted mode and attenuate vibration response along frequency bands.
- For dual absorber attached to plate structure using experimental modal analysis and finite element method the vibration amplitude reduce significantly on targeted mode. Meanwhile the vibration response reduce along the frequency band when analyze using EMA.

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