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The Application of Multiple Vibration Neutralizers for Vibration Control in Aircraft

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Abstract. A current challenge for researchers is the design and implementation of an effective vibration control method that reduces vibration transmission from vehicle structures such as aircraft. This challenge has arisen due to the modern trend of utilizing lightweight thin panels in aircraft structural design, which have the potential to contribute towards significant vibration in the structures. In order to reduce structural vibration, one of the common approaches is considering vibration neutralizer system attached to the structure. In this study, a vibration neutralizer is developed in a small scale size. The effectiveness of attached vibration neutralizers on a thin plate are investigated through experimental study. Prior to the experiment, a finite element analysis of Solidworks® and analytical modelling of Matlab® are produced in order to determine the structural dynamic response of the thin plate such as the natural frequency and mode shapes. The preliminary results of finite element analysis demonstrate that the first four natural frequency of clamped plate are 48Hz, 121Hz, 194Hz and 242Hz, and these results are in agreement with the plate's analytical equations. However, there are slight discrepancies in the experiment result due to noise and error occurred during the set up. In the later stage, the experimental works of thin plate are performed with attached vibration neutralizer. Result shows that the attachment of vibration neutralizer produces better outcome, which is about 41% vibration reduction. It is expected that by adding more vibration neutralizer to the structure, the vibration attenuation of thin plate can be significant.

Introduction

The development of aerospace industries demands the use of lightweight materials such as thin panels. However, thin panels have potential to contribute to substantial vibration, which leads to excessive noise [1, 2]. Previous researches have shown that adding resonant devices such as vibration neutralizer to vibrating structures can reduce vibration levels. Unfortunately, it produces drawbacks such as the obvious weight increase, and the most critical is that improper placement and tuning of passive vibration neutralizer may result in large increase of vibration level.

The idea of vibration neutralizer emerges due to a common vibration problem existed in engineering structural applications. Vibrations have became a major concern in most engineering fields. It has turned out to be a hazard that reduces the fatigue life of the structure. Excessive vibration can lead to structural failure and induce uncomfortable noise which ultimately can cause catastrophic and hearing damage [3–5]. With the increase number of dynamic systems such as in mechanical tools, machinery, automobiles and aircraft, environmental vibration and noise are expected to increase.

Over the past 20 years, there are numerous research works concerning on excessive vibration through the thin-walled structures such as plate [6–9]. This is a problem commonly found in automobiles, aircraft and the fairings of rocket launch vehicles. Many studies have been devoted in the past to develop a method to reduce vibration that generated by machines [10–13]. These include: (i) modifying the system such that the natural frequency does not coincide with operating speed, (ii) apply damping to prevent large response, (iii) installing isolating devices between adjacent sub-systems and (iv) adding discrete masses into equipment to reduce the response and absorb vibration.

In spite of these studies and knowledge gained thus far, it has not yet reached a situation where a person able to find a vibration control method to fit all vibration situations. In fact, the first three aforementioned methods are hard to be implemented and not really effective because of its design complexity, costly and unfeasible at lower frequency [12]. The latter approach using discrete masses or known as vibration neutralizers, however, is more sound because it does not contribute significant additional vibration energy to the structure, plus proven to yield substantial attenuation in structural vibration [6, 14]. Fig. 1 illustrates the application of vibration neutralizer which comprises of mass, spring and damper attached to a machine.

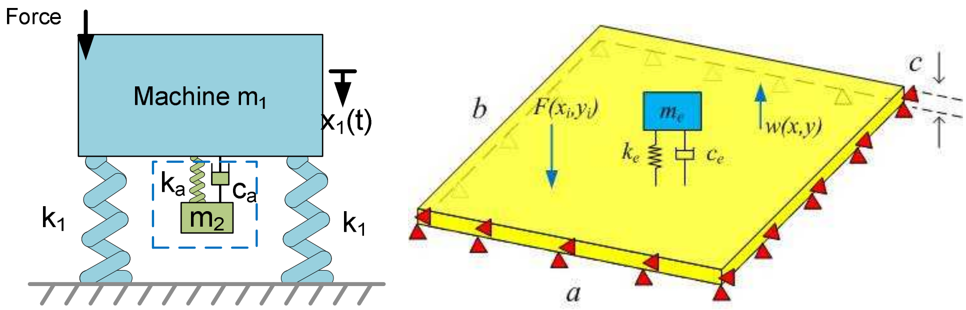

Figure 1: Illustration of vibration neutralizer attached to machine structure and panel

This paper aims to investigate the effectiveness of using multiple passive vibration neutralizer to reduce surplus vibration in a thin panel structure of an aircraft. An in-depth study on the tuning frequency of neutralizer will be conducted, either to address the forcing frequency or the natural frequency of the thin plate; in order to ensure that the whole panel vibration can be reduced significantly. The outcomes are expected to revolutionize the current performance of neutralizers over a broad frequency range, therefore improve their versatility and effectiveness in vibration control of aircraft.

Mathematical Modeling

Vibration response of thin panel. Consider a three dimensional thin plate with dimensions *a*, *b* and *c* which represent length, width and thickness subjected to a vertical point load *F*. The equation of motion of a plate can be written as in Eq. 1 [15]:

$$EI\left(\frac{\partial^4 w}{\partial x^4}+2\frac{\partial^2 w}{\partial x^2 \partial y^2}+\frac{\partial^4 w}{\partial y^4}\right)+\rho h\frac{\partial^2 w}{\partial t^2}=-F(x_i,y_i,t) \tag{1}$$

where E is the Young's modulus, I is the area moment of inertia, ρ is the density of plate and h is thickness of plate. The solution of transverse modal displacement for a plate is given in Eq. 2 which is summation of all of the individual modal amplitude responses multiplied by their mode shapes at that point [15].

$$w(x,y,t)=\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}W_{mn}\cdot\psi_{mn}(x,y)\,e^{j\omega_n t} \tag{2}$$

where W_{mn} is the modal amplitude, $\psi_{mn}(x,y)$ is the mode shape of plate, and m and n are modal integers.

By neglecting the exponential time varying term, an expression of the total response of clamped plate incorporating the viscous damping, ζ is simplified as in Eq. 3.

$$w(x,y,t)=\frac{F}{\rho abc}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\psi_{mn}(x,y)\cdot\psi_{mn}(x_i,y_i)}{\omega_{mn}^2-\omega^2+j2\zeta\omega\omega_{mn}} \tag{3}$$

Vibration neutralizer installed on thin panel. Consider a model of a clamped plate attached with a vibration neutralizer subjected by a harmonic load F as illustrated in previous Fig. 1. In order to calculate the response of thin plate after attached with vibration neutralizer, the receptance at a point of attachment of neutralizer is calculated from Eq. 4 [15].

$$\beta_{22}=\frac{m_e s^2+c_e s+k_e}{m_e s^2(c_e s+k_e)} \tag{4}$$

By manipulating Eq. 3 and Eq. 4, the response of a plate at point (x_1, y_1) with a spring-mass-damper system attached at a point (x_2, y_2) can be calculated using receptance method. The displacement equation for a thin plate with attached a vibration neutralizer is given as follows:

$$w(x_1,y_1)=\left[\alpha_{11}-\frac{\alpha_{22}^2}{\alpha_{22}+\beta_{22}}\right]\cdot F(x_i,y_i) \tag{5}$$

where,

$$\alpha_{11}=\frac{1}{\rho abc}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\psi_{mn}^2(x_1,y_1)}{\omega_{mn}^2-\omega^2+j2\zeta\omega\omega_{mn}} \tag{6}$$

$$\alpha_{21}=\frac{1}{\rho abc}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\psi_{mn}(x_1,y_1)\cdot\psi_{mn}(x_2,y_2)}{\omega_{mn}^2-\omega^2+j2\zeta\omega\omega_{mn}} \tag{7}$$

$$\alpha_{22}=\frac{1}{\rho abc}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\psi_{mn}^2(x_2,y_2)}{\omega_{mn}^2-\omega^2+j2\zeta\omega\omega_{mn}} \tag{8}$$

Finite Element Analysis

Two finite element (FE) models of a thin plate and vibration neutralizer are simulated using Solidworks® in order to determine their natural frequencies and mode shapes. Fig. 2 shows the FE models of the thin plate and vibration neutralizer, where the studied dimension are 450 x 450 x 2 mm and 280 x 30 x 30 mm, respectively. Both models use mild steel and are meshed using solid wedge 6-nodes element. The frequency modal analysis was solved over the frequency range from 0 to 300 Hz.

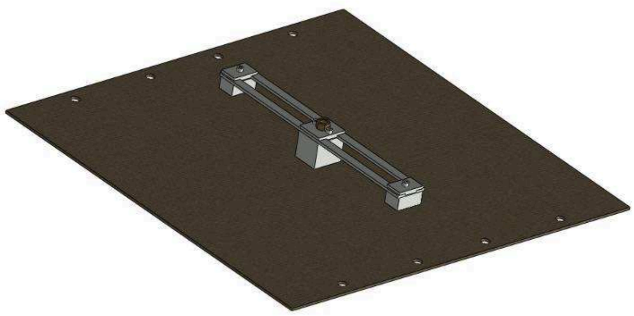

Figure 2: FE models of a vibration neutralizer attached to plate

Results and Discussion

Vibration neutralizer. The concept of vibration neutralizer is to match the frequency of vibration neutralizer with the plate's fundamental frequency. In order to achieve this, two small masses are designed as shown as in Fig. 2 so that each mass can be moved along the flyer. Fig. 3(a) shows the resonance frequency of neutralizer decreases when both masses moving away (along the axis) from the centre of neutralizer. Based on this result, the frequency of neutralizer can be tuned so that it coincides with the plate's fundamental frequency. Fig. 3(b) shows the first vibration mode of the neutralizer.

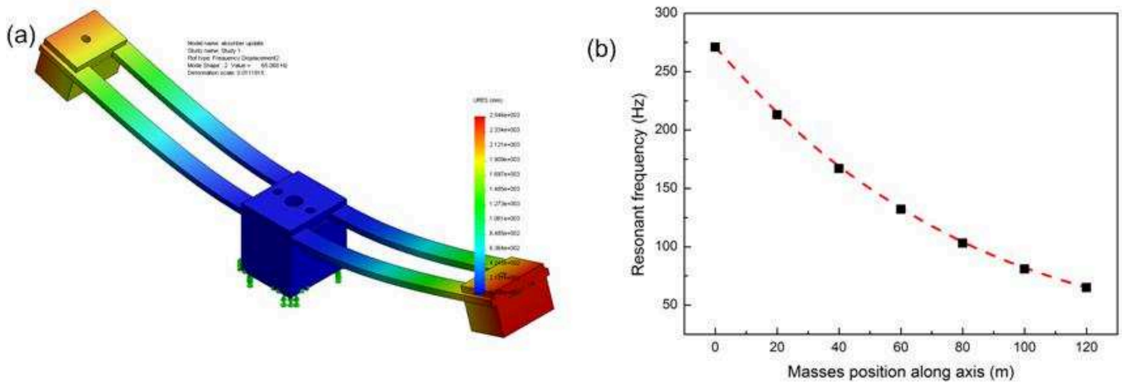

Figure 3: FE analysis of vibration neutralizer

Clamped thin panel. A dynamic characteristic of thin plate was determined by three approaches; finite element analysis by Solidworks®, mathematical modelling by Matlab® and experimental test. Table 1 tabulates the results of first four natural frequencies of a clamped thin plate obtained from these approaches. It shows that all these approaches corroborated well, although there is a small discrepancy in the experimental result due to error and noise measured during testing. Meanwhile, the mode shapes of the thin plate are displayed in Fig. 4.

Table 1: Natural frequencies of thin plate in unit Hz

Mode	Finite Element Analysis	Mathematical Model	Experiment
1	48	48	41
2	121	121	122
3	194	193	163
4	242	242	-

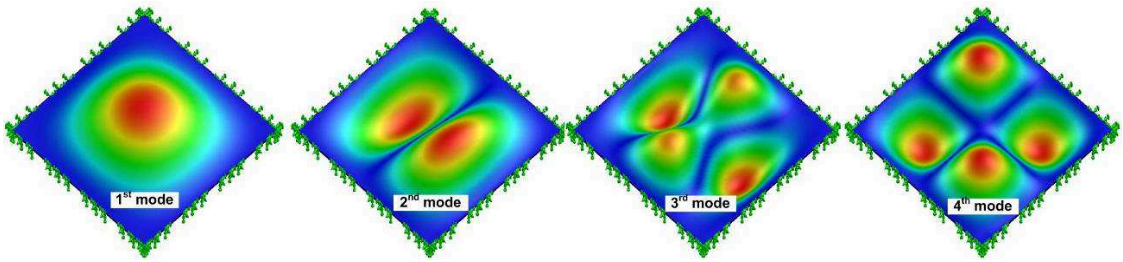

Figure 4: Mode shapes of a thin plate

Thin panel with attached vibration neutralizers. The next stage of analysis incorporates the attachment of vibration neutralizers on the thin plate via testing. The neutralizer's resonance frequency is tuned to the first fundamental frequency of the plate at 41 Hz in order to suppress the vibration. Figs. 5–8 show the superimposed result of the frequency response function of a plate measured at respective points 2, 3, 6 and 7 (refer Fig. 2 in Ref. [11]) by comparing the response without vibration neutralizer, with attached single vibration neutralizer and with attached dual vibration neutralizers. For single vibration neutralizer, the neutralizer was attached at the centre of plate (point 6) in order to address the first frequency mode of plate as shown in Fig. 4. While the neutralizers were fixed at points 6 and 10 (refer Fig. 2 in Ref. [11] for clarification) for dual vibration neutralizers attachment.

From Figs. 5–8, it can be visually compared the effect of adding neutralizers on the vibration amplitude of a thin plate. Obviously by attaching single and dual vibration neutralizer, the vibration of a thin plate structure was decreased at all points observed, especially in the first and second mode predominant frequency. It is noticed that the vibration amplitude reduced significantly through the thin plate structure when dual vibration neutralizers were employed. The average global vibration reduction obtained at points 2, 3, 6 and 7 with dual neutralizers attachment is about 41%. In this case, the neutralizers attachment was fixed at points 6 and 10 in order to counter the first bending motion of the thin plate which excited by a motor shaker.

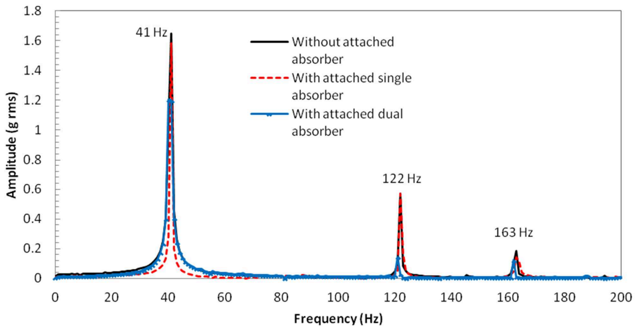

Figure 5: Frequency response at point 2

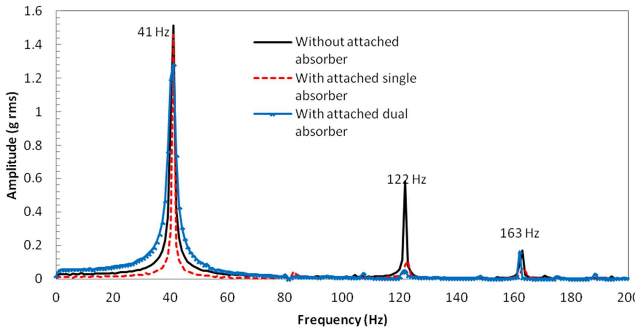

Figure 6: Frequency response at point 3

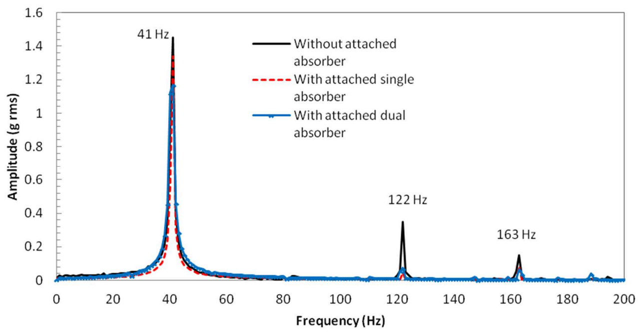

Figure 7: Frequency response at point 6

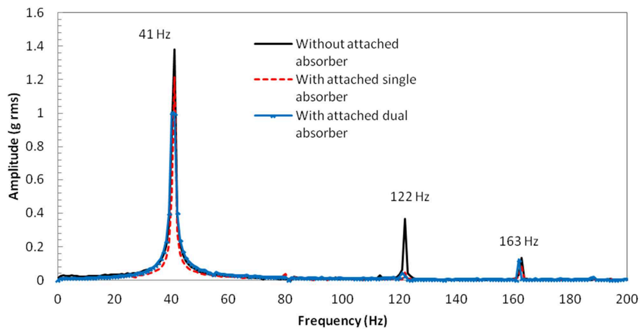

Figure 8: Frequency response at point 7

Conclusion

The attachment of vibration neutralizer on a thin panel structure was successfully prepared in this study. A preliminary result of finite element analysis of Solidworks® showed in agreement with mathematical modeling of Matlab® with error less than ±1%, though the experiment provides slightly different result. From subsequent in-depth testing, the structural vibration of the panel was found decrease with single vibration neutralizer and it was further reduced with dual vibration neutralizer attachment. However, the additional weight of vibration absorbers need to be optimized in order to compromise the weight limitations set by the aircraft. Overall, this study reveals that multiple passive vibration neutralizers attached on a structure proved to be more effective in vibration suppression.

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