SUPPRESSING STRUCTURAL VIBRATION USING MULTIPLE ARRAYS OF LINEAR DYNAMIC VIBRATION ABSORBER (DVA)

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2018

DECLARATION

I declare that this report entitled "Suppressing structrural vibration using multiple arrays of linear dynamic vibration absorbers (DVA)" is the result of my own research except as cited in the references. The report has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

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APPROVAL

I hereby declare that I have read this thesis and in my opinion this report is sufficient in terms of scope and quality for the award of degree of Bachelor of Mechanical Engineering.

Signature : ……………………………….

Supervisor Name : ………………………………..

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ABSTRACT

A fixed-free beam is a structural element supported at one side only and have no any constraint at other side. Exposure to vibration can cause excessive deflections and structure might failed. The aim of this research is to study the application of multi dynamic vibration absorber on a fixed- free beam strucure. For experimentation, two absorbers were fabricated to be installed to the beam and subjected to a force vibration frequency loading using shaker. The resonance frequencies of interest were 14Hz and 99.6Hz. The vibration level of beam is measured for comparing the effect with the presence of application vibration absorbers to see the reduction in its amplitudes. The results shows the reduction of amplitude with single vibration absorber. With application of multiple DVAs, the frequency bandwidth of suppression is increased. Configuration of vibration absorbers on location did not affect its performance in vibration suppression at second natural frequency. From these results, it showd that distributed vibration absorbers is the prefered method of configuration where the range of frequency bandwidth is greatest. The knowledge gained from this research can be used to minimize the vibration amplitdue of sturcutures and increase their life-span. This application can be apply in suppresing the vibration of machine, bridge, and fluttering of airplane wing.

ABSTRAK

Rasuk bebas tetap adalah elemen struktur yang disokong di satu pihak sahaja dan tidak mempunyai sebarang kekangan di sisi satu lagi. Pendedahan kepada getaran boleh menyebabkan pesongan berlebihan dan struktur mengalami kemungkinan gagal. Tujuan penyelidikan ini adalah untuk mengkaji penggunaan penyerap getaran pelbagai dinamik pada strucure rasuk tetap bebas. Untuk eksperimen, dua penyerap telah direka untuk dipasang ke rasuk dan tertakluk kepada frekuensi getaran daya getaran menggunakan penggetar. Frekuensi resonans yang menjadi tumpuan kajian adalah 14Hz dan 99.6Hz. Tahap getaran terhasil pada rasuk diukur untuk membandingkan kesan dengan kehadiran penyerap getaran permohonan untuk melihat pengurangan dalam amplitudnya. Keputusan menunjukkan hasil pengurangan amplitud dengan penyerap getaran tunggal. Dengan menggunakan pelbagai DVA, frekuensi jalur lebar kekerapan penindasan meningkat. Konfigurasi penyerap getaran di lokasi tidak mempengaruhi prestasinya dalam penindihan getaran pada frekuensi resonans kedua. Hasil daripada kajian ini menunjukkan bahawa penyerap getaran teragih adalah kaedah konfigurasi pilihan di mana julat frekuensi jalur lebar adalah lebih besar. Pengetahuan yang diperoleh daripada penyelidikan ini boleh digunakan untuk meminimumkan getaran getaran daripada pelebaran dan meningkatkan jangka hayat mereka. Permohonan ini boleh digunakan pada getaran mesin, jambatan, dan pergerakan sayap pesawat.

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LIST OF ABBEREVATIONS

DVA	Dynamic Vibration Absorber
TVA	Tuned Vibration Absorber
TVD	Tuned Vibration Damper
ATVA	Adaptive Tuned Vibration Absorber
SMA	Shape Memory Alloy
ATMD	Adaptive Tuned Mass Damper
EVA	Electromagnetic Vibration Absorber
PTVA	Passive Tuned Vibration Absorber
VA	Vibration Absorber
ODS	Operating Deflection Shape
HVAC	Heating Ventilation Air Conditioning
ODS	Operating Deflection Shape
FRF	Frequency Response Function
FFT	Fast Fourier Transform

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

Structural vibration is basically described as repetitive motion that can be measured and observed in the structure which occurs due to present of external forces and internal forces. External forces are the force that acts from the outside of the structure such as where these external forces can be due to wind induction, seismic activity, rail system or jackhammer. Internal forces are forces that generated from vibrating structure and transmit to another structure such as force generated from rotating imbalance of a pump inside a plant and transmission to the nearby machine. In most cases, this vibration is undesired as it will results in fatigues in structure, reducing the performance and the life span of the structures where these symptoms is not visible. In production plant, vibration can resulted in increase of energy loss, efficiency reduction, and increase of maintenance cost. At this point, vibration can be a source of engineering problem that needed to be solved.

Resonance is a phenomenon that occurs in vibrating machines or structure, when the external excitation frequency coincides with the natural frequency of the vibrating object. In most cases, resonance is detrimental as it will results in large vibrating magnitude and this will lead devastating effects on machines or collapsing of a structure. One of effects of resonance are incident collapsing of Tacoma Narrows Bridge shown in Figure 1.1, a suspended bridge on 7 November 1940 due to wind-induced vibration. There is also case of bridge collapsing due to induction of mechanical resonance such as collapse of Broughton Suspension Bridge due to marching steps by the troops. The Broughton Suspension Bridge is shown in Figure 1.2.

Figure 1.1: Collapse of Tacoma Bridge (Ellison, 2015).

Figure 1.2: Broughton Suspended Bridge after Rebuilt (Dan, 2016).

As Taiwan prone to natural phenomenon such as earthquake wind storm, The Taipei 101, one of the tallest buildings in the world had installed a 728 Ton pendulum suspended from 92th floor to 87th floor that acts as mass damper to cancel the movement of building induced by strong gust of wind. The Figure 1.4 shows that the location of installment of mass damper inside the buildings. For Millennium Bridge shown in Figure 1.5, precautions were taken by engineers to install 37 fluid-viscous dampers and 52 tuned mass dampers to dampen the vibration caused by pedestrians’ footsteps when crossing the bridge.

Figure 1.3: Tuned Mass Damper in Taipei 101 (Kaushik, 2014).

Figure 1.4 Millennium Bridge (McManus, 2015).

Stockbridge damper, is a dumbbell-shaped device that consists of two masses at the ends of a short length of cable or flexible rod is also used to suppress the oscillation of transmission line caused by induction of wind. The mass damper is usually clamped at its middle to the main cable due to the large amplitude at the middle of cable as it is clamped at both ends. The figure of application of Stockbridge Damper is shown in Figure 1.5.

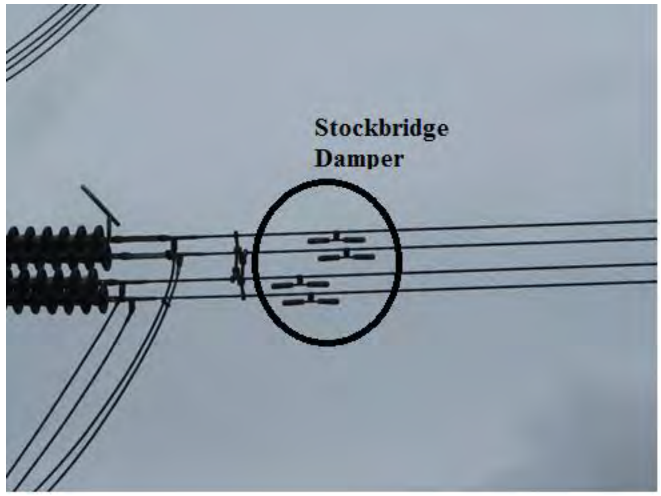


Figure 1.5: Application of Stockbridge Damper on Transmission Cable (Rao, 2015).

1.2 PROBLEM STATEMENT

Dynamic Vibration Absorber (DVA) is a designed mechanical devices that consists of spring-mass system that is able reduce unwanted vibration in machines or structure by exerting a counter force to couple the vibration force. However, DVA can only tuned to a single frequency and it's only effective for that particular resonance that is tuned. Figure 1.6 shows the graph of amplitude against frequency ratio. From Figure 1.6, there is one single point where amplitude is minimum at frequency ratio almost equal to one. The frequency ratio of one means that the DVA's frequency should be equal to the natural frequency of primary system. This indicates that any frequency miss-tuned of the dynamic vibration absorber's frequency result in larger vibration amplitude which will affect the main vibrating system.

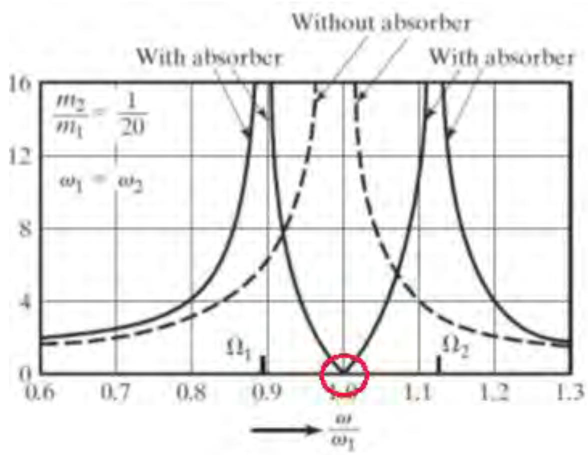


Figure 1.6: Frequency Response (Rao, 2015).

1.3 OBJECTIVE

The objectives of this project are as follows:

1. To characterize the property of a linear DVA.
2. To design and characterize DVA for main structure vibration suppression.
3. To investigate the performance of multiple arrays linear DVA to suppress structural vibration.

1.4 SCOPE OF PROJECT

1. The first and second natural frequency of beam will be considered.
2. Only two DVA will be used to suppress the vibration amplitude of the beam at the first and second natural frequency.

CHAPTER 2

LITERATURE REVIEW

2.1 Vibration

2.1.1 Vibration Terminology

Vibration is a periodic motion of particles of an elastic body or medium in alternately opposite from position of equilibrium where that equilibrium has been disturbed. In other words, vibration is the repetitive motion of after an interval of time, which also referred as oscillation.

2.1.2 Elementary Parts of Vibrating System

In a vibrating system, it consists of mass (inertia) for storing kinetic energy, spring (elasticity) for storing potential energy and damper for dissipating energy. The vibration of system involve the exchange of kinetic energy and potential energy and vice versa, in an alternate manner. Present of damper in vibrating system will dissipate energy for each cycles of vibration and eventually convert the energy of the vibrating system into other forms.

2.1.3 Free Vibration and Forced Vibration

For free vibration, the system is vibrating on its own without the influence of external force. For free vibration, the system is vibrating under external driving force and if the excited frequency coincide with one of the natural frequency of system, resonance will occurs and resulted large vibration amplitude.

2.1.4 Deterministic VS Random Vibration

In deterministic vibration, the magnitude of the force excitation acting on vibrating system is known at any instant. In this case, the excitation is deterministic. Example for deterministic vibration is sinusoidal force produce by shaker where the signal is generated by computer controlled and amplified by amplifier. For random vibration, the value of the excitation at that particular instant not able to be determine. Examples of random vibration are wind induction, seismic activity, or a car moving on a rough surface road.

2.1.5 Discrete VS Continuous

Number of degree of freedoms refer to number of independent coordinates that required to determine completely the position of system at any of time. For discrete or lumped parameter system, the number of degree of freedom are finite. In contrary, continuous or distributed system have infinite number of degrees of freedom and mostly related to deformable structural and machine system with elastic properties.

2.2 Beam Structure as Continuous System

Euler-Bernoulli Theorem, a linear theory of elasticity which relate between beam's deflections and applied load is widely use to study the propagation of flexural waves by considering the boundary condition of the beam. The boundary conditions of beam include pinned-pinned, free-free, fixed-fixed, fixed-free, fixed-pinned and pinned free. The Figure 2.1 shows the boundary conditions of the beam.

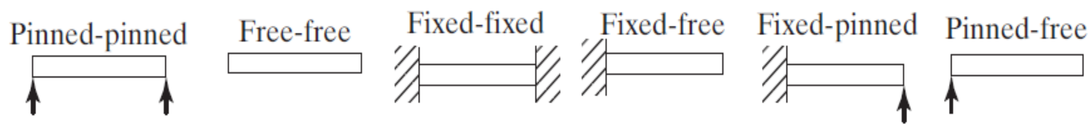
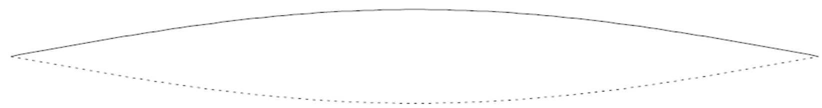


Figure 2.1: Boundary conditions of beam (Rao, 2011).

2.3 Mode Shape

Compared to single mass attached to a single spring, the vibration of complex body vibrate in more different ways. By changing the excitation frequency, the body will exhibit different vibrating pattern, where these patterns are termed as mode shape. Consider a string that is simply supported at both side. By exciting the lowest frequency, also known as the fundamental frequency at the string, the string will oscillates repeatedly as one– with the greatest motion in the center of the string. The Figure 2.2 shows the mode shapes of the string at the different frequency. When the frequency is increase twice the fundamental frequency, two halves of the spring vibrate in opposite manner as shown in Figure 2.3. The solid line is the maximum displacement at one instant and dotted line is the displacement at later instant. If the frequency is further increase to third mode as in Figure 2.4, vibration of string will be divided into three equal length section, each vibrating in opposite to adjacent pieces. Nodal point is point with zero displacement and antinode is point where displacement is maximum.



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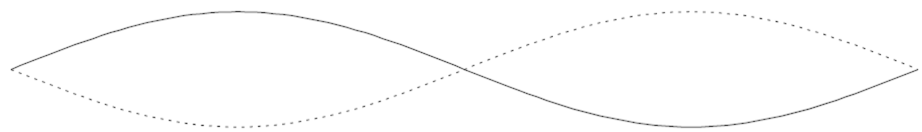

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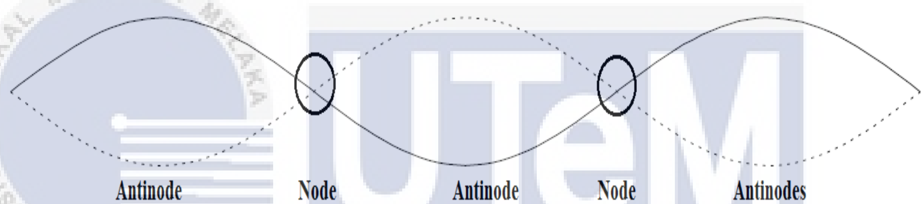


(c)

Figure 2.2: (a) First mode of vibration at fundamental frequency (b) Second mode of vibration (c) Third mode of vibration.

2.4 Terminology of Dynamic Vibration Absorber

Vibration absorber, also known as Dynamic Vibration Absorber (DVA), is a mechanical device used to reduce or eliminate unwanted vibration (Rao, 2010). It consists of another spring mass system that is attached to the primary structure for protection from vibration as shown in Figure 2.3. Due to its tunable characteristic, it is also referred as Tuned Vibration Absorber (TVA) or Tuned Vibration Damper (TVD). The term TVA is mostly adopted in papers that present studies about the tuning of absorber's stiffness. With proper amount of damping, TVA is the device that use for a structural resonance reduction by slightly tuned frequency. In the contrary, if the device with significantly low amount of damping and able to reduce the amplitude of vibrating structure to minimum, that device is known as vibration neutralizer.

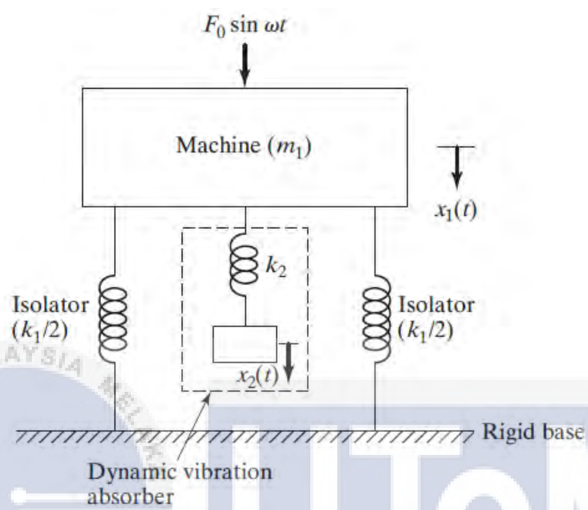


Figure 2.3: Attachment of DVA on primary structure (Rao, 2010).

2.5 Active Tuned Vibration Absorber

With the implementation of actuator force as active element, active DVA is used to provide active vibration suppression. An active structure is a structure with sensor and actuator that any changes detection by sensor will trigger the actuator to modify the response of the structure according to the environment. With proper control algorithm, the active DVA is more responsive to primary random disturbances. Due to its effectiveness in suppressing random disturbance, active tuned vibration absorber is used in suppression of vibration resulted from seismic activity (Nishimura, et. al., 1988). However, the disadvantage of active tuned vibration absorber is that it required larger energy and it required precise control method. Any control instability will reduce the effectiveness of active DVA (Jalili and Esmailzadeh, 2001).

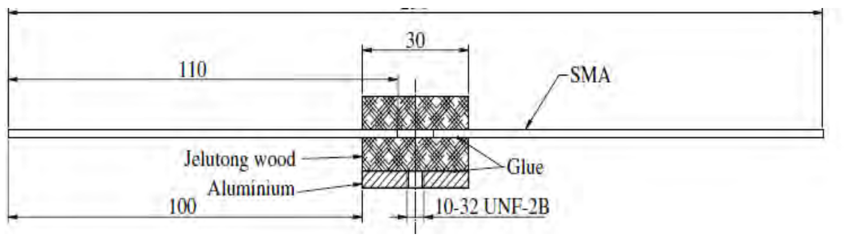
2.6 Adaptive Tuned Vibration Absorber (ATVA)

Adaptive Tuned Dynamic Vibration Absorber is introduced to overcome the weakness of active DVA which required high energy in absorber vibration control. Instead of exerting the force to change the response of absorber, method which required lower energy is introduced. The active force generator is replaced with variable element such as variable stiffness or damping (Jalili and Esmailzadeh, 2001). In ATVA, the stiffness of the absorber can be adjusted in real time. A few configuration of adaptive vibration neutralizers had been developed and their stiffness tuning method had been studied for its tuning ability. The spring component of these vibration neutralizers includes shape memory alloy (SMA), air spring (bellows), curved beam and cantilever beam. The linearity in adaptive tuned vibration absorber is more complicated as the linearity might arise from different factor such as electromagnetic force, mechanical properties of absorber material and etc.

2.6.1 Nonlinear Adaptive Vibration absorber based on Shape Memory Alloy (SMA)

Smart Material Alloy (SMA) is a material where the Elastic Modulus change nonlinearly with temperature, which contribute to the nonlinearity of SMA based vibration absorber. When the temperature of SMA increase, the elastic modulus increase as well, resulted in an increase of material stiffness and the natural frequency of the absorber. This indicate that Elastic Modulus as one of the changeable parameters in absorber's frequency tuning. By utilizing the mechanical behavior of this material, it is used in absorber for frequency tuning (Rushtighi et. al., 2004). Conducting the current through SMA will subsequently heat the beam and Young Modulus of SMA is altered and this resulted in changes in structure's natural frequencies. Figure 2.4(a) shows the

SMA configuration and Figure 2.4(b) shows the attachment of SMA vibration absorber on primary structure. Using same tuning method, ATVA using wired made out of SMA and insert eddy current as damping element is shown in Figure 2.5.



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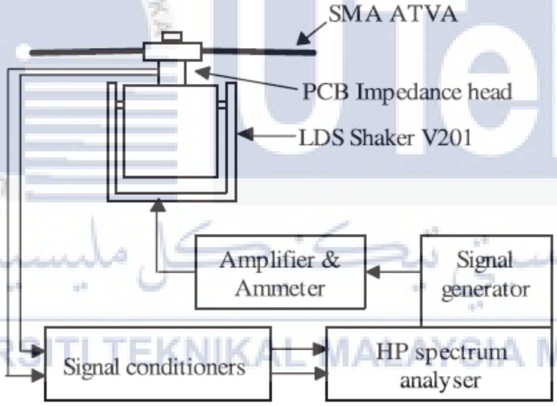


(b)

Figure 2.4: (a) Configuration of SMA (Rustighi et. al., 2004) (b) Attachment of SMA ATVA on Primary Structure (Rustighi et. al., 2004).

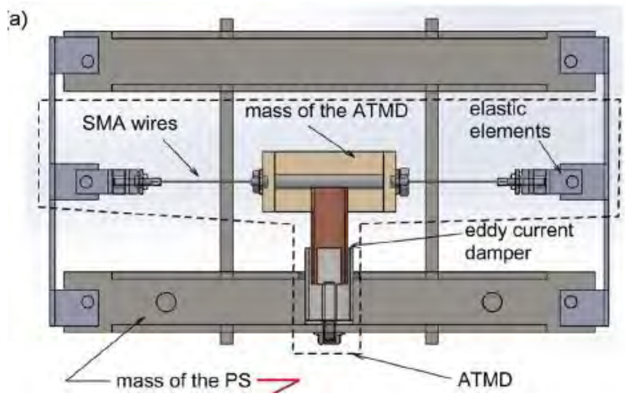


Figure 2.5: ATVA using wired made out of SMA and insert eddy current damping (Berardengo et. al., 2015).

2.6.2 Cantilever beam as Tuned Vibration Absorber

The Figure 2.6 shows cantilever beam with attached mass as tuned vibration absorber. The effectiveness stiffness of cantilever beam is described in equation (2.1), where *E* is the Elastic Modulus, *I* is the second moment of area, and *L* is the distance between the mass and the base. This indicates that the stiffness of the beam type absorbers can be altered by changing these parameters.

$$k = \frac{3EI}{L^3} \tag{2.1}$$

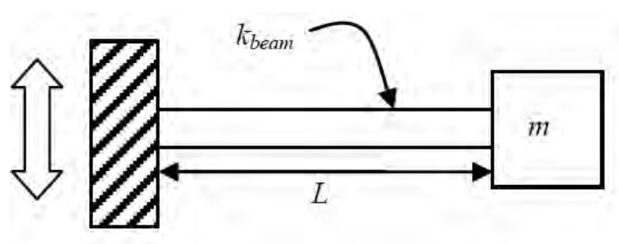
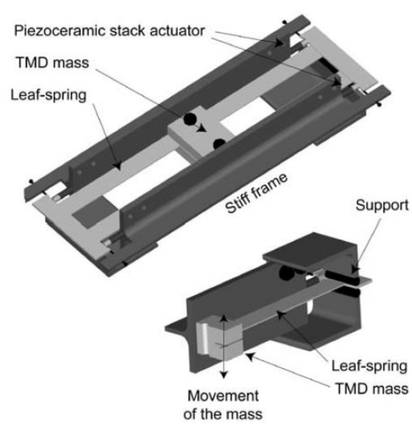
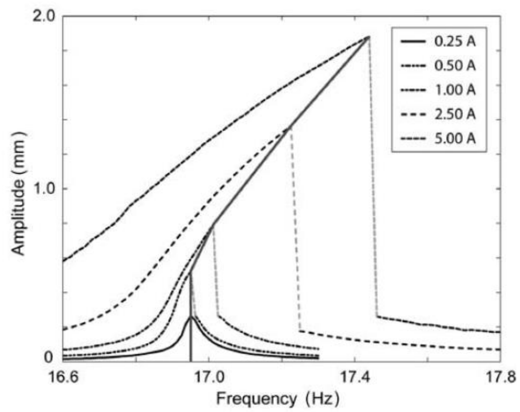


Figure 2.6: Cantilever beam with attach mass as tuned vibration absorber (Howard, 2009).

Figure 2.7(a) shows an adaptive tuned mass damper which consists of two parallel pre-stress able leaf spring with mass attached to the middle. Stiffness of the pre-stressed spring is controlled by applying normal force which is generated by piezoceramic stack actuators. Application of high voltage to piezecermic stack actuator will result in higher normal force which increase the stiffness. Increase in the stiffness will increase the natural frequency of the absorber. Figure 2.7(b) shows the nonlinearity using the leaf type adaptive vibration absorber. The nonlinearity arises from non-constant normal force in the leaf spring. Larger vibrational amplitude results in additional elongation in leaf spring. Using the same controlling theory, Adaptive Tuned Mass Damper (ATMD) is fabricate for aircraft turboprop noises suppression. In this cases, the piezoelectric actuation produce compressive force which will resulted S-shape in leaf spring. The length, geometrical dimension, second moment of inertia for leaf spring is considered in designing TMA (Keye et. al., 2009). The Figure 2.8 shows the design of absorber for aircraft turboprop noises suppression.



(a)



(b)

Figure 2.7: (a) Leaf type adaptive vibration absorber (Gsell et. al., 2007) (b) Nonlinear behavior of the system (Gsell et. al., 2007).

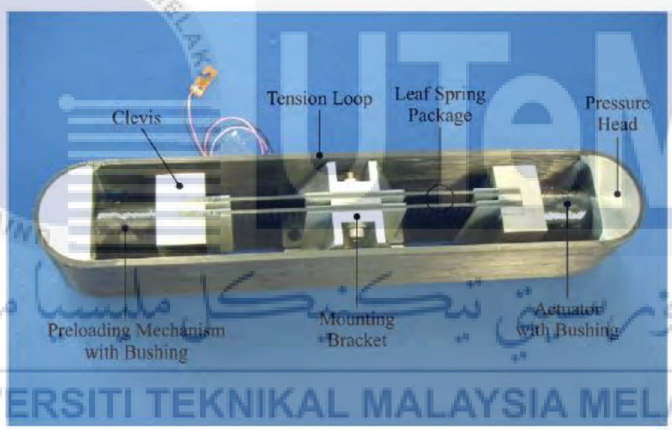


Figure 2.8: Design of absorber for aircraft turboprop noises suppression (Keye et. al., 2009).

A piecewise linear stiffness absorber is shown in Figure 2.9. The position of slider is set to 40 mm, 50 mm, 60 mm, and 70 mm. As shown in Figure 2.14, increase of position of slider, *h* will caused the blade spring to be constrained earlier, result in shorter length and increase the spring stiffness and finally the absorber frequency. The leaf spring is not constrained, the absorber behave as a linear system. However, the absorber start to behave non-linearly with increase of slider position as shown in Figure 2.15.

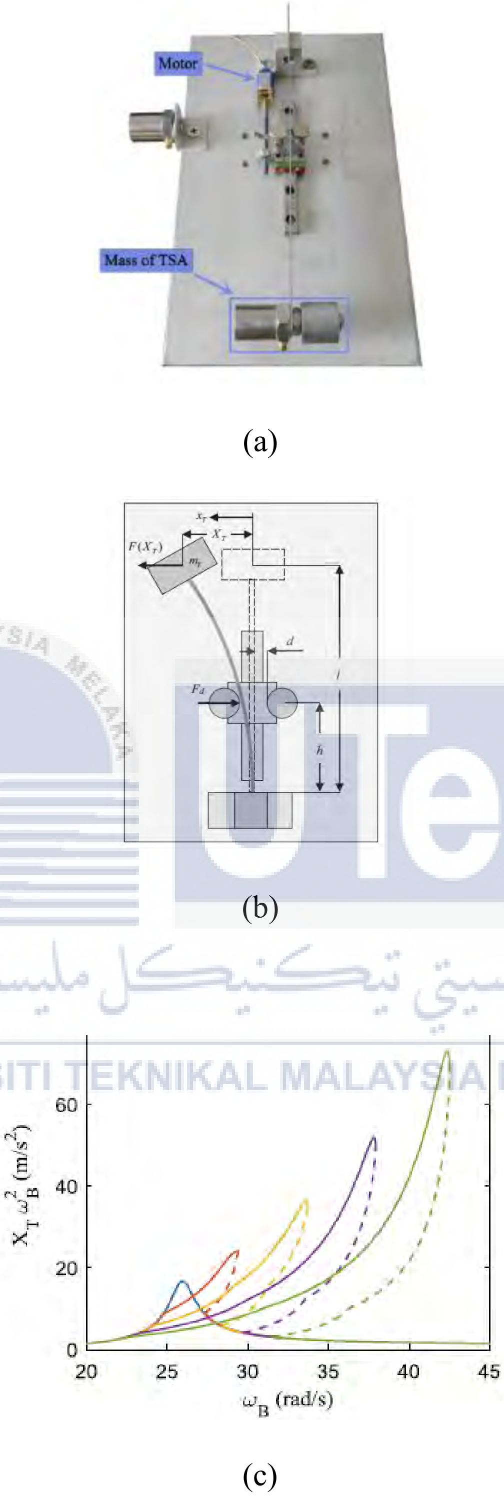


Figure 2.9: (a) Design of piecewise linear stiffness (Shui and Wang, 2018)

(b) Configuration of piecewiselinear stiffness absorber (Shui and Wang, 2018)

(c) Nonlinearity with increase of slider position (Shui and Wang, 2018).

Figure 2.10 shows a vibration absorber where two leaf spring are separated by actuator. By changing the distance between two leaf spring, the second moment of inertia, *I* can be altered.

Figure 2.11 shows a vibration absorber of thin beam with attached tip mass is activated by mounting two piezoelectric stack transducer in parallel to the beam. The frequency of absorber can be tuned in three ways. First, the absorber's stiffness can be altered by exerting a tensile normal force to the beam so as the absorber's stiffness is altered, the natural frequency also change. Second, the piezoelectric stack transducer can be tuned in out of phase to result in moment to bend the beam to alter the stiffness. Third, the piezoelectric stack transducer as element that contribute stiffness to the system. So any changes in stiffness of the spring will increase the stiffness of the absorber due to the parallel connection of the system.

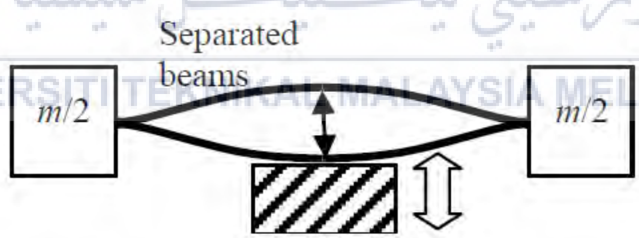


Figure 2.10: Leaf spring absorber (Howard, 2009).

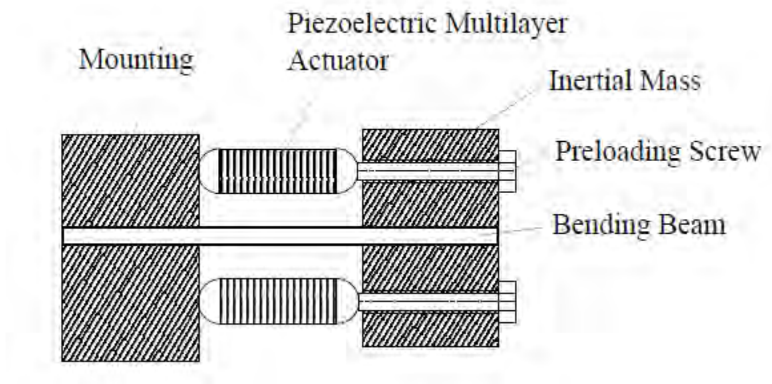


Figure 2.11: Design of vibration absorber (Herold and Mayer, 2016).

2.6.3 Implementation of Magnet in electromagnetic vibration absorber (EVA)

Figure 2.12 shows the electromagnetic vibration absorber (EVA), which consists of cantilever beam with flexible ferromagnetic, a ferromagnetic mass attached to the end of the beam and E-shaped electromagnet. The tuning of EVA is done by changing the current of the electromagnet, the interaction force between the ferromagnetic mass and electromagnet is varied. The design parameter of cantilever beam contribute passive stiffness to the system and the interaction of the ferromagnetic mass and electromagnet contribute to the tune-able stiffness to the system. The linearity of electromagnetic stiffness decreases with the increase air gap between.

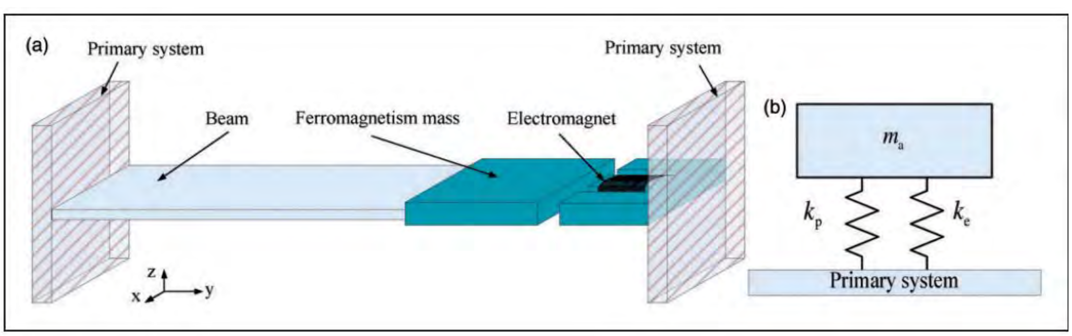


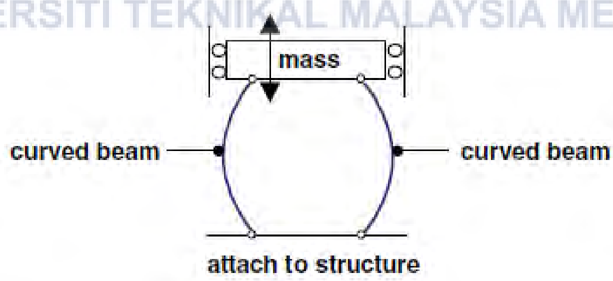
Figure 2.12: Design of electromagnetic vibration absorber (Liu et. al., 2014).

2.6.4 Curved Beams Tuned Vibration Absorber

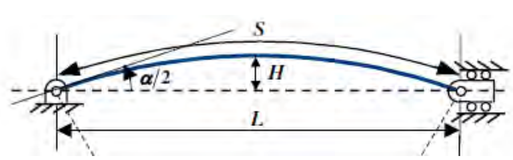
Figure 2.13(a) shows adaptive tuned vibration absorber which consists of two identical curve beam arranged in parallel connection. Actuated by the piezoelectric ceramic, the stiffness of the absorber is varied by changing the voltage of piezoelectric ceramic. The relationship between the stiffness and the crown height, H is described in (2.2).

$$\tilde{K} \approx \frac{2}{\pi^2 \frac{H^2}{S}} \tag{2.2}$$

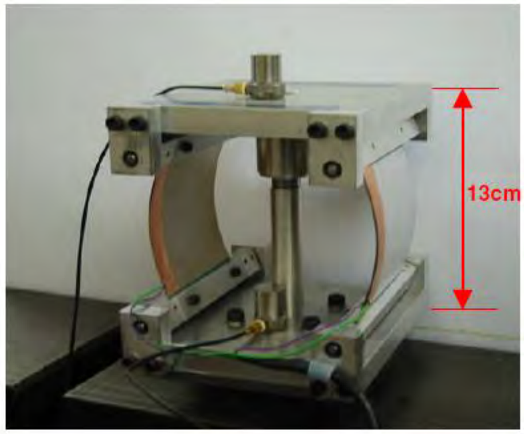
As the tuned frequency of the TVA is proportional to the square root of stiffness, inversely proportional to the crown height, H of the beams as shown in Figure 2.13(b). This indicates that higher bending load by piezoelectric ceramic will reduce the natural frequency of the absorber. The configuration of the piezoelectric ceramic is shown in Figure 2.13(c).



(a)



(b)



(c)

Figure 2.13: (a) Schematic diagram of dual beam absorber (b) Schematic diagram shows the crown height of the beam (c) Configuration of beam absorber.

2.7 Passive Tuned Vibration Absorber (PTVA)

In Passive Tuned Vibration Absorber (PTVA), stiffness is not able to be tuned in real time. In passive control, the linearity of the absorber can be distinguished more easily as the configuration is simpler as compared to adaptive vibration absorber.

2.7.1 Non Linear Tuned Vibration Absorber

An auto parametric vibration absorber using multiple pendulum is discussed (Vyas and Bajaj, 2001). By using different torsional springs, the different frequency mistuning is achieved and increase the effective bandwidth. Different method of designing nonlinear tuned vibration absorber is then studied using leaf spring is studied (Grappasonni et. al., 2014). Two different boundary conditions is studied in this paper, which is cantilever beam and doubly clamped beam. By using a thickness of 0.0005 m, a beam with nonlinear stiffness is designed. The bandwidth of the effective frequency can be broaden by increasing the mass proportion as shown in Figure 2.14. The frequency response in (a) is achieve by setting the mass ratio at 1 % and (b) is achieved by mass ratio at 5 %.

Other than using torsional spring and slender beam, the nonlinear effect in TVA can also be induced by implementation of magnet into vibration suppression system as shown in Figure 2.15. The stiffness of zigzag spring in the figure is provided by geometrical design. However, this metastructure it is not practical due to the modification of primary structure.

(Gratti, 2016) presented an analytical studies on vibration absorber harnessing the stiffness of the cubic spring. The frequency response of linear spring and nonlinear spring is plotted for comparison purpose as shown in Figure 2.16. It can be observed that the frequency ratio of nonlinear spring is higher than the linear spring. The nonlinearity of conical spring arise from the unequal diameter of the spring (Polukoshko, 2013), shown in Figure 2.17. Due to its nonlinearity, tuning is achieved by pre-compression conical spring to obtain desired frequency suppression (Qiu et. al., 2017).

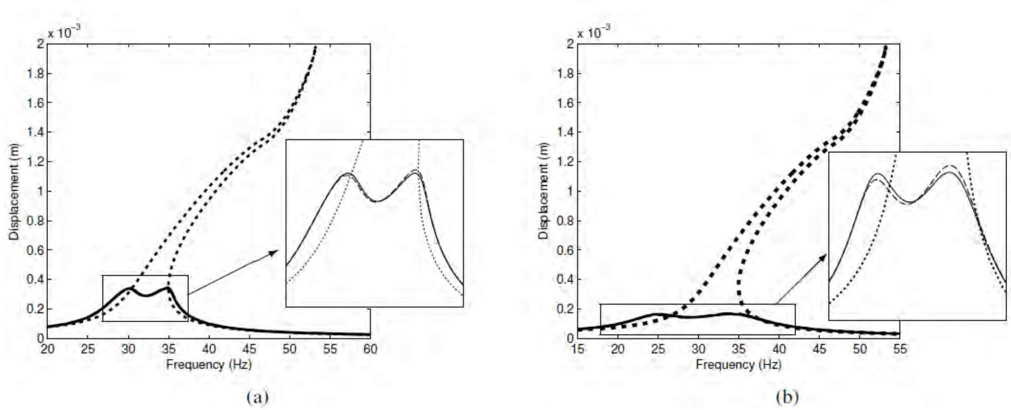


Figure 2.14: Comparison of linear and nonlinear absorber with (a) mass ratio of 1 % (Grappasonni et. al., 2014) (b) mass ratio of 5% (Grappasonni et. al., 2014).

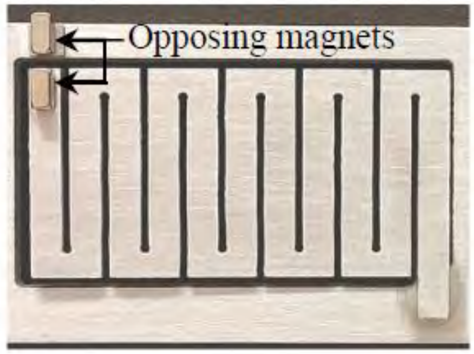


Figure 2.15: Attachment of magnet on metastructure structure (Hobeck and Inman, 2015).

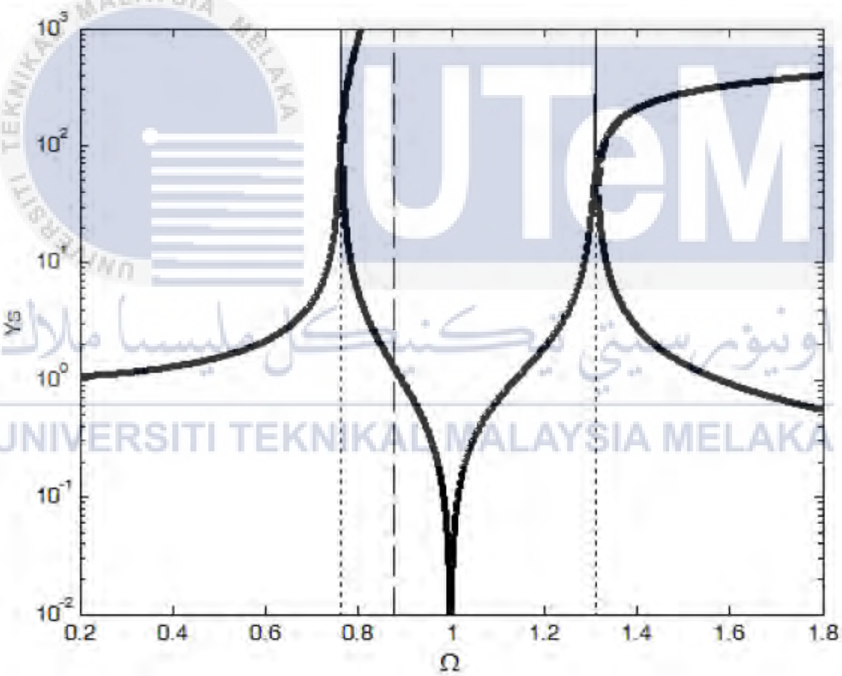


Figure 2.16: Comparison between linear stiffness and nonlinear stiffness (Gratti, 2016).

	Spring stretching									
Force F, N	5.0	6.0	9.0	10.0	11.00	14.	15.0	16.0	19.0	20.0
Deformation x, mm	14.0	17.0	26.0	28.5	31.0	39.0	42.0	45.0	54.0	57.0
	Spring compression									
Force F, N	5.0	6.0	9.0	10.0	11.0	14.0	15.0	16.0	19.0	20.0
Deformation x, mm	14.0	16.0	24.0	25.0	27.0	33.0	35.0	37.0	40.0	41.0

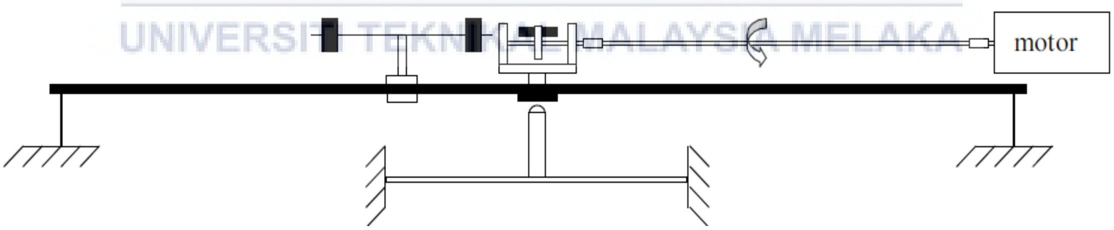
Figure 2.17: Nonlinearities in cubical spring (Polukoshko, 2013).

2.7.2 Linear Tuned Vibration Absorber

The first device for damping vibration of bodies is designed by Frahm to prevent the rolling motion of ship and to remove unwanted vibration (Frahm, 1909). Since then, extensive research have been done to improve the performance of vibration absorber. An analytical studies is conducted to study the effect of distribution of absorber mass on vibration suppression performance (Jalili and Esmailzadeh, 2001). A model of double ended cantilever beam clamped with two masses is proposed in the study. A similar study is conducted in the later by analytical and experimentation. Figure 2.18 shows the configuration of suspended multi-beam absorber for vibration suppression. In the Figure 2.18, the purpose of the mass attached in the middle is to act as the clamp support for two of the beams. Due to non-tune-able characteristic of this absorber, analytical studies is conducted to determine the absorber's frequency based on the designed parameter and tuning have to be made from time to time. The tuning of absorber can be done in two ways, which is changing the absorber mass and length of the leaf spring.

Studies related to the point of attachment of tuned vibration absorber is also studied (Bonsel, 2003). The Figure 2.27 shows the schematic diagram of experimental set up of piecewise linear absorber. The absorber is designed in a way that can moved in the parallel direction of primary structure manually. The Figure 2.28 shows the design of the absorber. The natural frequencies of absorber can be tuned by moving the masses that clamped on the steel plate.

Figure 2.18: Configuration of suspended multi-beam absorber for vibration suppression (Faal et. al., 2012).



(a)

(b)

Figure 2.19: (a) Schematic diagram of experimental set up of piecewise linear absorber (Bonsel, 2003) (b) The design of the absorber. (Bonsel, 2003).

Another type of tuned vibration absorber consists of leaf spring and helical spring is shown in Figure 2.20 to suppress the vibration suppression by providing rotational and translational motion. A fixed helical spring of 900 N/m is chosen and the rotational absorber is made up of fixed leaf spring with two rectangular aluminum plates. The attachment of these springs is shown in Figure 2.21.

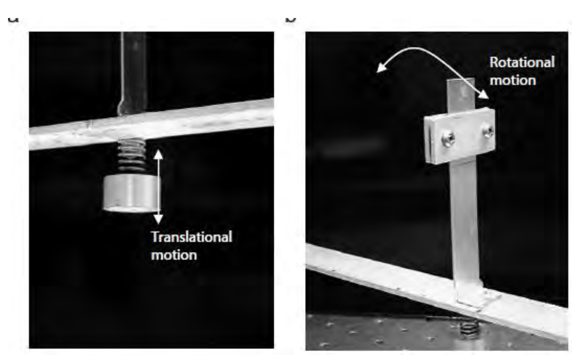


Figure 2.20: Vibration absorber consists of leaf spring and helical spring (Wong et. al., 2007).

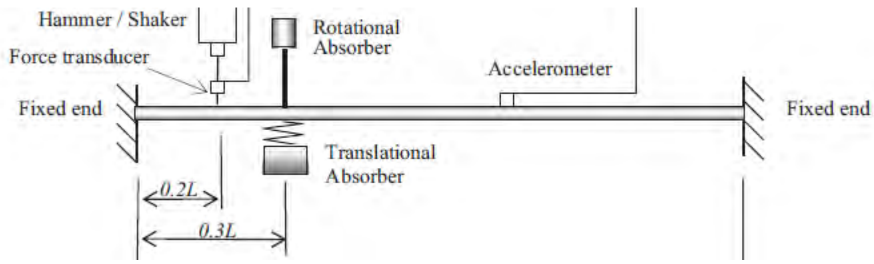


Figure 2.21: Attachment of spring on clamped-clamped beam (Wong et. al., 2007).

Attachment of vibration absorber on the primary structure will result a two degree of freedom with two new natural frequencies as shown in Figure. Solution of increasing the bandwidth using multiple array of local resonator is studied analytically (Xiao et. al., 2012). Eight LR beams are tuned to different resonance frequency respectively, resulted in different peak and increase of bandwidth is shown in Figure 2.31. By means of analytical studies and simulation, they demonstrated the possibility of attachment of multiple arrays of resonator in order to achieve broader gaps in vibration attenuation. In their later paper, they studied and concluded that the three keys parameter in absorber frequency tuning is length, absorber mass and stiffness (Xiao et. al., 2013). In both paper, only theoretical studies is conducted.

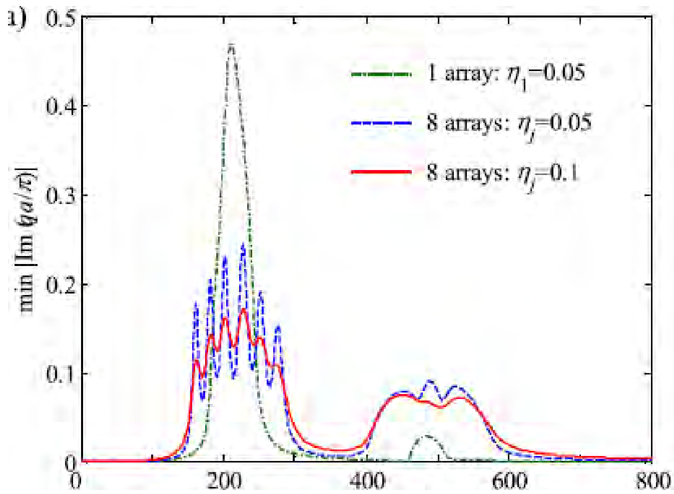


Figure 2.22: Response graph (Xiao et. al., 2012).

CHAPTER 3

METHODOLOGY

3.1 Overview

In this chapter, procedure of the research methodology will be discussed and presented. The methodology include analytical study, DVA characterization and performance assessment of designed and fabricated vibration absorber on vibration suppression. The procedure is started with analytical study to determine the natural frequency at the first two modes of vibration of cantilever beam. Next, the concept design of DVA is presented and impact testing will be conducted. Impact testing is conducted to determine the natural frequency of fabricated DVA. Operating Deflection Shape will be plotted to visualize the mode shape of beam and determine the points with maximum vibration amplitude. Then, experiment will be carried out to assess the performance of designed DVA in suppressing the vibration of beam. The Figure 3.1 shows the deployment flow chart that summarize the steps of research work.

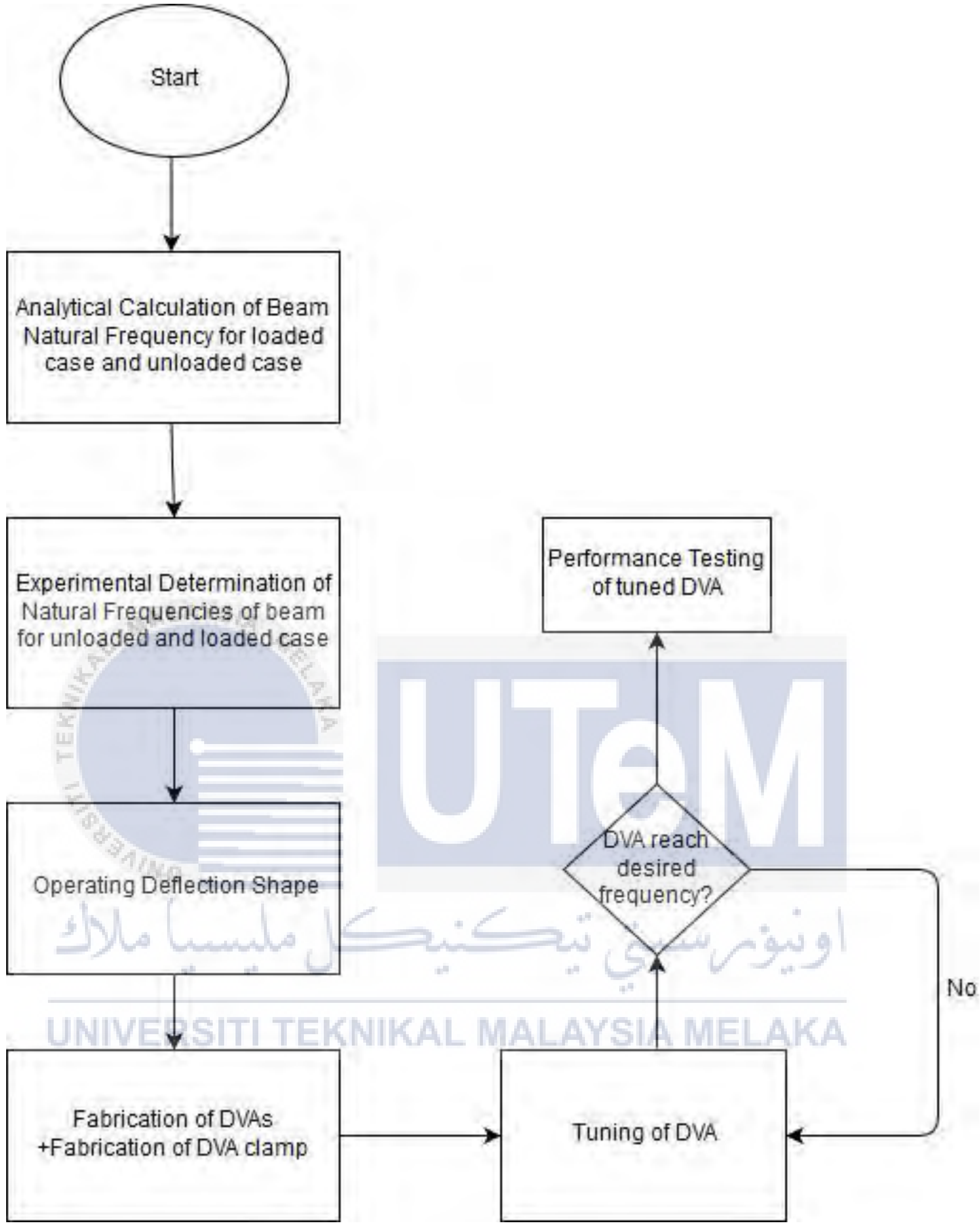


Figure 3.1: Summarized Deployment Flowchart.

3.2 Theoretical Calculation of Cantilever Beam

In continuous system such as beam, there are infinite points of degree of freedom. So analytical approach will be adopted to determine the frequency of beam. In this paper, only first and second mode of vibration of beam will be considered. Consider the cantilever beam shown in Figure 3.2, using the available formula derived from Euler- Bernoulli theory, the natural frequency at first and second mode are determined. The parameters of cantilever beam is listed in Table 3.1.

Figure 3.2: Cantilever Beam (Image Adopted From www.dailycivil.com).

Table 3.1: Parameter of the beam.

Parameter	Description	Value	Unit
L	Length	0.65	m
w	Width	0.05	m
h	Height	0.01	m
A	Cross-sectional Area	5×10^{-4}	m^2
E	Elastic Modulus	200	Gpa
I	Area Moment of Inertia	4.17×10^{-9}	m^{-4}
ρ	Density	7870	kg/m^3

3.2.1 Calculation of Natural Frequencies of Unloaded Beam

For the first mode of vibration, the mode shape are shown in Figure 3.3, the formula is given by

$$\omega_n = \beta l^2 \sqrt{\frac{EI}{\rho A L^4}} \tag{3.1}$$

Where,

$$\beta l = 1.875$$

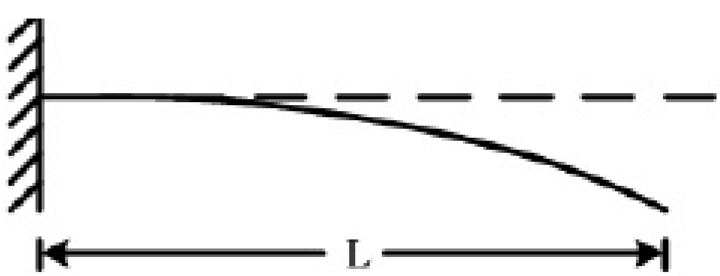


Figure 3.3: Mode shape of first mode of vibration (Image Adopted from ResearcherGate.net).

For the second mode of vibration, the mode shape are shown in Figure 3.5, the formula is given by

$$\omega_n = \beta l^2 \sqrt{\frac{EI}{\rho A L^4}} \tag{3.2}$$

Where,

$$\beta l = 4.694$$

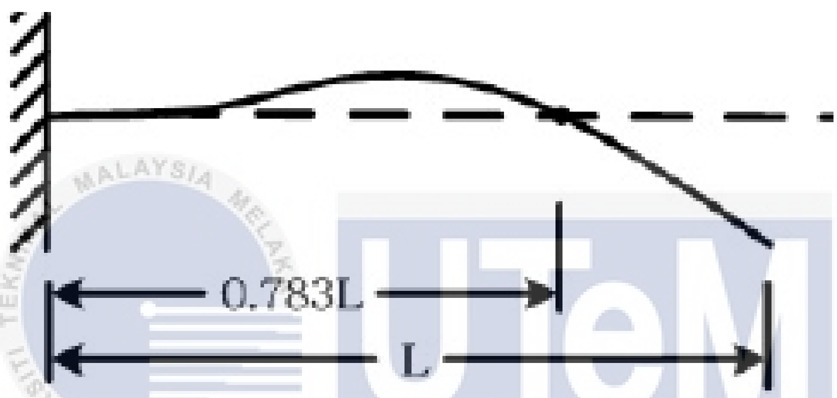


Figure 3.4: Mode shape of second mode of vibration (Image Adopted from ResearcherGate.net).

3.2.2 Calculation of Natural Frequencies of Loaded Beam

The DVA clamps are fabricated for the purpose of mounting DVAs to primary beam. The mounting of DVA clamps results in changes of the effective mass to the vibrating system of the beam, which caused the natural frequencies change as well. The changes in natural frequencies of the beam can be calculated using the following formulae:

$$m_{eff} = \frac{3EI}{L^3 \omega_{unloaded}^2} \tag{3.3}$$

$$M = m_{eff} + m_{added} \tag{3.4}$$

$$w_{loaded} = \sqrt[2]{\frac{3EI}{L^3 M}} \tag{3.5}$$

3.3 Excitation Technique of Beam

Figure 3.5 shows the experiment to determine the natural frequencies at the first two modes of loaded beam. In the figure, a mechanical shaker (TIRA GmbH, type S 50018), stinger (length of 8 cm, diameter of 0.6 mm), amplifier, 4 input channels signal analyzer (Data Physics SignalCalc ACE Dynamic Signal Analyzer), force sensor (Dytran, sensitivity, 455.2 mV/lbf), and accelerometer (Dytran, sensitivity, 10.34 mV/g) is used in the determination of natural frequencies. With an interface software of Data Physics installed in computer, it is able to choose the type of signal waveform generated to excite the beam. In this case, swept sine ranging from 10 Hz to 200 Hz is selected to excite the beam. The frequency range is selected at 10 Hz to 200 Hz is based on the calculated natural frequencies of the beam, which falls between the selected range. Figure 3.6 shows the interface of the software. The signal is amplified by the amplifier and excitation force will be transfer to the beam via stinger.

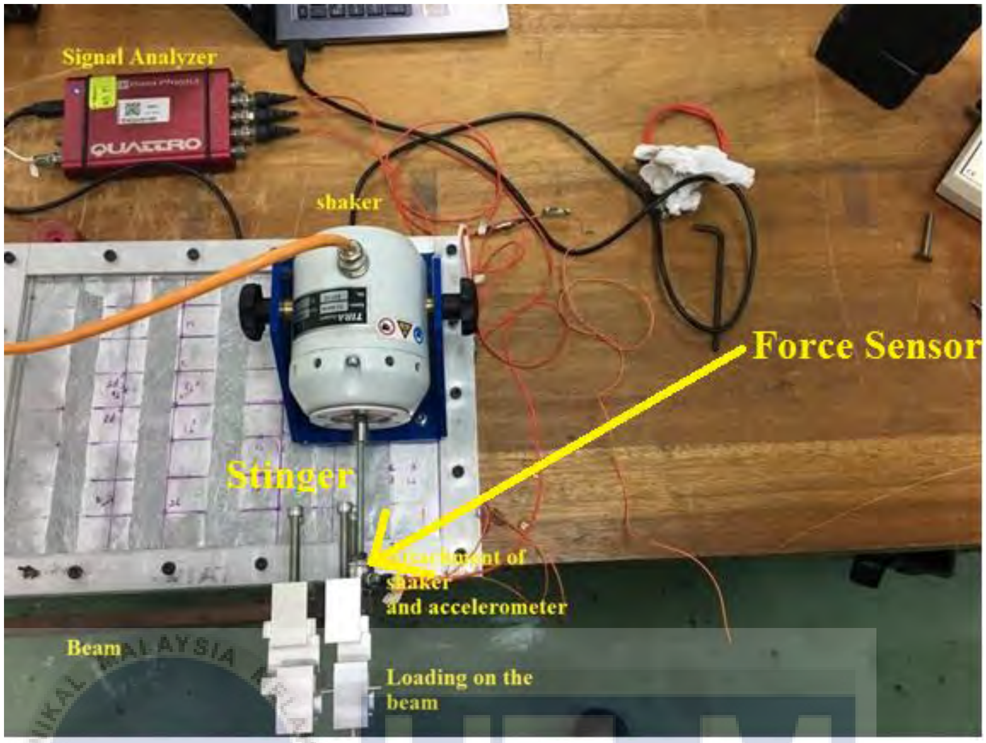


Figure 3.5: Experiment setup for determine natural frequency of loaded beam.

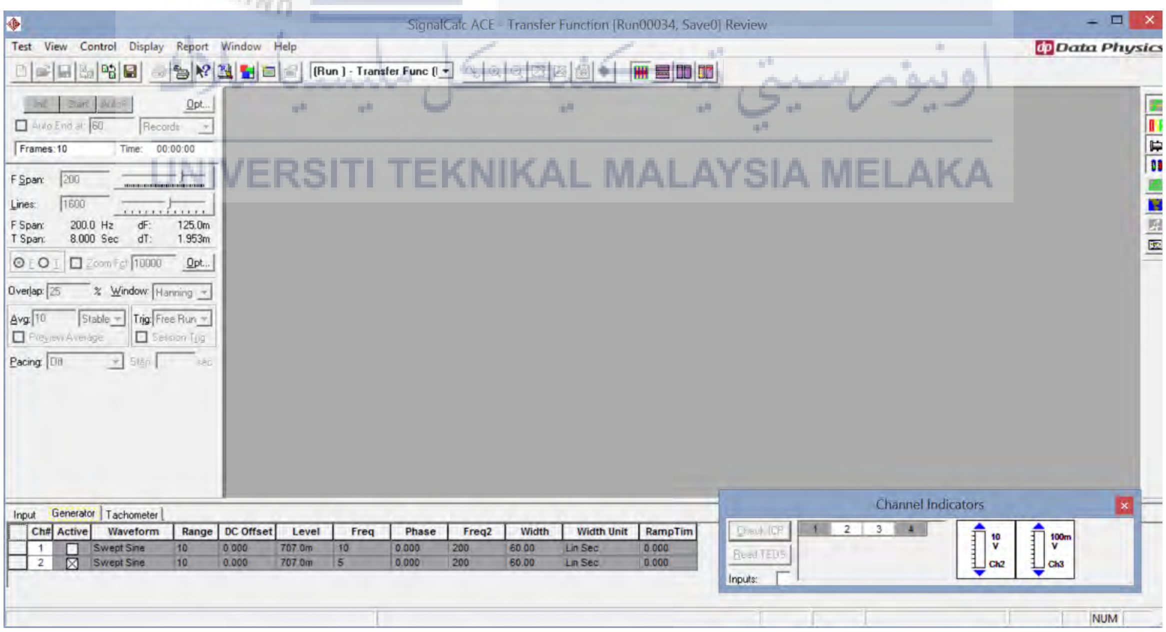


Figure 3.6: Software interface of Data Physics Analyzer.

3.4 Operating Deflection Shape

Operation Deflection Shape (ODS) analysis is a method used to study the vibration pattern of a vibrating structure. It is the relativity of motion between points in a structure. The obtained result in ODS analysis is used to determine the point for maximum deflection. To conduct ODS analysis, the beam is marked with number at specific points as constructed in the software as shown in Figure 3.7. This is for the purpose of roving of accelerometer to obtain data that describe the phase and magnitude at each defined points. By input the phase and magnitude for each point, visualization of the mode shape of cantilever beam can be visualized and determine the point with maximum vibrating amplitude. In ODS, increase of defined point allowed the visualization of beam vibration pattern at more smooth manner and determine the point with highest vibration amplitude. So for a 0.65 m beam, the distance between defined points is 0.04 m, so there will be 16 points defined. Based on the mode shape of beam, the deflection of points is set to be vibrate at Y-axis.

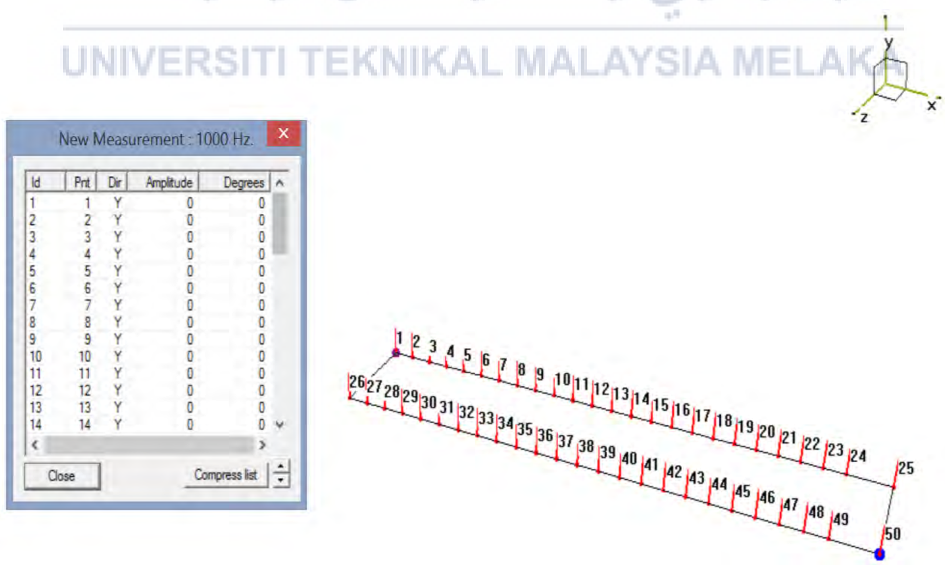


Figure 3.7: Examples of defined Points on Beam Using Available Software.

3.5 Conceptual Design of DVA

After the first and second mode of vibration with its respective frequency is determined, the conceptual design of DVA is proposed. In this paper, cantilever spring vibration absorber is designed. The objective for designing multiple DVAs is to increase the effective frequency bandwidth of linear absorber. So, multiple vibration absorbers will be designed and two vibration absorbers are chosen. In designing these vibration absorbers, certain parameters have to be constrained. The formulas involved in the design of absorbers are

$$\omega_f = \sqrt{\frac{k_a}{m_a}} \tag{3.6}$$

Where,

ω_f = Natural Frequency of Absorber, rad/s

k_a = Elastic Stiffness of Beam, N/m

m_a = Mass of Absorber, kg

$$K_a = \frac{3EI}{L^3} \tag{3.7}$$

Where,

E = Elastic Modulus of Material, GPa

I = Moment of Inertia, m^4

L = Length of Beam, m

$$\rho = \frac{m}{V} \tag{3.8}$$

Where,

ρ = Density of material, kg/m^3

m = mass, kg

V = Volume, m^3

With formulas stated, design natural frequency of absorbers is achieved by varying tune-able parameter that is length. The natural frequency of absorbers is tuned based on the natural frequencies of loaded beam. The table shows the tuning frequency of each DVA at first and second natural frequency. Using Equation 3.6 and Equation 3.7, the theoretical length of each DVA is calculated.

Table 3.2: Information for each DVA.

At 1st Natural Frequency				At 2nd Natural Frequency	
Absorber	Unit	DVA1	DVA2	DVA3	DVA4
Desired Natural Frequency, ω_f	Hz	14.1	13.9	101	97.5
Mass, *m*	G	40	50	20	20
Stainless steel strip thickness, *t*	M	0.001	0.001	0.001	0.001
Width of strip, *w*	m	0.032	0.032	0.042	0.042
Modulus of stainless steel, *E*	Gpa	205	205	205	205
Second Moment of Inertia, *I*	m^4	2.6667×10^{-12}	2.6667×10^{-12}	3.4167×10^{-12}	3.4167×10^{-12}
Calculated Length of Tuning, *L*	m	0.17	0.16	0.063	0.065

3.6 Design and fabrication of the Absorber Clamp

Figure 3.8 shows the mounting device used to attach the fabricated DVA to the beam. Aluminum is adopted as the design material of this mounting due to its lightweight properties. The total weight of the absorber is 0.408 kg, which is 16 % of the mass of the beam. From the figure, there is two identical end of the absorber which perform the same function, the first end is used to clamp the fabricated cantilever spring absorber and the other end is used to provide mounting at the beam.

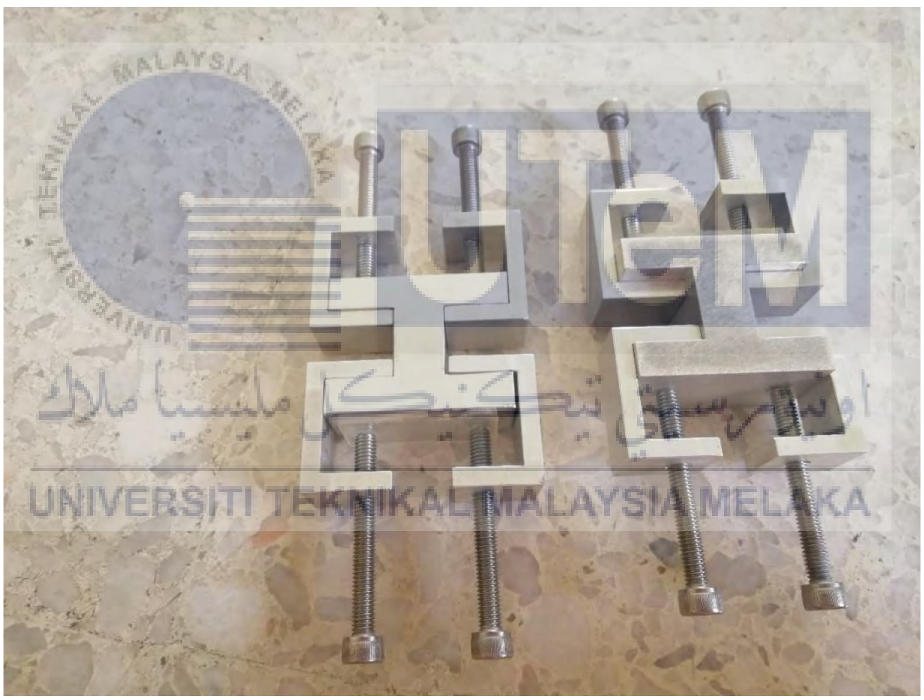


Figure 3.8: Two DVA mounting with total mass of 0.408 kg.

3.7 Tuning of DVAs using Impact Testing

Impact testing is the measure response of impact of the object, and it also known as bump testing. In this paper, impact testing is used to determine the natural frequency of fabricated vibration absorber. Basically, impact testing is conducted by using impact hammer to excite the selected beam and accelerometer is attached close to the point of impact to measure the excited natural frequency of the object. Impact testing will be carried out to identify the natural frequencies of fabricated DVAs. Figure 3.9 shows the schematic diagram of the configuration of the experiment. The red dot indicate the point of impact by the impact hammer while the black dot is the point of accelerometer attachment. The point of impact should be at the same spot for each time. In attachment of accelerometer to selected beam using wax, the amount of wax should be precise and minimum as over applied might affect the frequency measured. In this experiment, the force is neither controlled nor measured. Hard tip is proposed in this experiment as soft tip will required harder impact which might affect the linearity of the fabricated DVA. To determine the natural frequency of vibration, graph of Frequency Response Function should be referred. On the phase-frequency plot, the resonance can be spotted when there is sudden increase of amplitude and 90° phase change in the response. The length will be adjusted until the vibration absorbers reach its desired frequency. Figure 3.10 shows the impact testing of the vibration absorbers. The instruments used for the impact testing are 4 input channels signal analyzer (Data Physics SignalCalc ACE Dynamic Signal Analyzer), impact hammer (Dytran, Sensitivity, 10 mV/lbf), accelerometer (Dytran, Sensitivity, 10.43 mV/g) and computer.

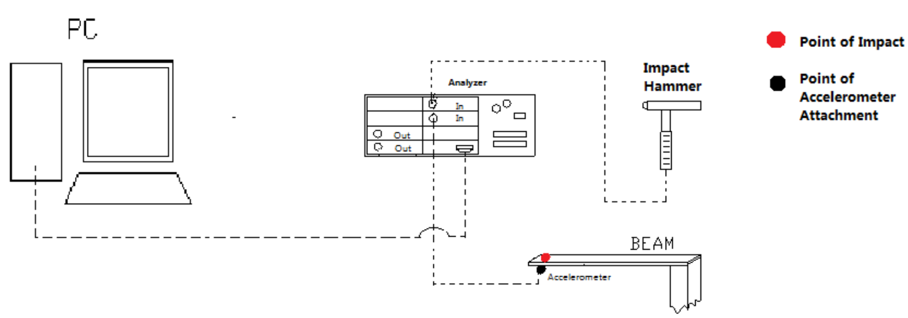


Figure 3.9: Schematic diagram of the configuration of the experiment.

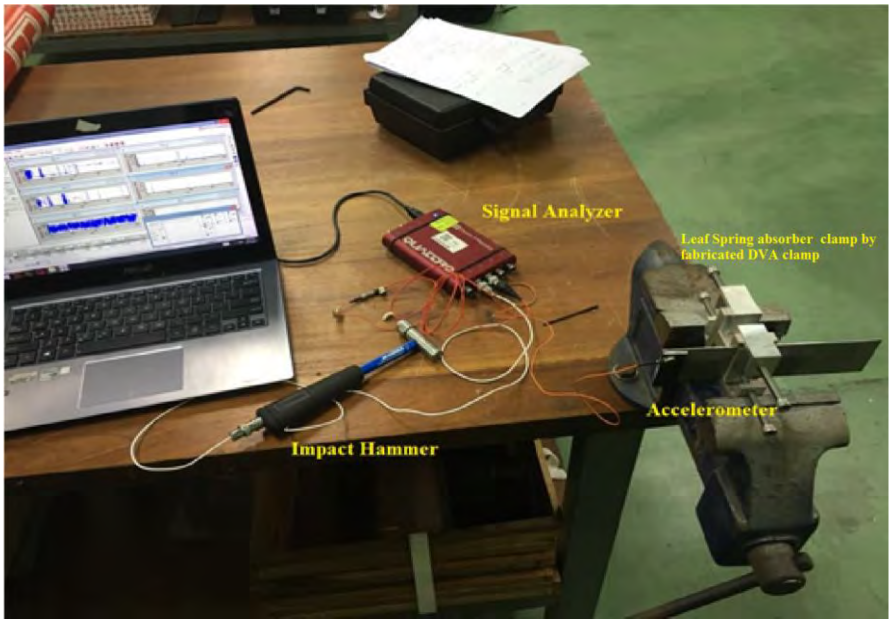


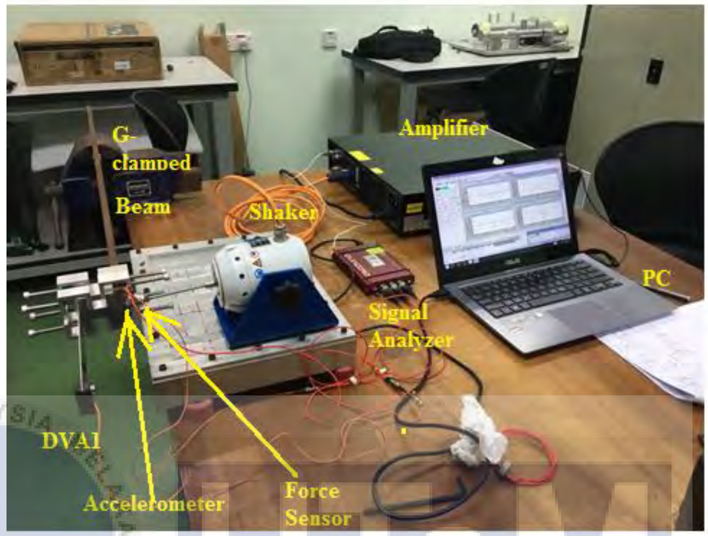
Figure 3.10: Impact Testing of Fabricated DVA.

3.8 Performance Testing of fabricated DVAs

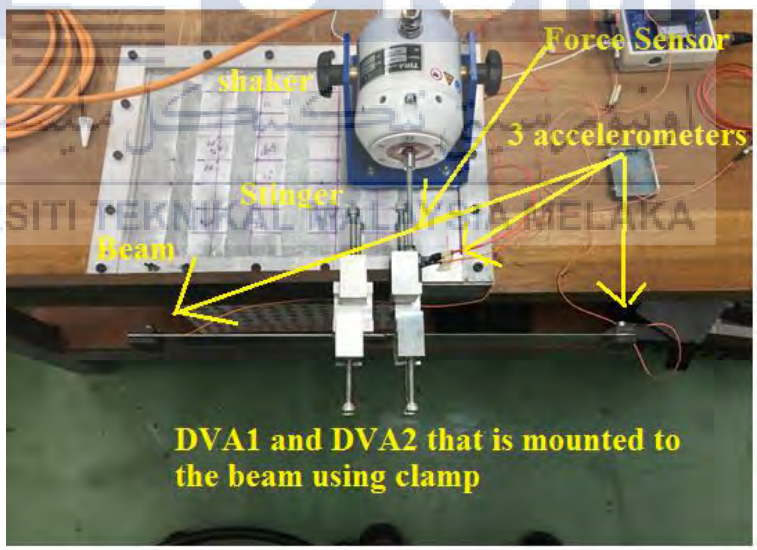
In performance testing of fabricated DVA, the testing will be categorized into first natural frequencies and second natural frequencies.

3.8.1 Performance testing of fabricated DVA at 1st Natural Frequencies

Figure 3.11a shows the configuration of performance testing of Single DVA, the instruments are mechanical shaker (TIRA GmbH, type S 50018), stinger (length of 8 cm, diameter of 0.6 mm), 4 input channels signal analyzer (Data Physics SignalCalc ACE Dynamic Signal Analyzer), force sensor (Dytran, Sensitivity, 455.2 mV/lbf), 2 accelerometers (Dytran, Sensitivity, 10.43 mV/g and 10.34 mV/g). The force sensor is mounted at the stinger before adhere to the tip of the beam. The excitation point is located at the tip of the beam, which is 0.65m from the clamped point. Figure3.11b shows the configuration of performance testing of multiple DVAs. In this configuration, there are 3 accelerometer used.



(a)

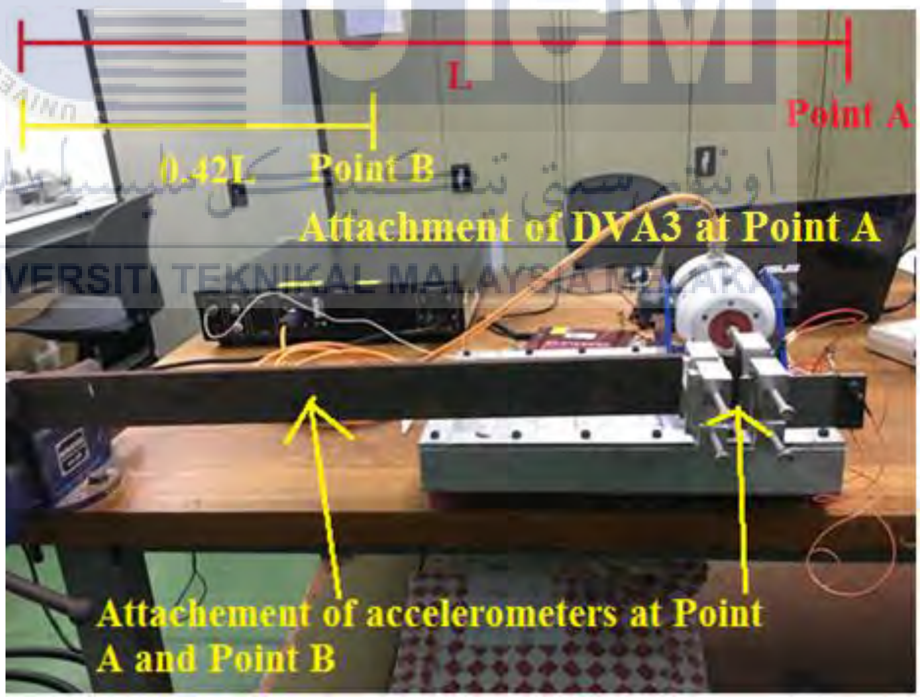


(b)

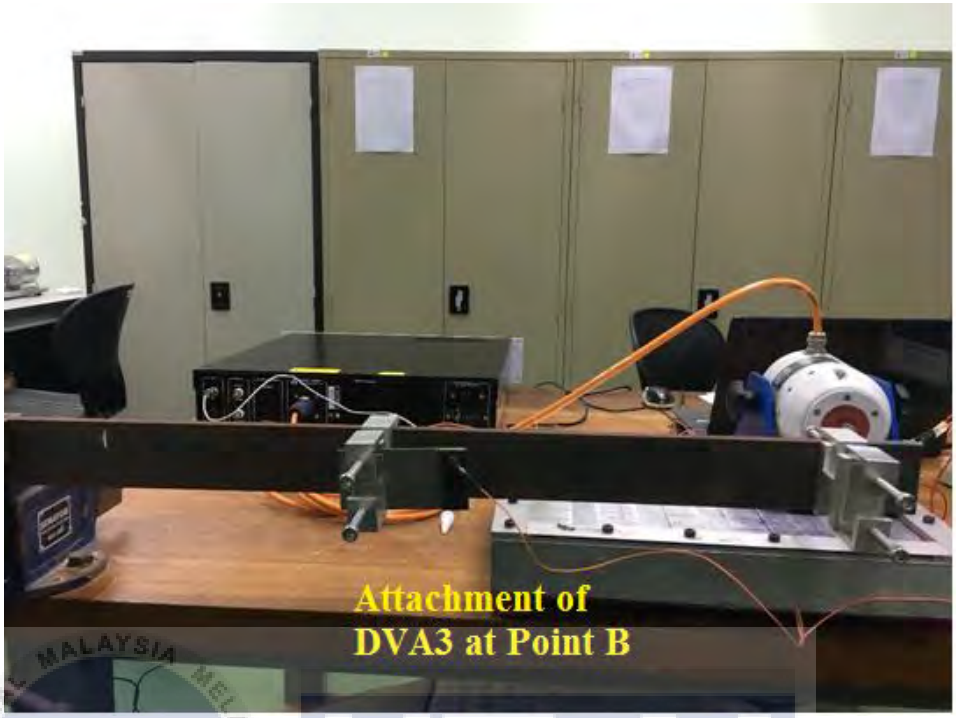
Figure 3.11: (a) Experiment Configuration of Single DVA (b) Experiment Configuration of Multiple DVA.

3.8.2 Performance testing of fabricated DVA at 2nd Natural Frequencies

Figure 3.12a and Figure 3.12b shows the configuration of performance testing of Single DVA at point A and point B, using the instruments are mechanical shaker (TIRA GmbH, type S 50018), stinger (length of 8cm, diameter of 0.6 mm), 4 input channels signal analyzer (Data Physics SignalCalc ACE Dynamic Signal Analyzer), force sensor (Dytran, Sensitivity, 455.2 mV/lbf), 3 accelerometers (Dytran, Sensitivity, 10.43 mV/g, 10.34 mV/g and 10.00 mV.g). Same as the configuration in performance testing at first natural frequency, the excitation point is located at the tip of the beam, which is 0.65 m from the clamped point. The only difference is the DVA3 of different length and 2 accelerometer is used, located at the Point A and Point B.



(a)

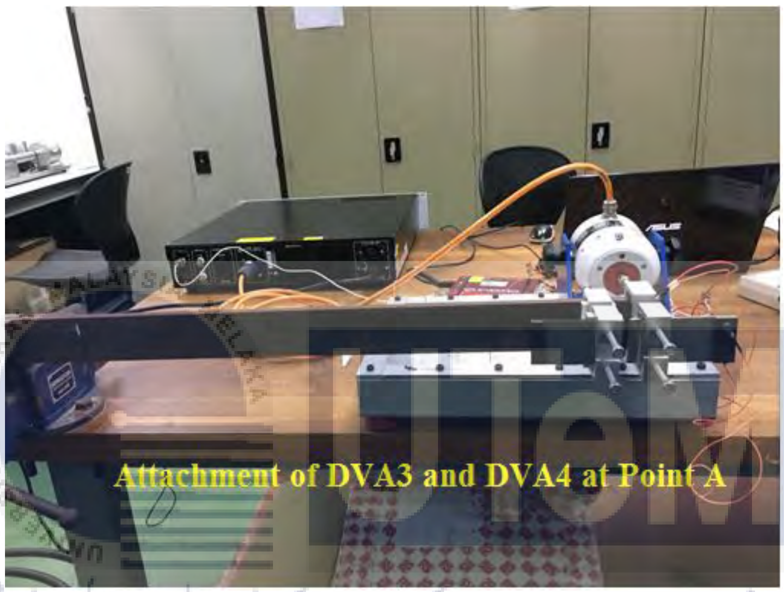


(b)

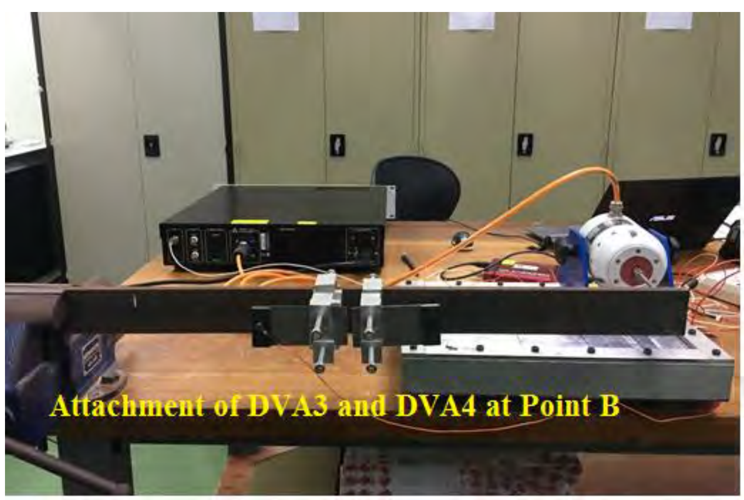
Figure 3.12 (a) Experiment configuration of single DVA at point A (b) Experiment configuration of single DVA at point B.

Figure 3.13a shows the configuration of performance testing of multiple DVA at point A and point B, with the same instruments, mechanical shaker (TIRA GmbH, type S 50018), stinger (length of 8 cm, diameter of 0.6 mm), 4 input channels signal analyzer (Data Physics SignalCalc ACE Dynamic Signal Analyzer), force sensor (Dytran, Sensitivity, 455.2 mV/lbf), 2 accelerometers (Dytran, Sensitivity, 10.43 mV/g, 10.34 mV/g). Using the same configuration as in performance testing of DVA3 at second natural frequency, the excitation point is located at the tip of the beam, which is 0.65 m from the clamped point. In this case, the accelerometers is still attached in point A and Point B. The only difference is application of DVA3 and DVA4 on the beam. Figure

3.13a shows the concentrated attachment of DVA3 and DVA4 at the point A and Figure 3.13b shows concentrated attachment of DVA3 and DVA4 at point B. In Figure 3.13c, the DVAs are distributed where DVA3 is attached at point A and DVA 4 is attached at point B.



(a)



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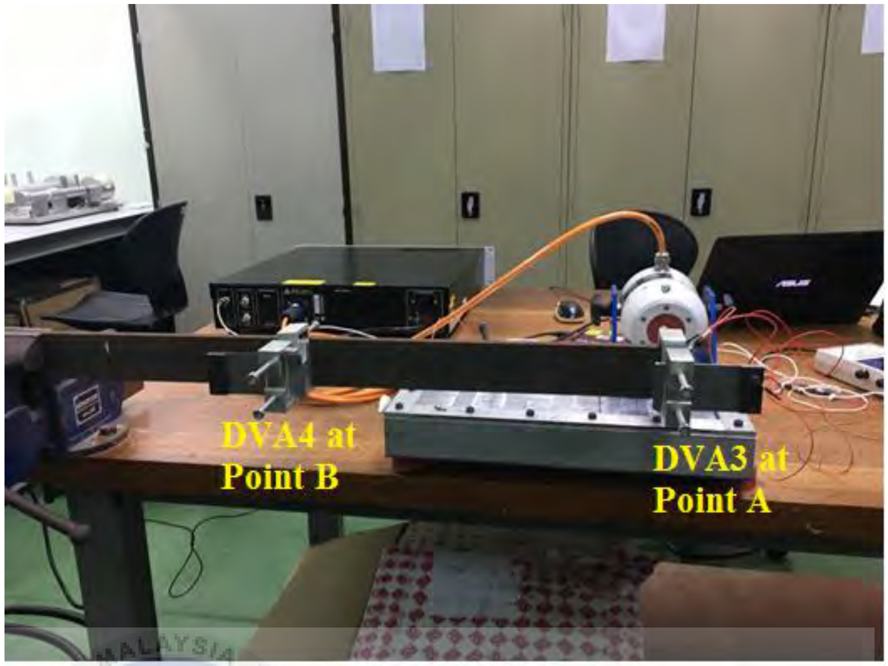


(c)

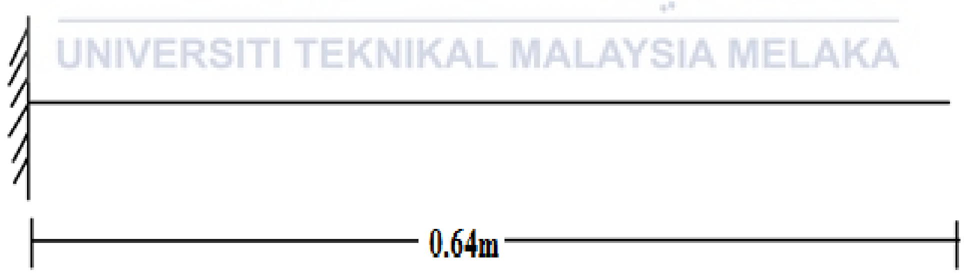
Figure 3.13: Experiment configuration of multiple DVAs at point A (b) Experiment configuration of multiple DVAs at point B (c) DVAs are distributed where DVA3 is attached at point A and DVA 4 is attached at point B.

CHAPTER 4

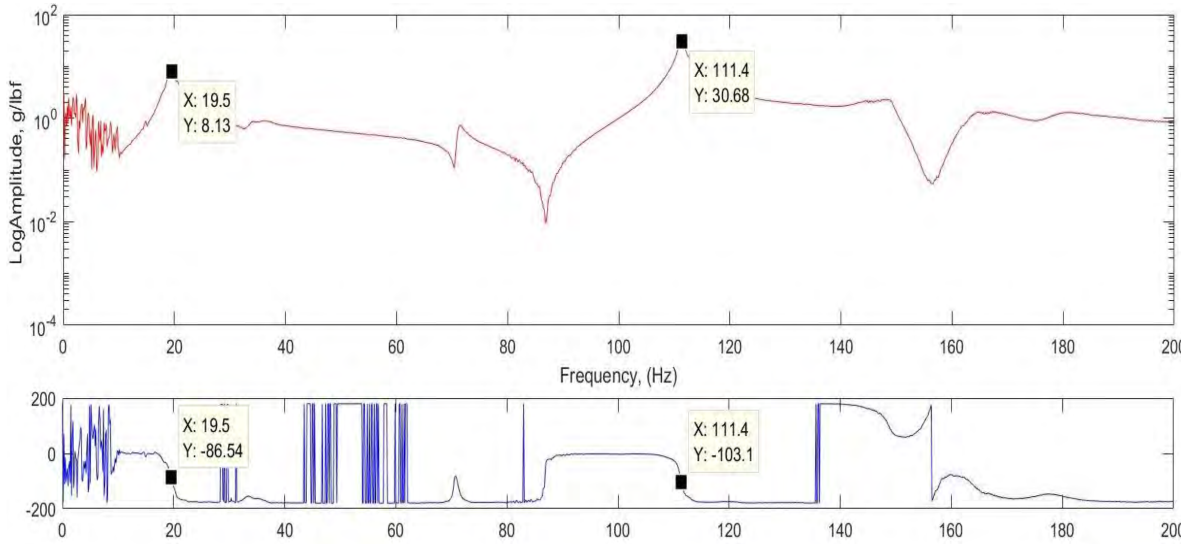
RESULT AND DISCUSSION

4.1 First and Second Natural Frequency of loaded and unloaded Beam

Figure 4.1a shows the schematic diagram of unloaded beam with length of 0.64 m and Figure 4.1b shows the natural frequency of the beam at 1st and 2nd mode of vibration. From the result, the natural frequencies of unloaded beam are 19.5 Hz and 111.4 Hz. This justification is based sudden increase of vibration amplitude and phase difference of approximate 90°. From the Figure 4.1b, the vibration amplitude is relatively higher at 19.5 Hz and 111.4 Hz, which are 8.13 g/lbf and 30.68 lbf while the phase difference at 19.5 Hz and 111.4 Hz is around 86.54° and 103.1° respectively.



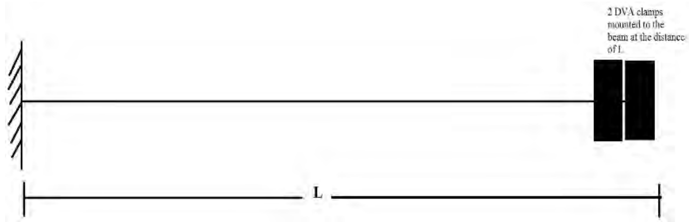
(a)



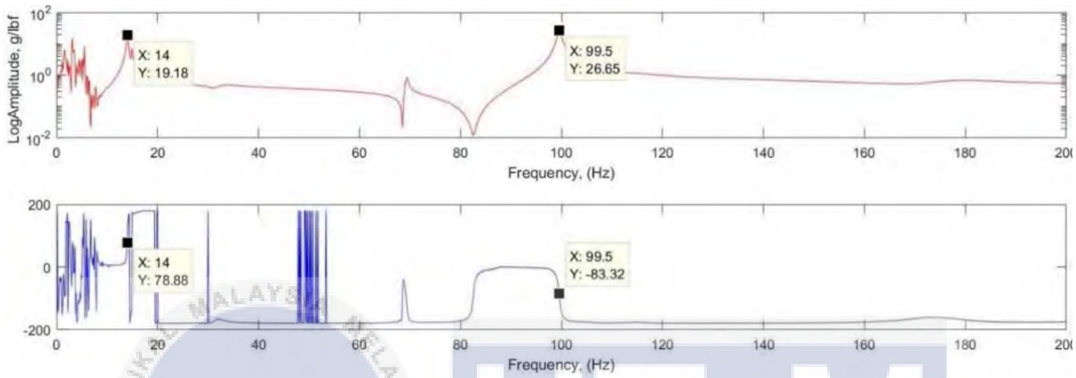
(b)

Figure 4.1: (a) Schematic diagram of unloaded beam (b) Natural frequencies (1st and 2nd) of the unloaded Beam.

Figure 4.2a shows the schematic diagram of beam with 2 DVA clamps of mass 0.408 kg, which is 16% of the total mass of the beam attached to the beam. In Figure 4.2b, the first and second natural frequencies of loaded beam is 14 Hz and 99.5 Hz respectively. The attaching of the two clamps resulted in the reduction of natural frequencies of the beam. Figure 4.2c compares the natural frequencies of unloaded beam and loaded beam, the natural frequencies of loaded beam is lower than the unloaded one. This is due to the application of loads increased the effective mass for the vibratory system of the primary structure.



(a)



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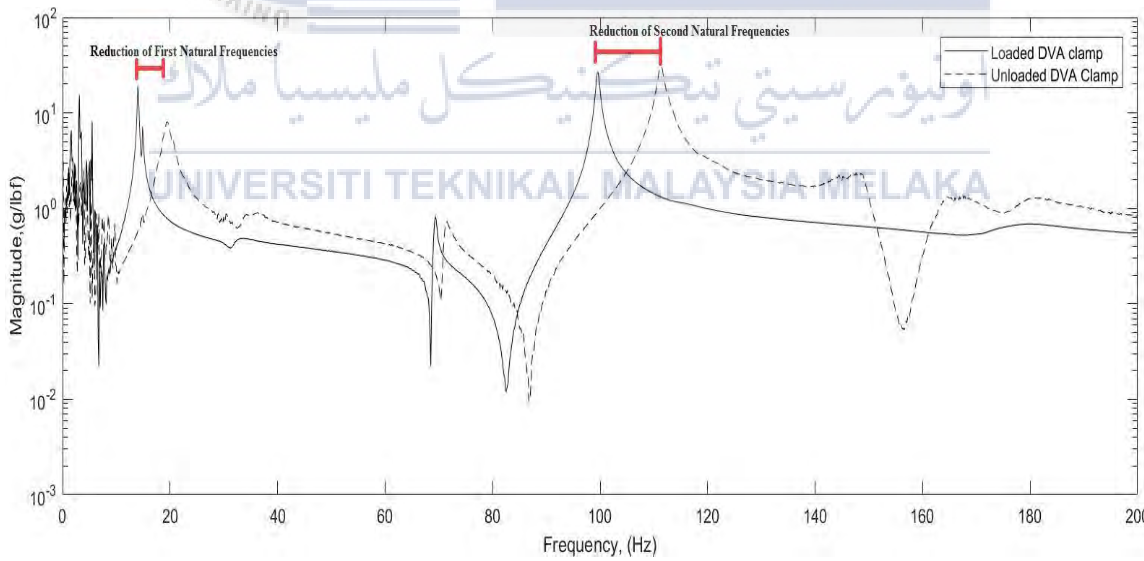


(c)

Figure 4.2: (a) Schematic Diagram of loaded Beam (b) Natural Frequencies (1st and 2nd) of the loaded Beam (c) Comparison of natural frequencies of loaded Beam and unloaded Beam.

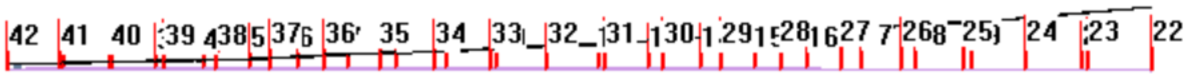
Table 4.1 summarized the percentage difference of natural frequencies of the loaded beam and unloaded beam. The theoretical natural frequencies of unloaded beam and loaded beam is calculated using methods shown in Section 3.1.1 and 3.1.2. The percentage difference for first and second natural frequency of the loaded beam deviates from its theoretical value with percentage difference of 7.3% and 11.1%.

Table 4.1: Percentage Difference of Natural Frequencies

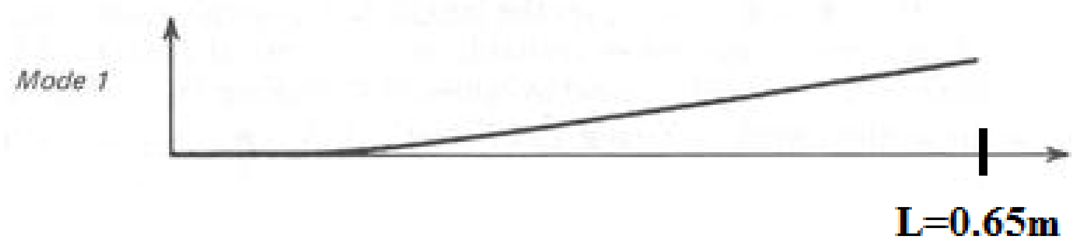
Mode	Theory, Hz	Experiment, Hz	Percentage Difference, %	Theory, Frequency with effective mass, Hz	Frequency of loaded Beam	Percentage Difference, %
First Mode	19.0	19.6	3.2	15.1	14.0	7.3
Second Mode	121.0	111.5	7.9	112.0	99.6	11.1

4.2 Operating Deflection Shape of Beam

Figure 4.3a and Figure 4.3b show the mode shape of the vibration of the beam at first natural frequency and location of point with maximum vibration amplitude of the beam. Based on the result from the experiment, the point of maximum amplitude is located at the distance of *L*, which is 0.65 m from the clamped location.



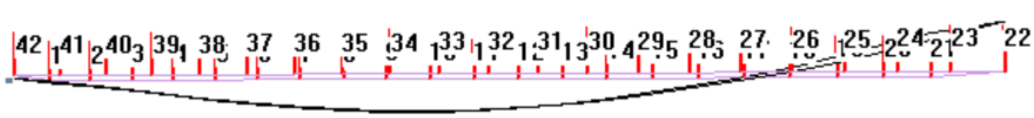
(a)



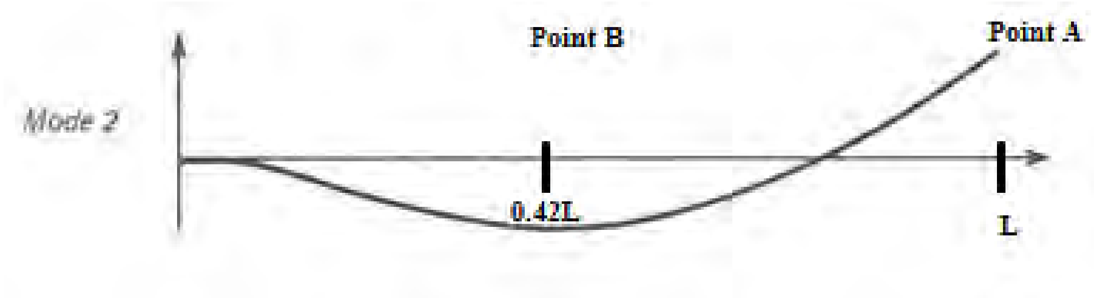
(b)

Figure 4.3: (a) ODS for first mode of vibrating beam (b) Schematic diagram showing the point with maximum amplitude at first natural frequency.

Figure 4.4a and Figure 4.4b show the mode shape of the vibration of the beam at second natural frequency and 2 locations with maximum vibrating amplitude. The first point, Point A is located at the distance of L, which is 0.65 m from the clamped location. The second point, Point B located at the distance of 0.42L from the clamped location, which is 0.27 m from the location where beam is clamped.



(a)

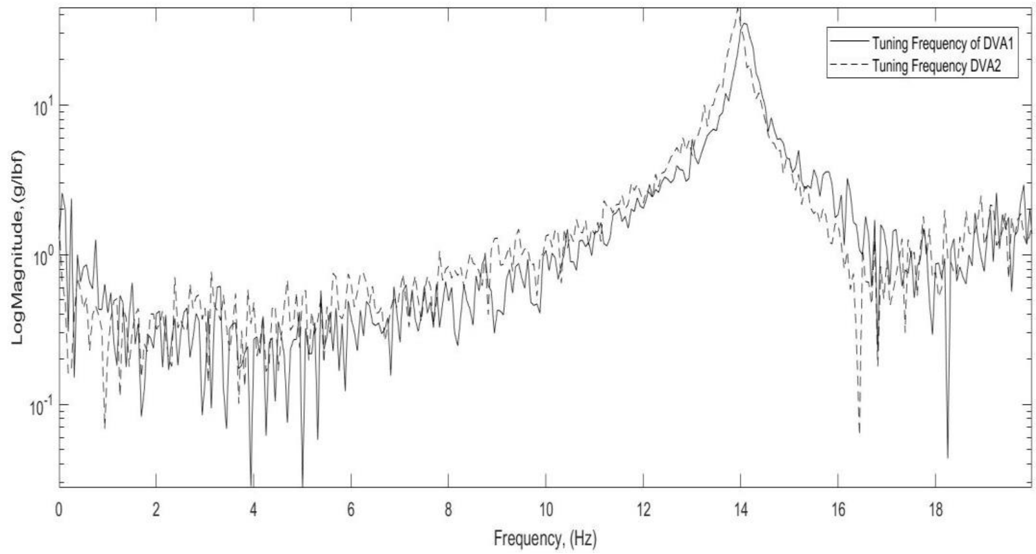


(b)

Figure 4.4: (a) ODS for second mode of vibrating beam (b) Schematic diagram of points with maximum amplitude at second natural frequency.

4.3 Tuning of DVA

Figure 4.5a shows result of frequency tuning of DVA1 and DVA2 respectively. DVA1 and DVA2 is tuned to 13.9 Hz and 14.1 Hz, with a frequency difference of 0.2 Hz. Figure 4.5b shows the result of frequency tuning of DVA3 and DVA4. The frequency of DVA3 and DVA 4 is tuned to 101 Hz and 97.5 Hz respectively. From Figure 4.5, there is presence of noise in the tuning of 4 of the DVAs. This problem arise using impact testing for determining the natural frequency of each DVA. The noise level is lower in DVA3 and DVA4 compared to DVA1 and DVA2 could be due to higher damping ratio.



(a)

(b)

Figure 4.5: (a) Frequency Tuning of DVA1 and DVA2 (b) Frequency Tuning of DVA3 and DVA4.

Table 4.2 summarizes the percentage difference of the theoretical beam tuning length and actual length of the beam to be tuned for each DVA to its desired natural frequency of each. From the table, the percentage difference of each DVA is high. This is due to the improper excitation technique using impact testing to determine the natural frequency of each DVA.

Table 4.2: Percentage Difference of Calculated Tuning Length and Actual Tuning Length.

	At 1st Natural Frequency		At 2nd Natural Frequency	
Vibration Absorber	DVA1	DVA2	DVA3	DVA4
Desired Natural Frequency, Hz	14.1	13.9	101	97.5
Theoretical Beam Tuning Length, m	0.17	0.16	0.064	0.066
Actual Beam Tuning Length	0.14	0.13	0.038	0.039
Percentage Difference,%	18.7	17.6	39.7	40.0

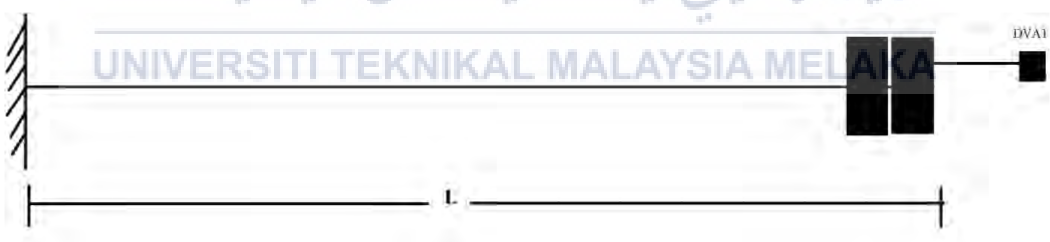
4.4 Performance of DVA at 1st Natural Frequency

To assess the performance of fabricated DVA1 and DVA2 in suppressing the vibration, the frequency response of the beam using single DVA will first be discussed and then to be compared with frequency response of the beam using multiple DVAs. The performance of the single DVA and multiple DVAs will be assessed in terms of bandwidth and also vibration amplitude reduction.

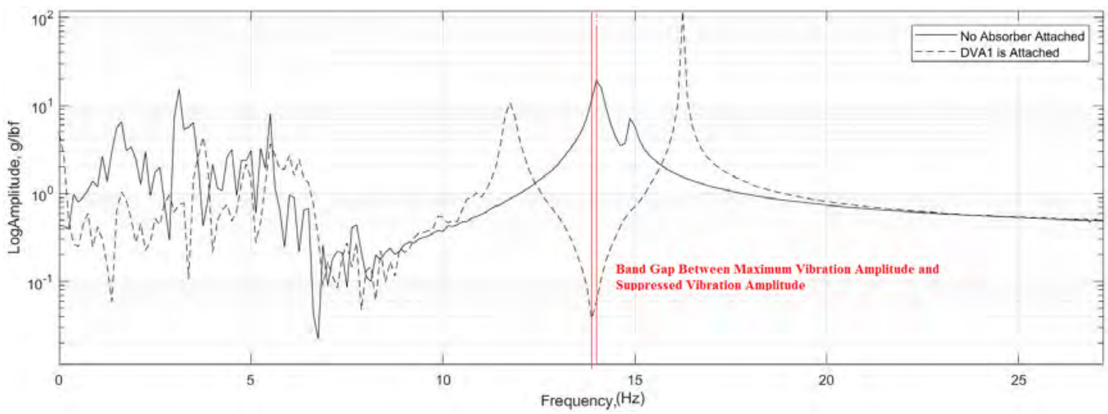
4.4.1 Performance of Single DVA at 1st Natural Frequency

In this section, the performance of single DVA will be discussed. Figure 4.6a shows the schematic diagram of DVA1 attached to the loaded primary beam. The DVA 1 is attached at the distance of L, which is 0.65m, located at the tip of the beam. Figure

4.6b shows the result for the frequency response of primary beam of no absorber attached and with DVA1 is attached at the tip of the primary beam. On the graph, frequency response of beam without attachment of DVA1 is plotted using solid line and dashed line respresent the frequency response of beam with DVA1 attached. There is band gap between maximum vibration amplitude and suppressed vibration amplitude. The fabricated DVA1 is tuned to 14.1 Hz to suppressed the vibration amplitude at 14.0 Hz and the result shows that the vibrating amplitude is suppressed at frequency of 13.9 Hz. This small band gap can be said that arise when transfer the fabricated DVA1 from location of testing to attaching point on primary beam. Although there is mistuning issue in matching the frequency of DVA1 with first natural frequency of the primary structure, but the tuned DVA1 still able to provide counteract force in attenuating the vibration of primary structure, which is shown in Figure4.6c, at the phase difference between the primary structure and the DVA is approximate 90°.



(a)



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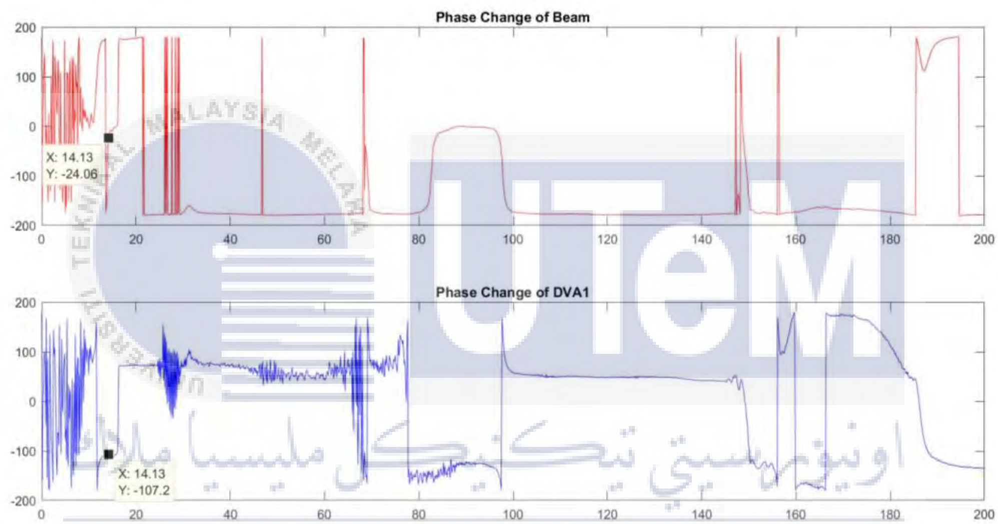


(c)

Figure 4.6: (a) Schematic Diagram of DVA1 attached to beam (b) Frequency Response of Beam with Single DVA attached (c) Phase Changes of Beam and DVA1.

4.4.2 Performance of Multiple DVAs at 1st Frequency

In this section, the performance of multiple DVAs will be discussed. Figure 4.7 shows the schematic diagram of DVA1 and DVA2 attached to the loaded primary beam. The DVA 1 is attached at the distance of L, which is 0.65 m, located at the tip of the beam and DVA2 is attached to the distance of 0.60m from the clamped location.

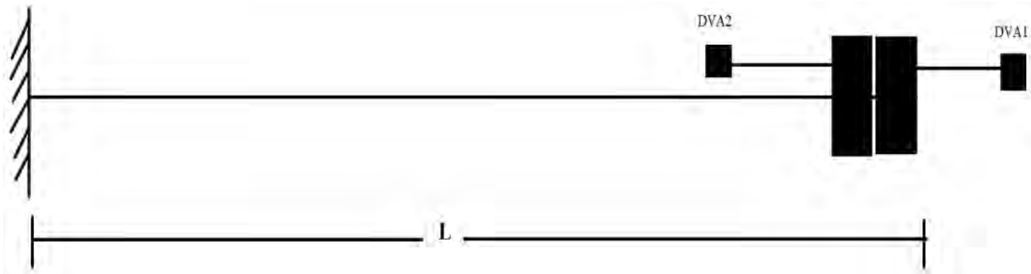


Figure 4.7: Schematic diagram of DVA1 and DVA2 attached to beam.

Figure 4.8 shows the comparison of performance in terms of bandwidth using single DVA and multiple DVAs in vibration amplitude attenuation of beam. For comparison purpose, vibration amplitude is fixed at 0.5517 g/lbf. From the graph, it can be observed that using single DVA, the effective bandwidth for vibration suppression is approximately around 0.8 Hz. As compared to single DVA, the bandwidth of using multiple DVAs in vibration suppression of the beam is 1.30 Hz, which increase the bandwidth by 0.42 Hz. The application of multiple DVAs also increase the frequency bandwidth of the peaks. In application of single DVA, the bandwidth between two peaks is 4.5 Hz. For application of multiple DVAs, the bandwidth between two peaks is 5 Hz, which increase 0.5 Hz. The distance of the peak for single DVA and multiple DVAs is 0.25 Hz. This value is slightly different to the difference frequency of tuned DVA1 and DVA2, which is 0.2 Hz

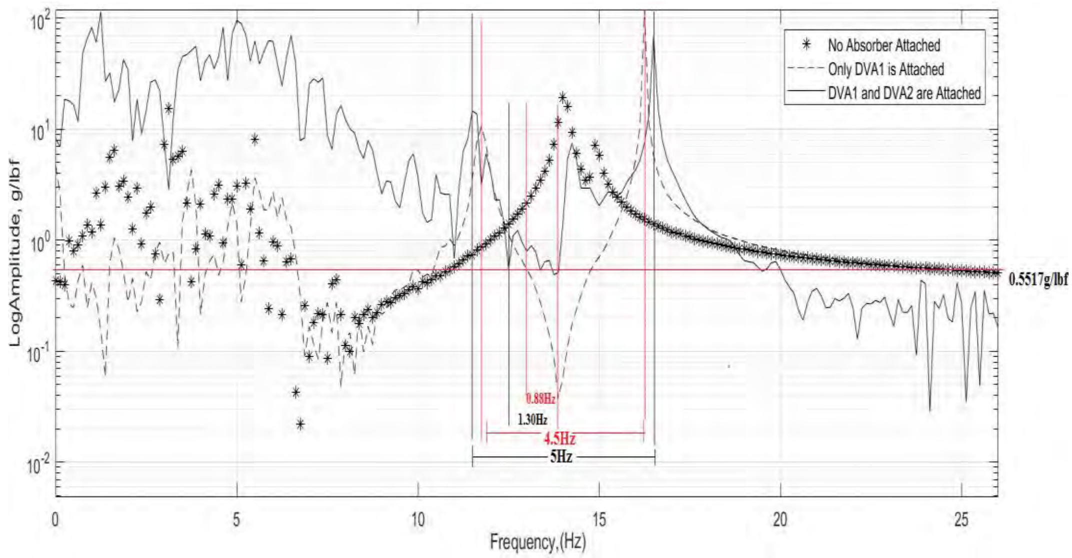


Figure 4.8: Comparison of Performance in terms of bandwidth increase of Single DVA and Multiple DVAs on suppressing the beam vibration amplitude.

From the Figure 4.9 shows comparison of performance in terms of vibration amplitude reduction of Single DVA and Multiple DVAs. At 14 Hz, the unsuppressed vibration amplitude of beam is 19.18 g/lbf and with the application of the single DVAs, the vibration amplitude is reduced to 0.0377 g/lbf, which reduced 99.8%. With the application of multiple DVAs, the vibration amplitude is only reduce to 0.5517 g/lbf at 14 Hz, which reduced 97.4%. In terms of vibration amplitude reduction, there is penalty of using multiple DVAs as compared to single DVA. This could be justified from the increasing damping effect to the system when multiple DVAs is used in vibration amplitude suppression.

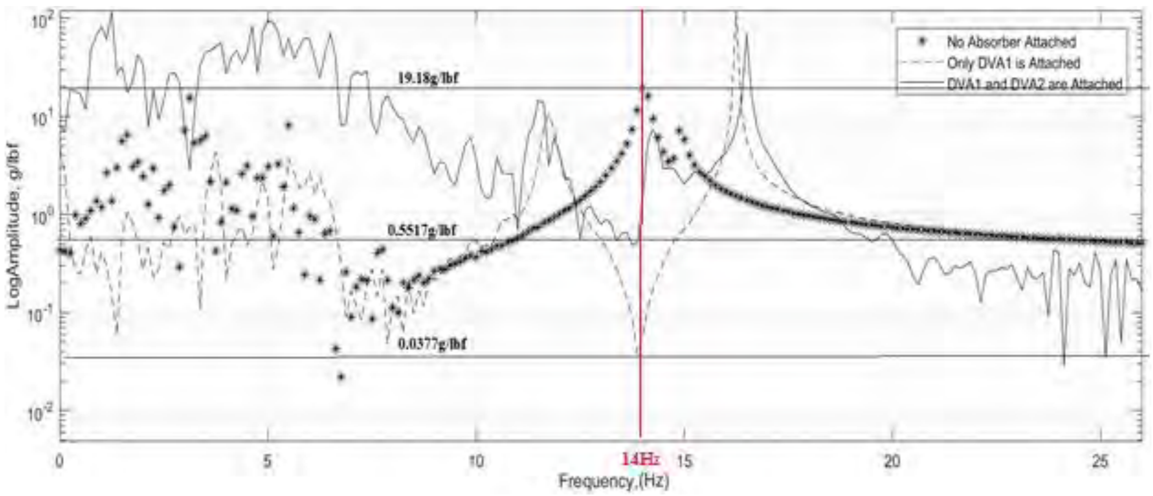


Figure 4.9: Comparison of Performance in terms of vibration amplitude reduction of Single DVA and Multiple DVAs on suppressing the beam vibration amplitude at 14 Hz.

4.5 Performance of DVA at 2nd Natural Frequency

For 2nd natural frequency of beam, the performance of the fabricated DVA3 and DVA4 in vibration attenuation of beam will be assessed by attachment of DVAs to different location points. From Section 4.2, it is identified that at 2nd natural frequency, there is 2 points with maximum vibration amplitude, the point A located at the distance of L, which is 0.65 m from the clamped point and point B located from 0.42L from the clamped point, which is 0.27 m from the clamped point (Refer Figure 4.4b).

4.5.1 Performance of Single DVA at 2nd Natural Frequency

In this section, the performance of single DVA at 2nd natural frequency will be assessed. DVA3 that is tuned to 100.9 Hz will be attached to Point A and Point B where vibration amplitude is maximum. Figure 4.10 shows the difference in vibration amplitude of Point A and Point B. From the result, it can be observed that, at the frequency of 99.6 Hz, the vibration amplitude at Point A is higher than Point B, this is due source of excitation is located at point A.

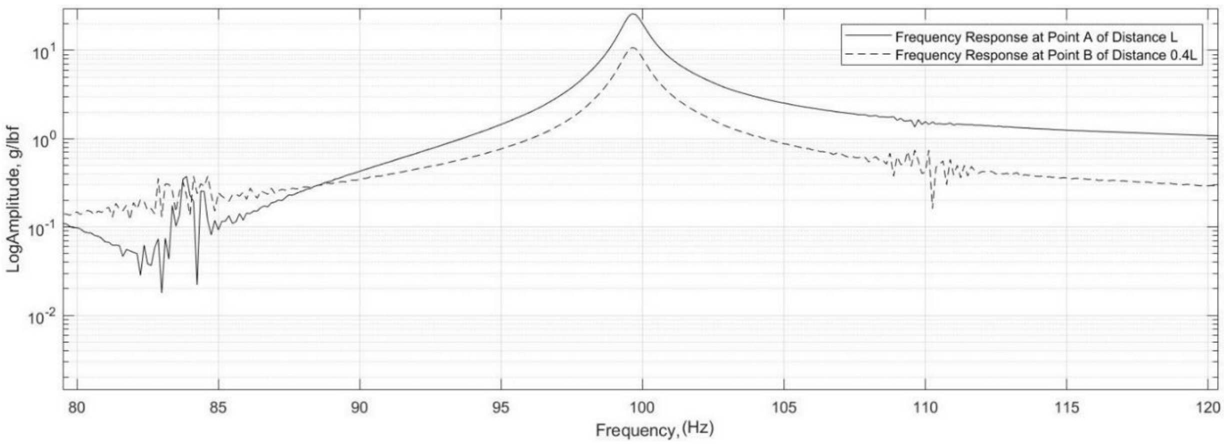
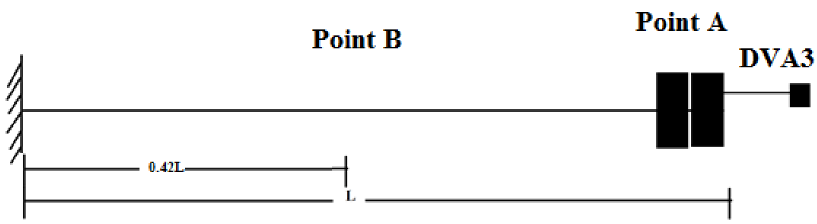


Figure 4.10: Vibration amplitude of Point A and Point B.

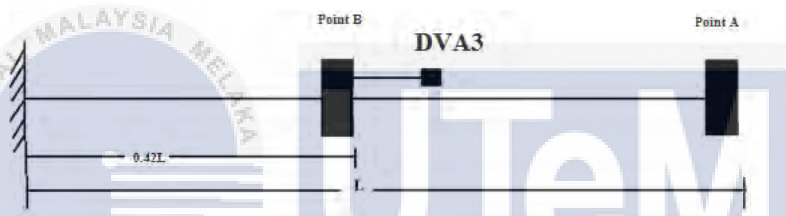
Figure 4.11a shows the attachment of DVA3 at the Point A and Figure 4.10b shows the attachment of DVA3 at Point B. Figure 4.11c shows the beam frequency response at Point A. When DVA3 of tuning frequency 101 Hz is attached at Point A, the vibration is suppressed at the frequency of 103.3 Hz, where there is a band gap of 3.7 Hz. When DVA3 is attached to Point B, the vibration is suppressed at the frequency of 106.4 Hz the band gap increases to 6.8 Hz. As the band gap increase, the performance of DVA3 in suppressing vibration decreases. Performance of vibration suppression can be improve as DVA3 is tuned closer to 99.6 Hz. From the Figure 4.11c, the peaks of DVA3 attached at point A is also higher as compared to point B. which is 20.2 Hz. As shown in Figure 4.11d, the frequency response of beam at Point B shows similarity as in A, except that the amplitude of vibration is lower.

From the Figure 4.11e, it can also be observed where DVA3 is attached at Point A shows higher range of bandwidth, which is 13.4 Hz. When DVA3 is attached at Point

B, the range of the bandwidth reduced to 4.1 Hz. For comparison purpose the amplitude is fixed at 0.1 g/lbf. This shows that attachment of DVA3 in Point A is more effective in terms of frequency bandwidth and vibration amplitude suppression.



(a)



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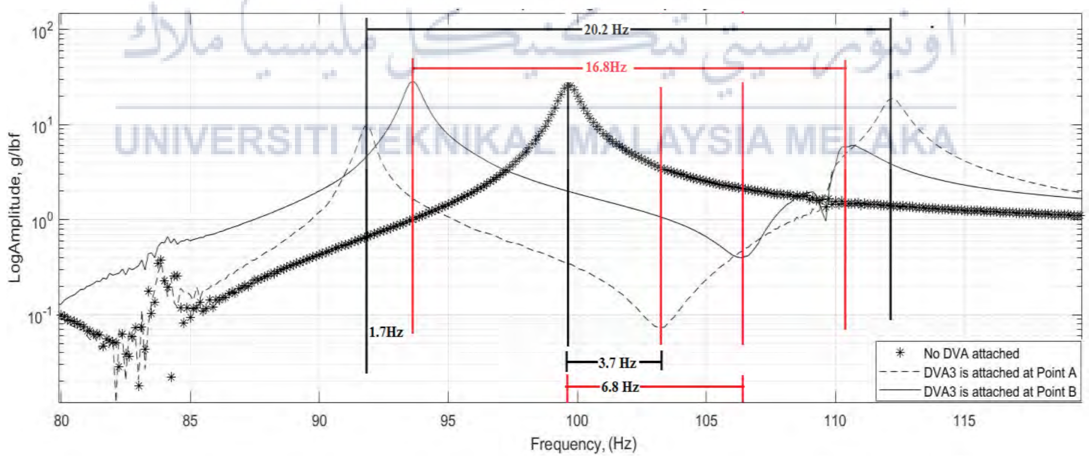

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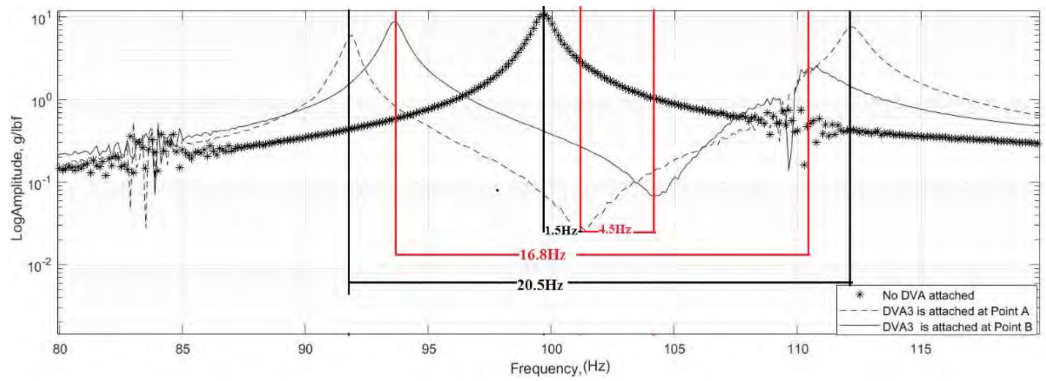

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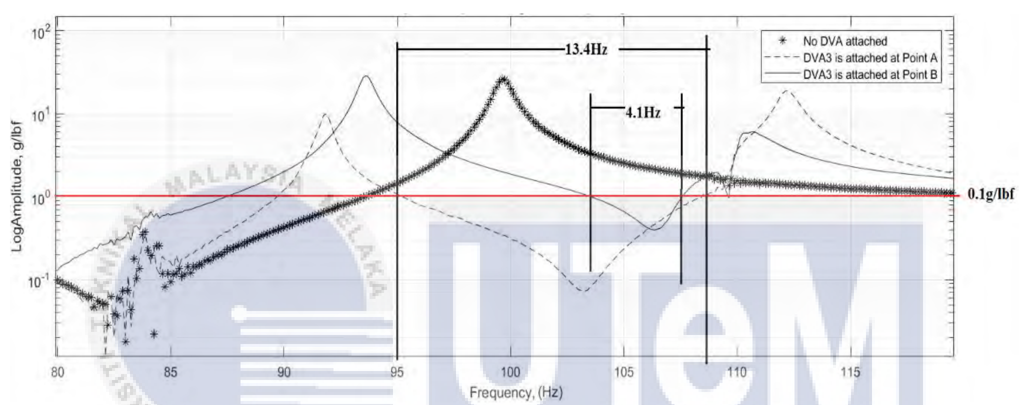

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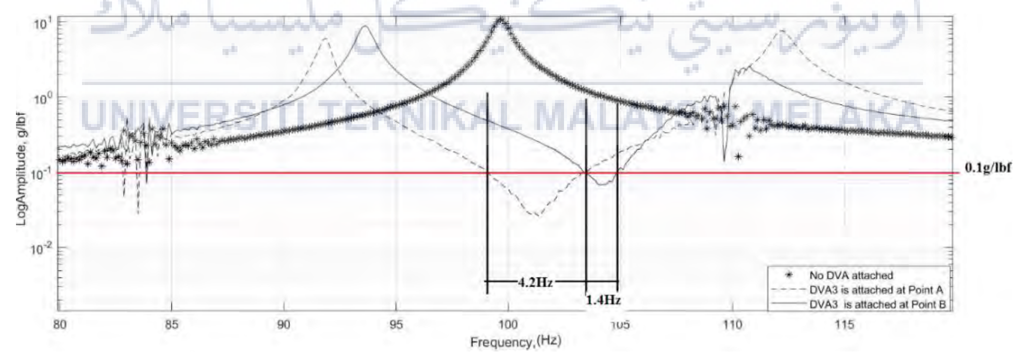


(f)

Figure 4.11: (a) Schematic diagram of DVA3 attached to point A (b) Schematic diagram of DVA3 attached to point B (c) Frequency response of beam at point A with DVA3 attached (d) Frequency response of ream at point B with DVA3 attached (e) Bandwidth increase in Frequency response at point A (f) Bandwidth increase in frequency response at point B.

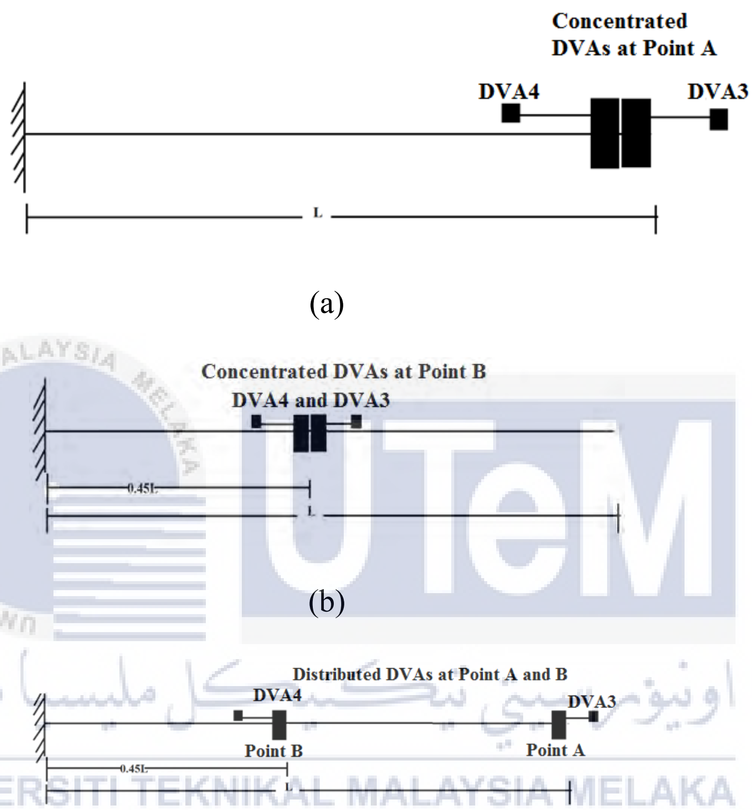
Table 4.3 shows the summarization of percentage of vibration amplitude and range of bandwidth for point A and B with attachment of DVA3. From the table, it can be said that the location of attachment of DVA3 affects both the vibration amplitude reduction percentage and range of bandwidth. As the DVA3 is attached at Point A, the vibration amplitude reduction percentage is higher, where the vibration of amplitude is reduced from 26.1 g/lbf to 0.07 g/lbf, which is approximately, 98.7%. It could be said that the performance is better with single DVA3 attached at point A as point A is vibrating at higher amplitude and closer to the source of excitation.

Table 4.3: Summarization of Percentage of Vibration Amplitude and Range of Bandwidth for Single DVA.

Location of DVA3	Frequency Response at Point A			Frequency Response at Point B		
	Range of Bandwidth Hz	Amplitude, g/lbf	Percentage Reduction, %	Range of Bandwidth , Hz	Amplitude, g/lbf	Percentage Reduction, %
Without DVA	-	26.11	-	-	10.8	-
Point A	13.4	0.34	98.7	4.2	0.07	99.4
Point B	4.1	2.00	92.3	1.4	0.42	96.1

4.5.2 Comparison of Performance of Multiple DVAs at 2nd Natural Frequency

In this section, the performance of multiple DVAs at 2nd natural frequency will be assessed based on different locations of attachment. DVA3 and DVA4 that is tuned to 101 Hz and 97.5Hz will be attached to Point A and Point B where vibration amplitude is maximum. Different point of attaching of multiple DVAs will be considered in this section. Figure 4.12 shows the different methods of attaching multiple DVAs on beam.



(c)

Figure 4.12: (a) Schematic Diagram of concentrated DVAs attached to Point A (b) Schematic Diagram of concentrated DVAs attached to Point B (c) Schematic Diagram of distributed DVAs attached to Point A and DVA4 attached to Point B.

Figure 4.13 shows the frequency response at Point A with different location of attaching of multiple DVAs. From the result, with the attaching of DVA3 and DVA4 at point A, the presence of noise disturb the performance of the DVAs (see Appendix A).

From the figure, there is periodic drop of coherence and noise occur in the spectrum with interval of approximate 26 Hz. This result in the frequencies where the vibration amplitude of the beam is suppressed using multiple DVAs unable to be determined due to presence of noise around 110 Hz. The noise in frequency response is lower when multiple DVAs are attached at Point B.

For the comparison of bandwidth, due to the presence of noise in the frequency response, so the performance in terms of frequency bandwidth will be assessed only based on the range the peaks of each frequency response. From the bandwidth of the peaks, the bandwidth between peaks for distributed DVAs is the highest which is 27.6 Hz, followed by peaks for concentrated DVAs at point A, which is 22.3 Hz and then concentrated DVAs at point B which is 20.2 Hz. It can be said that distributed DVAs is more effective in increasing the bandwidth as compared to concentrated attachment of multi DVAs at point A or point B.

In terms of vibration amplitude, from the Figure4.13, it is unable to determine the frequency for multiple DVAs attached at Point A due to unexpected noise that present during the experiments and the result for vibration reduction for concentrated DVA's at point A is omitted for comparison. In this paper, only the result for multiple DVAs attached to Point B and DVA3 attached to point A and DVA4 at point B is compared.

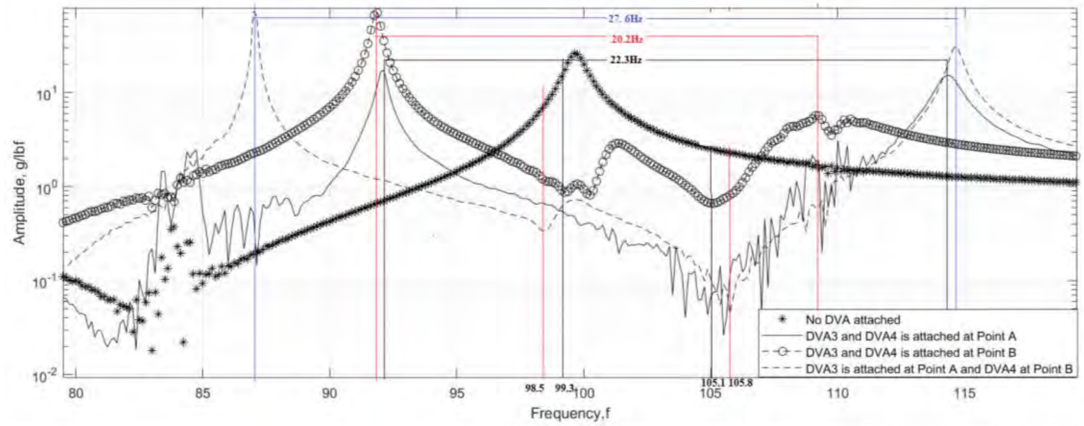


Figure 4.13: Comparison of Frequency Response at Point A with different location of attachment of multiple DVAs.

Figure4.14 shows the frequency response at Point B, the result is similar with frequency response at Point A, only at lower vibration amplitude, which is located further from the source of excitation.

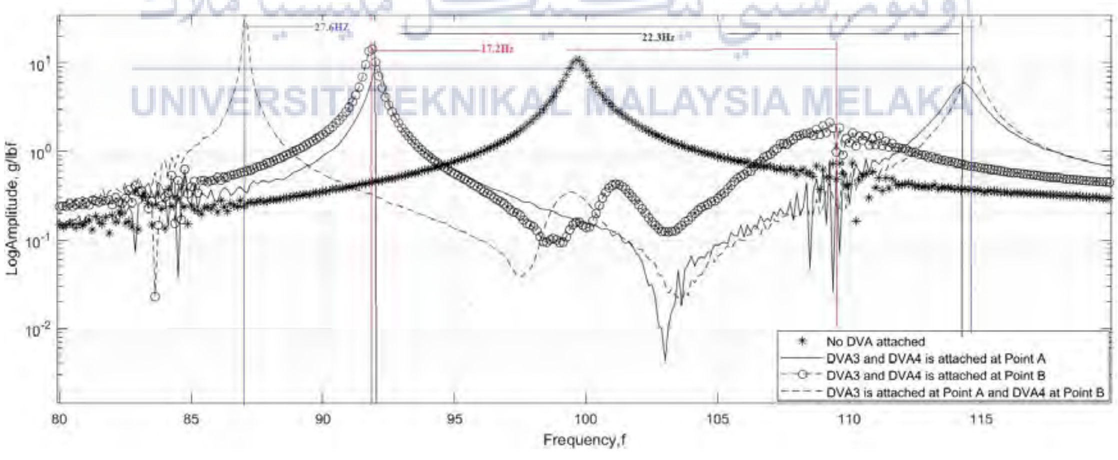


Figure 4.14: Comparison of Frequency Response at Point B with different location of attachment of multiple DVAs.

Table 4.4 summarized the percentage of vibration amplitude and range of bandwidth for Multiple DVAs. From the table, it can be said that the attachment of DVA3 at point A and DVA4 at point B shows higher range of frequency bandwidth. In terms of vibration amplitude percentage reduction, the location of the DVAs attachment shows similar performance in attenuating the vibration amplitude. In other words, the location of DVAs attachment have no effect in vibration suppression of beam.

Table 4.4: Summarization of Percentage of Vibration Amplitude and Range of Bandwidth for Multiple DVAs.

Location	Frequency Response at Point A			Frequency Response at Point B		
	Range of Bandwidth for peaks, Hz	Amplitude, g/lbf	Percentage Reduction, %	Range of Bandwidth	Amplitude, g/lbf	Percentage Reduction, %
No DVA	-	26.11	-	-	10.8	-
Point A	22.3	N/A	N/A	22.3	N/A	N/A
Point B	20.3	1.03	96.1	17.2	0.16	98.5
Distributed DVAs	27.6	0.72	97.2	27.6	0.33	96.9

4.5.3 Comparison for Performance of Single DVA and Multiple DVAs at 2nd Natural Frequency

In the comparison for performance of single DVA and multiple DVAs at 2nd natural frequency, the data obtained using single DVA and multiple DVAs at point A will be plotted in same graph for comparison purpose.

Figure 4.15 shows comparison between single DVA3 attached in Point A and distributed DVA3 in point A and DVA4 in point B. From the figure, the frequency response of the beam with multiple DVAs attached at Point A is omitted for comparison due to noise problem as shown in previous section (See Figure 4.13a). The vibration amplitude will be fixed at 0.35 g/lbf. For using single DVAs, the effective bandwidth is 5.7 Hz. With the application multiple DVAs, the effective bandwidth increase to 7.3 Hz. The peaks of distributed DVAs is also shows greater frequency range compared to concentrated DVAs. From the graph, it can be observed that using multiple DVAs, the vibration is suppressed at two frequency and with single DVA, the vibration is only suppressed at one frequency.

In terms of amplitude reduction, at frequency of 99.6 Hz, the vibration amplitude is reduce to 0.72 g/lbf with multiple DVAs and 0.35 g/lbf with single DVAs.

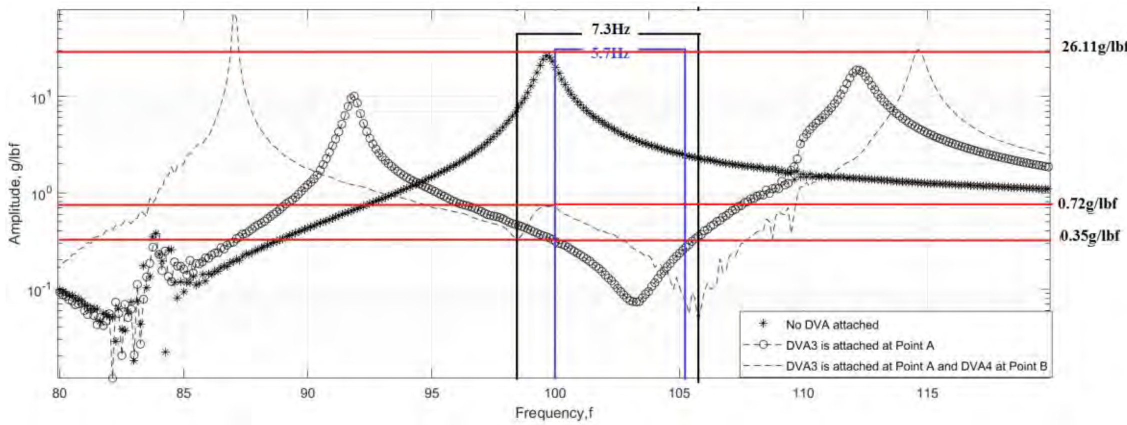


Figure 4.15: Comparison of Single DVA and Multiple DVAs at Point A.

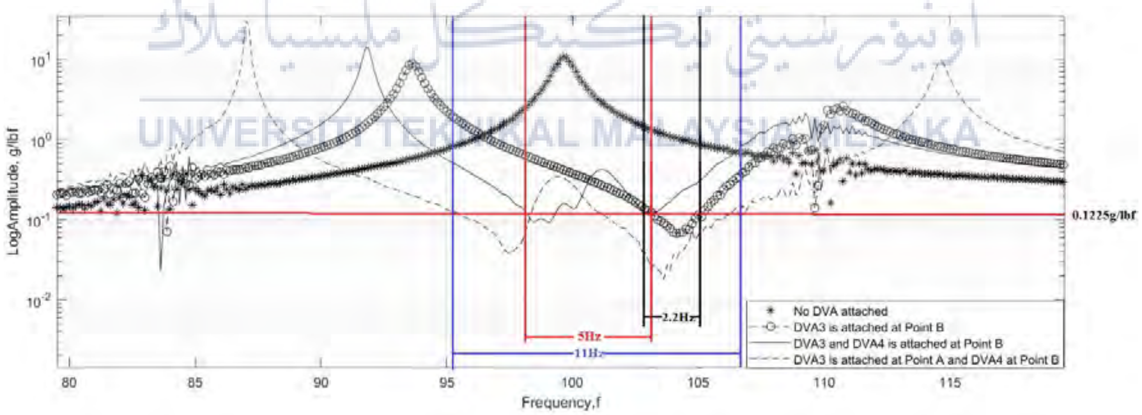
Table 4.5 shows comparison of performance of single DVAs and multiple DVAs at Point A. From the tables, it shows that distributed DVAs shows greater range of bandwidth in suppressing the vibration of beam. The Single DVA results in greater vibration amplitude reduction percentage.

Table 4.5: Comparison of performance of Single DVAs and multiple DVAs at Point A.

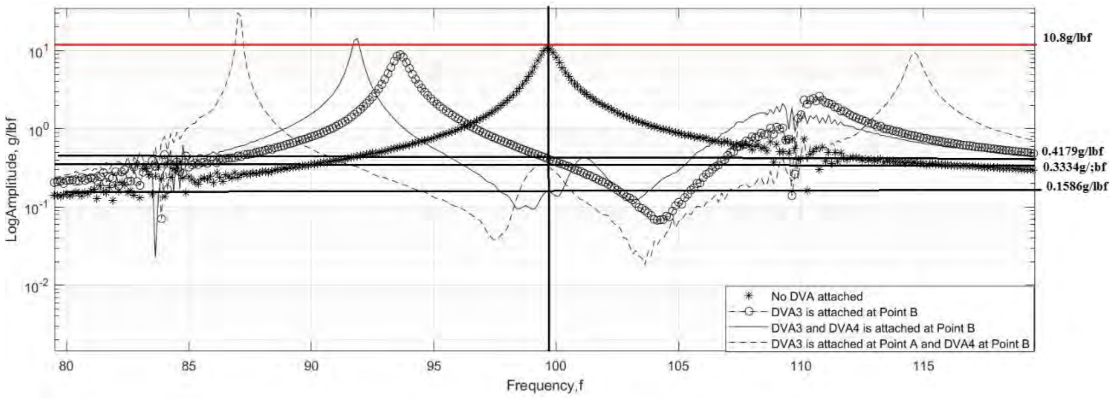
Location	Frequency Response at Point A		
	Range of Bandwidth, Hz	Amplitude, g/lbf	Percentage Reduction, %
No DVA	-	26.11	-
Single DVA at A	5.7	0.35	98.7
Multiple DVAs at A	N/A	N/A	N/A
Distributed DVAs A	7.3	0.72	97.2

Figure 4.16a shows the comparison of single DVA and multiples DVAs with the frequency response of Point B. For comparison purpose, the vibration amplitude is fixed at 0.1225 g/lbf. From the results, the frequency response with multi DVAs that is attached at Point A and Point B, the range of the bandwidth is 11 Hz, which is higher as compared to the bandwidth of multiple DVAs attached at point B that is 5 Hz. The bandwidth range of using single DVA is lowest, which is 2.2 Hz only.

Figure 4.16b shows result for comparison of vibration amplitude reduction for Single DVA and multiples DVAs at Point B. At the frequency of 99.6 Hz, the greatest vibration amplitude reduction is achieved by multiple DVAs attached to Point B, which reduced by 98.5%. The percentage reduction of vibration amplitude using distributed multiple DVAs at point A and point B is 96.9%. The percentage reduction of vibration amplitude using single DVAs is 96.1%.



(a)



(b)

Figure 4.16: (a) Comparison of Frequency Bandwidth for Single DVA and multiples DVAs at Point B (b) Comparison of vibration amplitude reduction for Single DVA and multiples DVAs at Point B.

Table 4.6 shows comparison of performance of single DVAs and multiple DVAs at Point B. From the tables, it shows that distributed DVAs shows greatest range of bandwidth in suppressing the vibration of beam. The concentrated multiple DVAs results in greatest vibration amplitude reduction percentage.

Table 4.6: Comparison of performance of Single DVAs and multiple DVAs at Point B.

Location	Frequency Response at Point B		
	Range of Bandwidth. Hz	Amplitude, g/lbf	Percentage Reduction, %
Without DVA	-	10.8	-
Single DVA at B	2.2	0.42	96.1
Multiple DVAs at B	5	0.16	98.5
Distributed DVAs B	11	0.33	96.9

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

In this research, damped DVA is fabricated and used to suppress the vibration of fixed-free beam at first and second natural frequencies. Based on the result, there are different alternatives in suppressing the vibration with different configuration in second natural frequency.

In designing and characterization of DVA, based on the classical equation of stiffness and natural frequency, cantilever beam absorber is fabricated and tested using impact testing. DVA clamps that is fabricated using lightweight material is designed. With the designed DVA clamp, the vibration absorber is mounted to the beam to suppress the vibration of the beam.

Based on the result from the experiment, although it can be said that multiple DVAs able to increase the frequency bandwidth as compared to single DVA, especially in first natural frequency, the result is not obvious as natural frequencies of multiple DVAs is tuned closed to each other. In second natural frequency, due to external disturbances from the surrounding condition affected the performance of fabricated DVA in vibration suppression of beam. It shows the sign of increased frequency bandwidth especially using the distributed DVAs.

Overall, this project is achieving its objective which involves in comparing the performance of single DVA and multiple DVAs. But improvement can be done especially in reducing the noise which affect the performance of DVA.

5.2 Recommendation

There are several improvement recommendation to be taken for improvement in performance assessment and a better result of analysis. These recommendations are:

1. The DVA clamp for mounting the cantilever beam absorber should be designed in smaller size to reduce it mass loading effect on the natural frequencies of the beam.
2. A better DVA in new sophisticated design that meets the requirement should be created. The fabrication also should be precise to improve the obtained results.
3. The number of DVA should be increase and its frequencies should be tuned at greater differences so that the result of using multiple DVAs is more obvious.
4. Shaker excitation should be adopted as the methods for more precise tuning of frequencies of fabricated DVA. Using shaker excitation allows better control of vibration amplitude and signal. Besides, the noise of using shaker is lower compared to using impact hammer.
5. If possible, the grounding should be improvise, to provide a based or foundation which is sufficiently rigid.
6. Trial and error method should be conducted experimentally using different stiffness of stinger to ensure that the occurrence of resonance did not contaminated the results obtained.
7. DVA should be tuned to 1st and 2nd natural frequency simultaneously and mount to DVA for measuring the frequency response of the beam.

REFERENCES

Berardengo, M., Cigada, A., Guanziroli, F., & Manzoni, S. (2015). Modelling and control of an adaptive tuned mass damper based on shape memory alloys and eddy currents. *Journal of Sound and Vibration, 349*, 18-38. doi:10.1016/j.jsv.2015.03.036

Bonello, P., Brennan, M. J., & Elliott, S. J. (2005). Vibration control using an adaptive tuned vibration absorber with a variable curvature stiffness element. *Smart Materials and Structures, 14*(5), 1055-1065.doi:10.1088/0964-1726/14/5/044

Bonsel, J. H., Fey, R. H., & Nijmeijer, H. (2004). Application of a Dynamic Vibration Absorber to a Piecewise Linear Beam System. *Nonlinear Dynamics, 37*(3), 227-243. doi:10.1023/b:nody.0000044646.70030.31

Dan, M. (2016). Marching Soldiers Cause Suspension Bridge Collapse. [online]
Available at: https://www.historyandheadlines.com/april-12-1831-marching-soldiers-cause-suspension-bridge-collapse/ [Accessed on 25 October 2017].

Ellison, J. (2015). Tacoma' Galloping Gertie' Bridge Collapse. [online]
Available at: http://www.seattlepi.com/science/slideshow/Tacoma-Galloping-Gertie-bridge-collapse-120055.php [Accessed on 25 October 2017].

Faal, R. T., Amiri, M. B., Pirmohammadi, A. A., & Milani, A. S. (2011). Vibration analysis of undamped, suspended multi-beam absorber systems. *Meccanica, 47*(5), 1059-1078. doi:10.1007/s11012-011-9493-2

Frahm, H. (1909) Decive for damping vibration of bodies. *US Patent No 989958.*
Gatti, G. (2016). On the undamped vibration absorber with cubic stiffness characteristics. *Journal of Physics: Conference Series, 744*, 012225. doi:10.1088/1742-6596/744/1/012225

Grappasonni, C., Habib, G., Detroux, T., FengWen, W., Kerschen, G., Jensen, J. (2014). Practical design of a nonlinear tuned vibration absorber. *In Proceeding of the ISMA 2014.*

Gsell, D., Feltrin, G., & Motavalli, M. (2007). Adaptive Tuned Mass Damper based on Pre-stressable Leaf-springs. *Journal of Intelligent Material Systems and Structures, 18*(8), 845-851. doi:10.1177/1045389x06073641

Herold, S., & Mayer, D. (2016). Adaptive Piezoelectric Absorber for Active Vibration Control. *Actuators, 5*(1), 7. doi:10.3390/act5010007

Hobeck, J. D., & Inman, D. J. (2015). Magnetoelastic metastructures for passive broadband vibration suppression. *Active and Passive Smart Structures and Integrated Systems 2015*. doi:10.1117/12.2083887

Howard, C. (2009). Review of adaptive tuned vibration absorber. *In Proceeding of Acoustics 2009.*

Jalili, N., & Esmailzadeh, E. (2002). Adaptive-passive structural vibration attenuation using distributed absorbers. *Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics, 216*(3), 223-235. doi:10.1177/146441930221600303

Kaushik. (2014). The 728- Ton Tuned Mass Damper of Taipei 101. [online]
Available at: http://www.amusingplanet.com/2014/08/the-728-ton-tuned-mass-damper-of-taipei.html [Accessed on 25 October 2017].

Keye, S., Keimer, R., & Homann, S. (2009). A vibration absorber with variable eigenfrequency for turboprop aircraft. *Aerospace Science and Technology, 13*(4-5), 165-171. doi:10.1016/j.ast.2008.10.001

McManus, D. (2015). Wobbly Bridge London: Architecture. [online]
Available at: https://www.e-architect.co.uk/london/wobbly-bridge. [Accessed on 25 October 2018].

Nishimura, H., Yoshida, K. & Shimogo, T. (1988). Active dynamic vibration absorber for seismic response control, Proceedings of 9th World Conference on Earthquake Engineering, Vol 8, p.477.

Qiu, D., Seguy, S., & Paredes, M. (2017). Tuned Nonlinear Energy Sink With Conical Spring: Design Theory and Sensitivity Analysis. *Journal of Mechanical Design, 140*(1), 011404. doi:10.1115/1.4038304

Rao, S. S. (2011). *Mechanical vibrations*. Upper Saddle River, NJ: Prentice Hall.

Rustighi, E., Brennan, M. J., & Mace, B. R. (2004). A shape memory alloy adaptive tuned vibration absorber:a design and implementation. *Smart Materials and Structures, 14*(1), 19-28. doi:10.1088/0964-1726/14/1/002

Shui, X., & Wang, S. (2018). Investigation on a mechanical vibration absorber with tunable piecewise-linear stiffness. *Mechanical Systems and Signal Processing, 100*, 330-343. doi:10.1016/j.ymssp.2017.05.046

Vyas, A., & Bajaj, A. (2001). Dynamics of Autoparametric Vibration Absorbers Using Multiple Pendulums. *Journal of Sound and Vibration, 246*(1), 115-135. doi:10.1006/jsvi.2001.3616

Wang, T., Sheng, M., & Qin, Q. (2016). Multi-flexural band gaps in an Euler–Bernoulli beam with lateral local resonators. *Physics Letters A, 380*(4), 525-529. doi:10.1016/j.physleta.2015.12.010

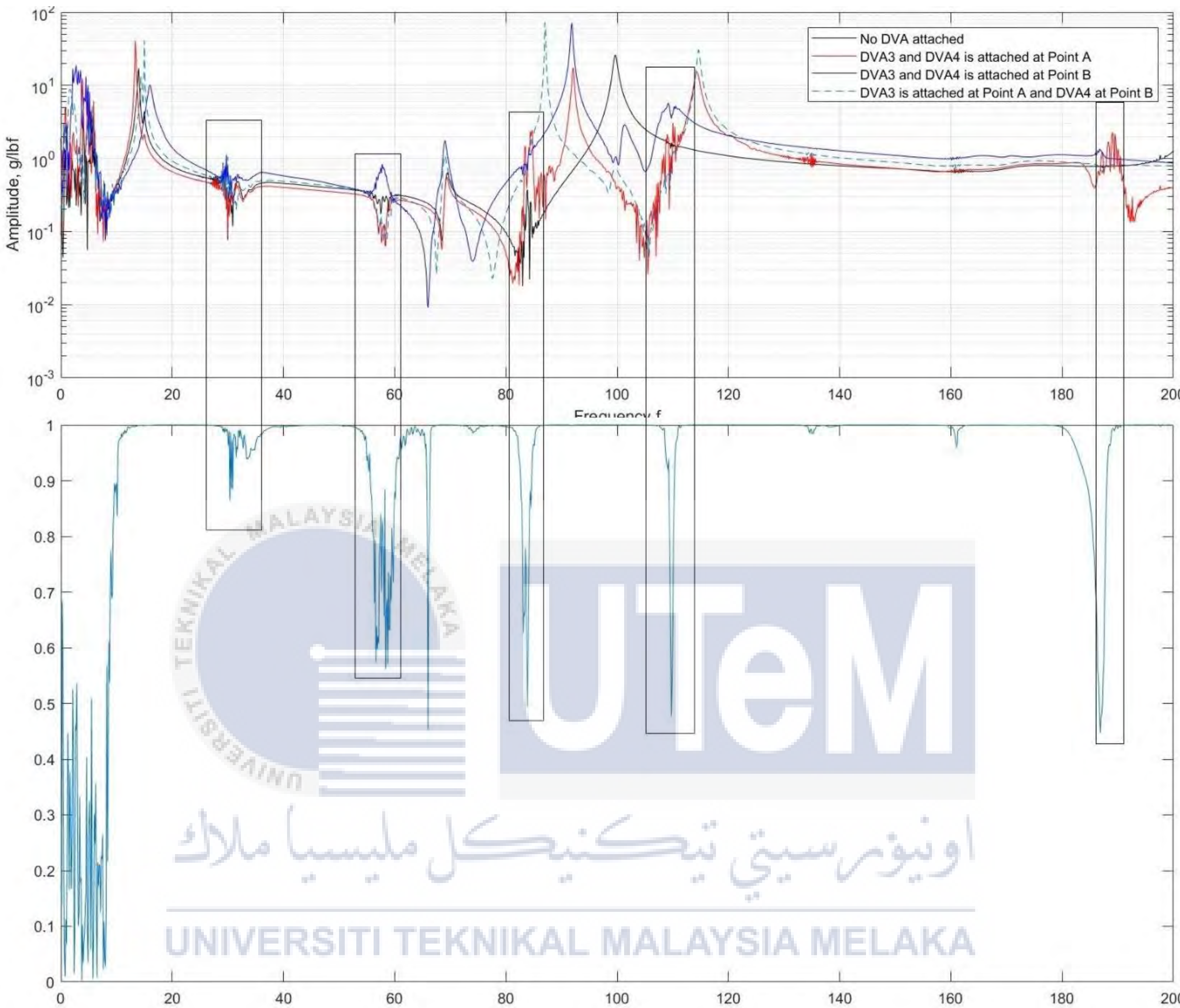
Wong, W., Tang, S., Cheung, Y., & Cheng, L. (2007). Design of a dynamic vibration absorber for vibration isolation of beams under point or distributed loading. *Journal of Sound and Vibration, 301*(3-5), 898-908. doi:10.1016/j.jsv.2006.10.028

Xiao, Y., Wen, J., & Wen, X. (2012). Broadband locally resonant beams containing multiple periodic arrays of attached resonators. *Physics Letters A, 376*(16), 1384-1390. doi:10.1016/j.physleta.2012.02.059

Xiao, Y., Wen, J., Yu, D., & Wen, X. (2013). Flexural wave propagation in beams with periodically attached vibration absorbers: Band-gap behavior and band formation mechanisms. *Journal of Sound and Vibration, 332*(4), 867-893. doi:10.1016/j.jsv.2012.09.035

Zihao, L., Wanyou, L., & Yali, Y. (2014). A study of a beam-like electromagnetic vibration absorber. *Journal of Vibration and Control, 22*(11), 2559-2568. doi:10.1177/1077546314547730

APPENDICES



Appendix A: Presence of Noise at the interval of 26Hz in Experiment