CHARACTERIZATION OF A WIDEBAND NONLINEAR VIBRATION ABSORBER USING CURVED CONSTRAINED CANTILEVER BEAM

MUHAMMAD FAEZ BIN ABDUL GHAPAR

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Faculty of Mechanical Engineering

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

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DECLARATION

“I hereby declare that the work in this thesis is my own except for summaries and quotations which have been duly acknowledged.”

Signature : ....

Author : MUHAMMAD FAEZ BIN ABDUL GHAPAR

Date : ....

SUPERVISOR 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 the degree of Bachelor of Mechanical Engineering (Plant & Maintenance) with honours"

Signature : ....

Supervisor : DR. ROSZAIDI BIN RAMLAN

Date : ....

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ABSTRACT

In many industrial structure or appliances and also other structure such as a building, are involved in having an unwanted vibration such as from earthquake and hard storm. This unwanted vibration will cause a severity to the structure when the structure is excited to its natural frequency. In order to counter with the problem a device called vibration absorber is introduced. The vibration absorber is installed on the structure will suppress the unwanted vibration that occur. There is two type of absorber which is the linear absorber and the nonlinear absorber. The nonlinear absorber study is conducted and from the previous study it proves that the nonlinear absorber have a better performances than the linear absorber which is having wider safe operating frequency range. The study conducted is the characterization of a wideband nonlinear dynamic vibration absorber using curved constrained cantilever beam. The hardening method is used in order to have a nonlinear behaviour of the vibration absorber. The hardening method is by the application of profile curved block with quadratic profile of x^3, x^4, and x^6 on the designed vibration absorber. The designed absorber consist of a cantilever beam that is lies between two curved block and having a mass at the free end of the beam. From the theoretical and analytical study, the curve will constrain the deflection of the absorber beam thus hardened the absorber beam stiffness and make the absorber behave nonlinearly. However in the characterization conducted the profile block curved used on the designed vibration absorber is not stiffen up the deflection of the absorber beam thus the absorber behave linearly. The modification of the nonlinear absorber is taken place by attaching a stopper on the curved block profile to assist in constraining the deflection of the absorber beam. There will only the suppression of the first mode of vibration on the primary structure will be conducted which is modelled by a beam. The first mode of vibration of the primary structure beam is determined to find its natural frequency. The designed nonlinear absorber is tuned to have the same natural

frequency with the primary structure in order to suppress the vibration at its first mode of vibration thus the characterization of the absorber done. From the study conducted, it is proved that the designed nonlinear absorber by using the curved block with some modification is able to suppress the vibration on the first mode of vibration of the primary structure and providing a wider operating frequency range on the primary structure.

ABSTRAK

Banyak struktur industri atau peralatan dan juga struktur lain seperti bangunan, terlibat dengan getaran yang tidak diingini seperti gempa bumi dan ribut. Getaran yang tidak diingini ini akan menyebabkan bencana pada struktur apabila struktur mengalami gegaran pada frekuensi tabii. Untuk mengatasi masalah tersebut peranti dipanggil penyerap getaran diperkenalkan. Penyerap getaran dipasang pada struktur akan menyerap getaran yang tidak diingini. Terdapat dua jenis penyerap getaran iaitu penyerap getaran linear dan penyerap getaran tak linear. Kajian penyerap getaran tak linear dijalankan dan daripada kajian sebelumnya, ia membuktikan bahawa penyerap getaran tak linear mempunyai prestasi yang lebih baik daripada penyerap getaran linear yang mempunyai julat frekuensi operasi selamat yang lebih luas. Kajian yang dijalankan adalah mengenai pencirian sifat-sifat tak linear bagi penyerap getaran dinamik menggunakan rasuk julur yang dikekang oleh blok melengkung. Penyerap getaran yang direka terdiri daripada rasuk yang dikekang antara dua blok lengkung dan mempunyai sebuah jisim di hujung bebas rasuk tersebut. Kaedah pengerasan digunakan untuk menghasilkan tingkah laku tak linear penyerap getaran. Kaedah pengerasan dilakukan dengan aplikasi blok lengkung dengan profil kuadratik x^3, x^4, dan x^6 pada penyerap getaran yang direka. Kajian secara teori dan analisis, rasuk akan dikekang oleh blok lekuk lalu ia akan mengeraskan rasuk penyerap getaran dan membuatkan penyerap getaran bersifat tidak linear. Walau bagaimanapun dalam pencirian yang dijalankan blok profil melengkung digunakan pada penyerap getaran tidak berfungsi seperti yang sepatutnya. Pengubahsuaian penyerap getaran tak linear dilakukan dengan menambah penahan padai profil lengkung blok bagi membantu mengekan pesongan rasuk penyerap getaran. Hanya penyerapan mod pertama getaran pada struktur utama akan dijalankan yang dimodelkan oleh rasuk. Mod pertama getaran struktur rasuk

utama ini telah dipilih untuk mencari frekuensi semula tabii. Penyerap tak linear direka untuk mempunyai frekuensi semula jadi yang sama dengan struktur utama untuk menyerap getaran pada mod pertama.. Daripada kajian yang dijalankan, ia membuktikan bahawa penyerap getaran tak linear yang direka dengan menggunakan lengkung blok dengan beberapa pengubahsuaian boleh mengurangkan getaran pada mod pertama getaran pada struktur utama dan menyediakan julat frekuensi operasi yang selamat yang lebih luas ke atas struktur utama.

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LIST OF ABBREVATIONS

DVA	Dynamic Vibration Absorber
NLDVA	Nonlinear Dynamic Vibration Absorber
FRF	Frequency Response Function
ODS	Operational Deflection Shape

LIST OF SYMBOL

m	Mass
m_a	Absorber mass
k	Stiffness
c_a	Absorber Damping Coefficient
ζ	Damping Ratio
x_a	Absorber Displacement
x	Displacement
$\dot{x}$	First derivatives of x
$\ddot{x}$	Second Derivatives of x
f_t	Harmonic Force
f^*	Tuning Frequency
ζ^*	Optimal Damping Ratio
μ	Mass Ratio
E	Modulus of Elasticity
I	Moment of inertia
L	Beam Length

CHAPTER 1

INTRODUCTION

1.0 BACKGROUND

Vibration is a non-desired phenomenon because of it can caused a disturbance, damage and sometimes might resulting destruction in machinery and structure as well. This unwanted phenomenon must be controlled or should be eliminated from the system of machinery or structure. The widely application used in reducing or eliminating the disturbance vibration occur in a system is by implementing the vibration absorber which is also known as dynamic vibration absorber (Liu & Liu, 2005). The dynamic vibration absorber (DVA) is a vibration control device consisting mass and stiffness which is as an auxilary system that is attached on a primary structure that needs to be control and protected from an unwanted vibration. The DVA can be devided into undamped DVA which consist of mass-spring system and damped DVA which consist of mass-spring-damper introduced by Ormondroyd and Den Hartog (Bekdaş & Nigdeli, 2013).

Besides that, the case of the optimum DVA are also the important strategies in vibration control. The optimization of a DVA icludes several parameters that must be consider such as mass, stiffness and damping (Bekdaş & Nigdeli, 2011). Various of study have been conducted by past and recent researchers in order to optimize the design parameter to improve the performance of the vibration absorber. By implementing the optimum tuning parameters of the absorber, it will provide larger suppression of resonant vibration amplitude of the primary system and also will generating wider safe operating frequencies range of the primary system.

Although the absorber mention above which is categorized as a linear absorber is a famous and familiar device in mitigating unwanted vibration in mechanical structure, its only effective when it is precisely tuned to the frequency of a vibration mode (Viguié & Kerschen, 2009). Furthermore, the first study of non-linear absorber by Roberson, Pipes and Arnold have attract an attention in many literature and after realizing the limitations of linear absorbers ,the non-linear vibration absorber were developed for their performance ability to widen the suppression bandwith (Viguié & Kerschen, 2010).

In this research, a cantilever beam is used as an absorber in the characterization of a wide-band non-linear vibration absorber and the stiffness of the cantilever beam is to be consider in order of the absorber optimization. The hardening of the beam stiffness will be associated by a curve metal in which will constrain the deflection of the absorber when vibration was excited. The absorber will lies against the solid curve, thus shortening its free length and becoming stiffer. Hence, its force characteristic becomes steeper for increasing deflection. Then, the performance of the absorber when put on the vibrating structure will be studied as well as the bandwith of the response.

1.1 PROBLEM STATEMENT

Undesired vibration in practical life can be found in many mechanical structure and machinery that can cause destruction of a structure. The implementation of vibration absorber when perfectly tuned will mitigate the undesired vibration in the primary system due to the supression of the vibration amplitude and thus providing a safe operating frequency range. Due to the primary mass which is at it initial state was a single degree of freedom system is attached with an auxilary single degree of freedom system (absorber) make the system constitute a two degree of freedom system and will have two natural frequencies. Due to the vibration absorber is tuned to only one particular frequency, it is only effective only over a narrow band of frequencies or safe operating frequency range as shown in Figure 1. Hence, this research will conduct an experimental study on the performance of the non-linear vibration absorber which is based on related work of research claimed that the non-linear absorber have a better performance than the linear absorber. Knowledge of the nonlinear characteristics of a vibration absorber is important if its performance is to be predicted accurately when connected to a host structure. This can be achieved theoretically, but experimental validation is necessary to verify the modelling procedure and assumptions. This research describes the characterization of such an absorber using an. experimental procedure

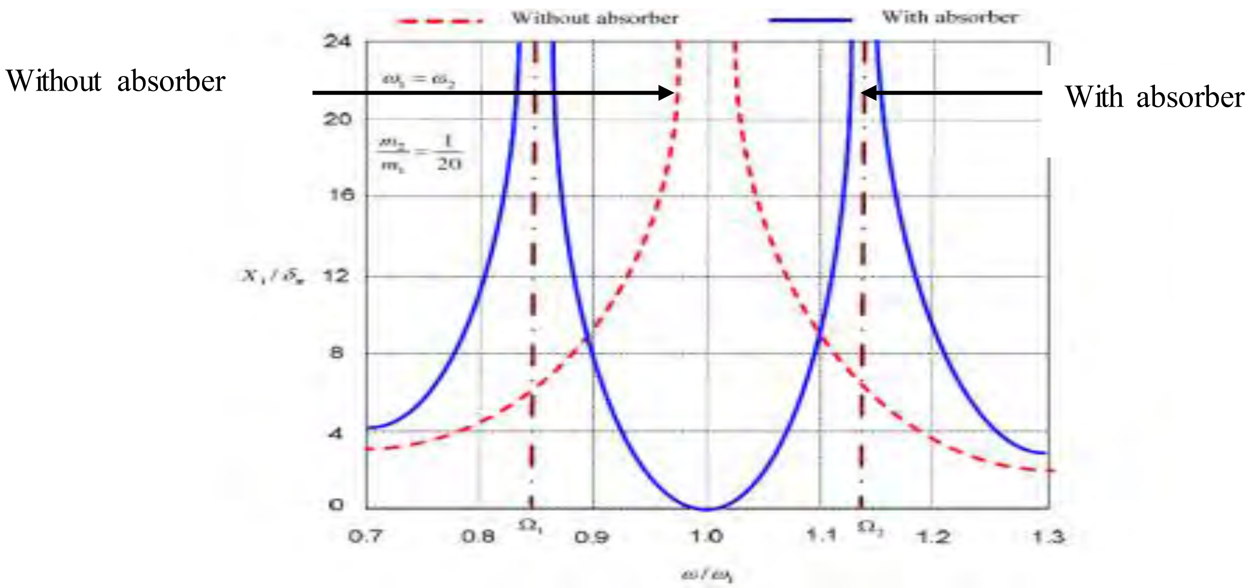


Figure 1.0 Vibration response of a primary structure when an absorber is attached (red line = without absorber, blue line = with absorber).

1.2OBJECTIVE

The objective of research studies are:

1. To study the properties of a non-linear single degree of freedom system.
2. To characterize the dynamic properties of the proposed absorber.
3. To investigate the performance of the absorber when put on vibrating structure.

1.3 SCOPE OF PROJECT

Scope of this project are:

1. Characteristic of absorber investigated until first mode of vibration only.
2. Only nonlinear characteristic of stiffness is investigated.

CHAPTER 2

LITERATURE REVIEW

2.1 Dynamic Vibration Absorber

The dynamic vibration absorber (DVA) which is also known as tuned mass damper (TMD) nowadays are widely implemented as a vibration control device on a structure as a primary system in order to mitigate the vibratory movement or undesired vibration. Mostly, in practical applicaton DVA is designed in very small mass which usually of a few percentage of the primary structure (Hoang et al, 2008). The simplest form of DVA consist of a mass and spring. When a primary system is subjected to a harmonic force, it will experiencing an unwanted vibration and this severity which is the undesired vibration phenomenon can be suppressed by attaching a DVA.

Fundamentally let say before the DVA was attached on the primary system, the primary system was a single degree-of-freedom (Dof) system which is only have one natural frequency, thus the addition of DVA which supressed the vibration amplitude at once providing a safe operating frequency range, resulting in a new 2-Dof system and it will be also having 2 new natural frequencies. Therefore if the operating frequency coincides either one of these frequencies, the system will again be at resonance (Liu & Liu, 2005). Thus, in order to overcome this problem, a damper is added to the DVA so as to supressed the new vibration amplitude to remains small at resonance. Many of successful aplication of DVA installed on a structure can be found such as skyscrappers and towers to supress undesired vibration induced by wind force and etc. These structures include the CN Tower in Canada, The John Hancock Building, the tallest building in the world Taipei 101 and the Center-Point Tower in Sydney (Lee et al, 2006).

2.2 Optimal design of vibration absorber

The dynamic vibration absorber was mentioned in the previous section suppress the original resonance amplitude in the vibration system response of the machine then at the same time generating two new peaks. Thus, the machine will suffer large amplitude of vibration as it pass through either one of the first or second peak such as when starting and stopping of the machine. As stated earlier this severity can be overcome by introducing a damper in the DVA system to reduces the amplitude of the machine. This section will describe on the optimal design theories and method done by previous researchers in order to get an optimally tuned DVA that provide better performance and control of the undesired vibration. Optimization is the method or technique in which some specific restriction that depend on engineering problem must be taken under consideration such as the optimization result must be economic physical and technically applicable (Bekdaş & Nigdeli, 2011). These consideration of optimization depend on the system of the DVA and the primary structure itself such as the linear or non-linearity and etc. In fact the fundamental design and concept of the DVA is quite simple, in order to result in a better control performance design of DVA the parameters such as mass, damping and stiffness of the DVA system must be obtain through optimal design procedures (Lee et al, 2006). These optimal design procedures can be found in method introduced by previous studies. Two step design technique was suggested by Den Hartog for optimum design of DVA that is still in use today and have been a fundamental and basic understanding to the other researchers. Den Hartog state that the resonance frequency must first be chosen and secondly determine the optimal level of damping (Jang et al, 2012). In his classical textbook of mechanical vibration, Den hartog introduced the fixed points theory as shown in Figure 2 which is showed that where all of the curve in the system response with different damping ratio intersect at two points called point independent of damping.

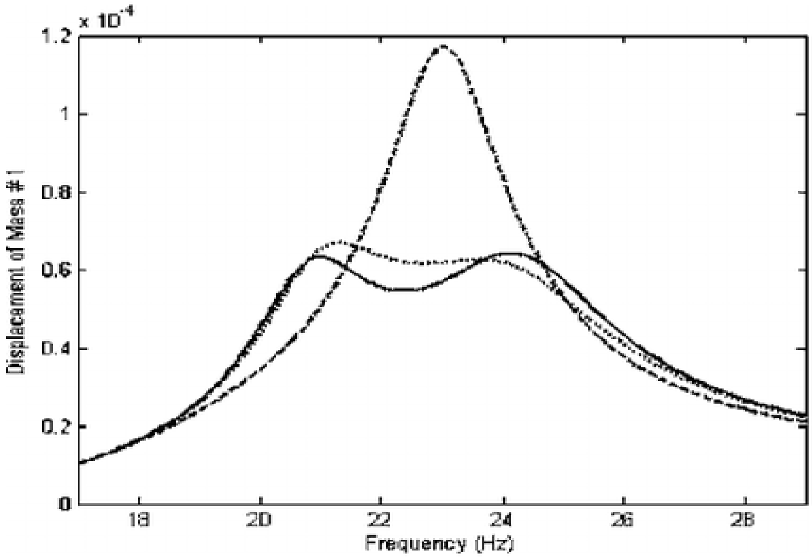


Figure 2.0 Den Hartog's method for multiple degree of freedom system.

He then further found that the parameter of optimum tuning such that resulting the ordinates of two point independent of damping will be in equal height and also revealed that the optimum value of damping ratio is the value that at which the response curve passes through horizontally either one of the point independent of damping (Liu & Liu, 2005).

Other design optimization aproach also can be observe such as in the previous study a comparison between different robust optimum design aproaches was investigated which is SORO (Single Objective Robust Optimization) and the MORO (Multi Objective Robust Optimization) in analyzing the DVA performance to understand it characterstics in structural optimization (Marano et al, 2010). In his studies, showed that both approaches minimized the mean and standard deviation of the system but the result later showed that the MORO approach was a good choice of design control approach compared to the another one due to its provide significant improvement in performance and better control of vibration reduction. Furthermore in another related studies such as the optimum parameters of the DVA under seismic excitation, a Harmony Search which is the method of metaheurestic optimization was used to optimize the DVA parameter including mass, stiffness and damping (Bekdaş & Nigdeli, 2011). Bekdas & Nigdeli in their studies in 2013 (Bekdaş & Nigdeli, 2013), showed again that the feasibility of Harmony Search approaches in optimizing the DVA parameter. From all of the related studies mentioned above, it shows that the optimization of the DVA is a must in order to get a better performance and system control efficiency on the vibration reduction and infact there is many consideration and approaches depending on the engineering natures.

2.3 Semi-active Dynamic Vibration Absorber

The mitigation of vibration by the DVA on the primary structure normally seen to be efficient when the primary system vibrates at focussed resonance frequency to which the DVA was applied. Some cases the primary system also may vibrate at another frequencies due to forced vibration which is the exciting of disturbing force on the primary system at the different frequencies other than target resonance frequency. Furthermore, it is also due to other resonance frequencies which is different resonance frequency of the primary system is excited than the resonance frequency that was used for the DVA design. Thus, due to all of this deficiency of the DVA, a various type of controllable DVA was developed nowadays. The active DVA are basically having different types of actuators in which the active part of the DVA are able to allow precisely both frequency and damping tunings at the same time.

Later on realizing that although the active DVA having a great performance and control system it has a drawbacks which is have a high power consumption needed.

To overcome the drawbacks experienced by the active DVA, a vast variety of semi-active DVA concept have been developed which is the system that assisted by controlled spring system, Piezo stacks and other actuators that will adjust the frequency of the semi-active DVA to the dominant frequency of the primary system. The implementation of controllable dampers also have been widely used to control the semi-active DVA energy dissipation or also in the purpose of increase the relative motion of the semi-active DVA for higher efficiency. The semi-active DVA robustness upon changes in the primary systems was it big advantage. In the cases of when the natural frequency of the primary structure changes such as due to some damage, the passive DVA will lose it effectiveness and may causing a higher response than the primary system with no passive DVA. In this situation for the semi-active DVA, it will continues to mitigate the vibration robustly. In the previous study on one of the semi active DVA type which is semi-active model as shown in Figure 2.1 Tuned Liquid Coulumn Dampers (sTLCD) by (Sonmez et al, 2016) shows that the semi-active DVA type provides more robust reduction of vibration amplitude than the passive type of TLCD when the excitations of the primary system vary.

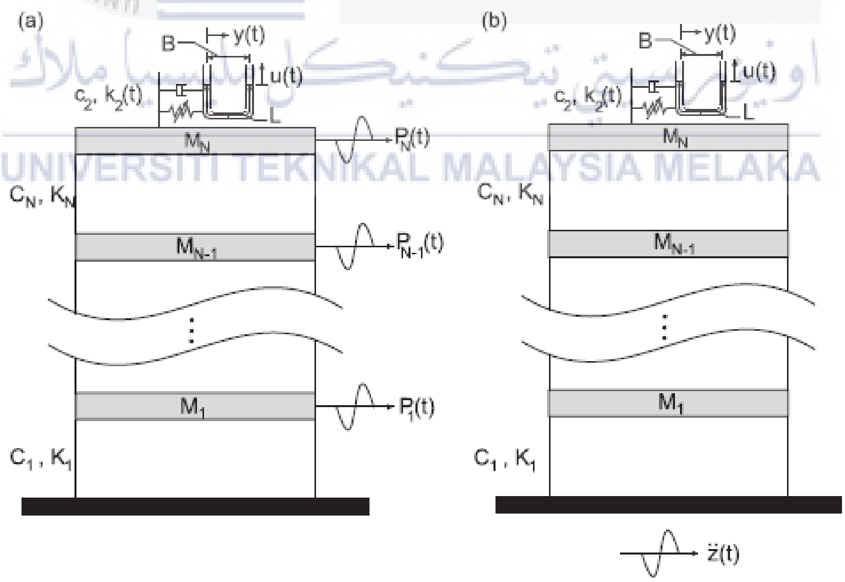


Figure 2.1 MDOF structural model with sTLCD: (a) forced excitation and (b)base excitation

Besides that, semi-active DVA types obviously can outperforms the passive DVA at some significant percentage, which again shows the performance and vibration control effectiveness of the semi-active DVA type (Weber, 2014).

2.4 Active Dynamic Vibration Absorber

Since the past few decades have witnessed various of research and works was carried out to overcome the passive DVA weaknesses by the suppression bandwidth enlargement of the DVA. Some of these research consider the application of an active DVA. An active DVA consist of an actuator that parallel with the damper and spring of the passive DVA. This will create an ability in incorporating control theory to provide cancellation forces. Besides that, a wider bandwidth of safe operating frequency range near the natural frequency of the primary structure and lower system frequency response within the suppression band can be yield or provide by the optimum active DVA compared to the optimum passive DVA. The previous study on the investigation of the efficiency of an active DVA type in controlling resonant vibration of linear vibratory system proposed an active DVA as shown in Figure 2.2 which utilize the state feedback taken from the absorber mass shows that with the tuning optimal parameter it performed better vibration supression than passive DVA (Chatterjee, 2010).

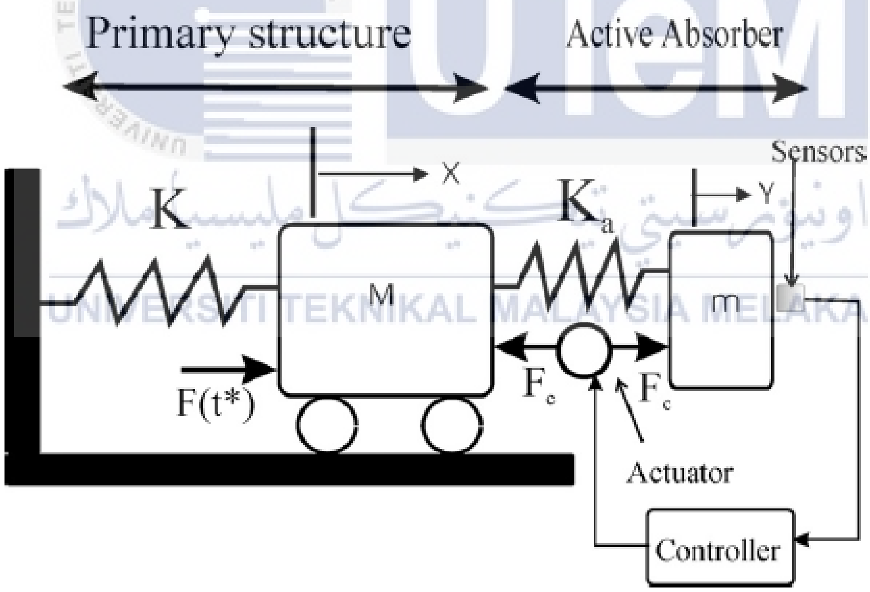


Figure 2.2 Model of the single-degree-of-freedom oscillator with the active absorber.

Many application of various smart material was used recently such as piezoelectric stack or patch form to suppress the vibration of the single degree of freedom , multi-degree of freedom and also in continuous system. Furthermore, due to the use of these sensor and actuator in an active DVA, as mentioned previously it provide counteracting force to the vibrating primary structure in order to mitigates its vibration (Mohanty & Dwivedy, 2016). By many of the research past and present it can be shows and prove that, the active DVA in vibration control have a benefit of the easily tunable parameters and can adapt for a more wider operating frequency range (Wu et al, 2007).

2.5 Non-linear Dynamic Vibration Absorber

The non-linear characteristics of a vibration absorber knowledge was a crucial in order if it performance and vibration control effectiveness is to be predicted precisely when it is to be connected to a primary structure. Although it can be predicted theoretically, but an experimental validation is important to verify all of the procedure and assumptions included. A study shows on how the nonlinear DVA can be experimentally determined by performing a free vibration test which is by having the nonlinear DVA attached to a mechanical vibration shaker then from the free vibration time history the amplitude and damped natural frequency was estimated by using Hilbert Transform (Tang et al, 2016). Both stiffness and damping of the nonlinear DVA are then calculated and estimated from these quantities. Furthermore, the nonlinear DVA also was applied in various purposes such as seismic mitigation, aeroelastic instability suppression, acoustic mitigation and chatter suppression as well and most of the nonlinear DVA targets the suppression of a nonlinear resonance (Detroux et al, 2015).

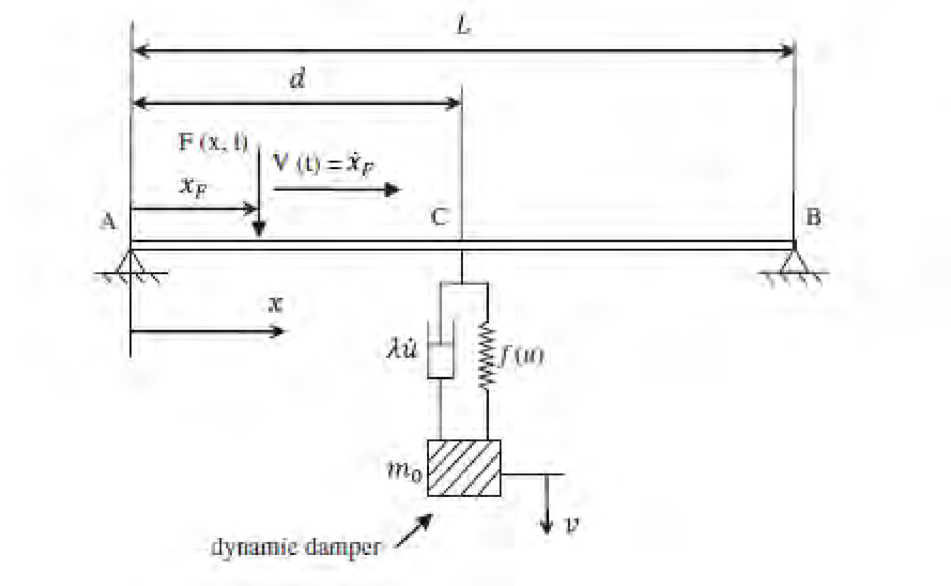


Figure 2.3 The beam model.

In the paper of study the linear and nonlinear DVA performances applied to a beams as shown in Figure 2.3 that was subjected by moving loads are investigated and the results of the study shows that essentially the nonlinear DVA are more effective in reducing the amplitude of vibration compared to the linear DVA (Samani & Pellicano, 2009).

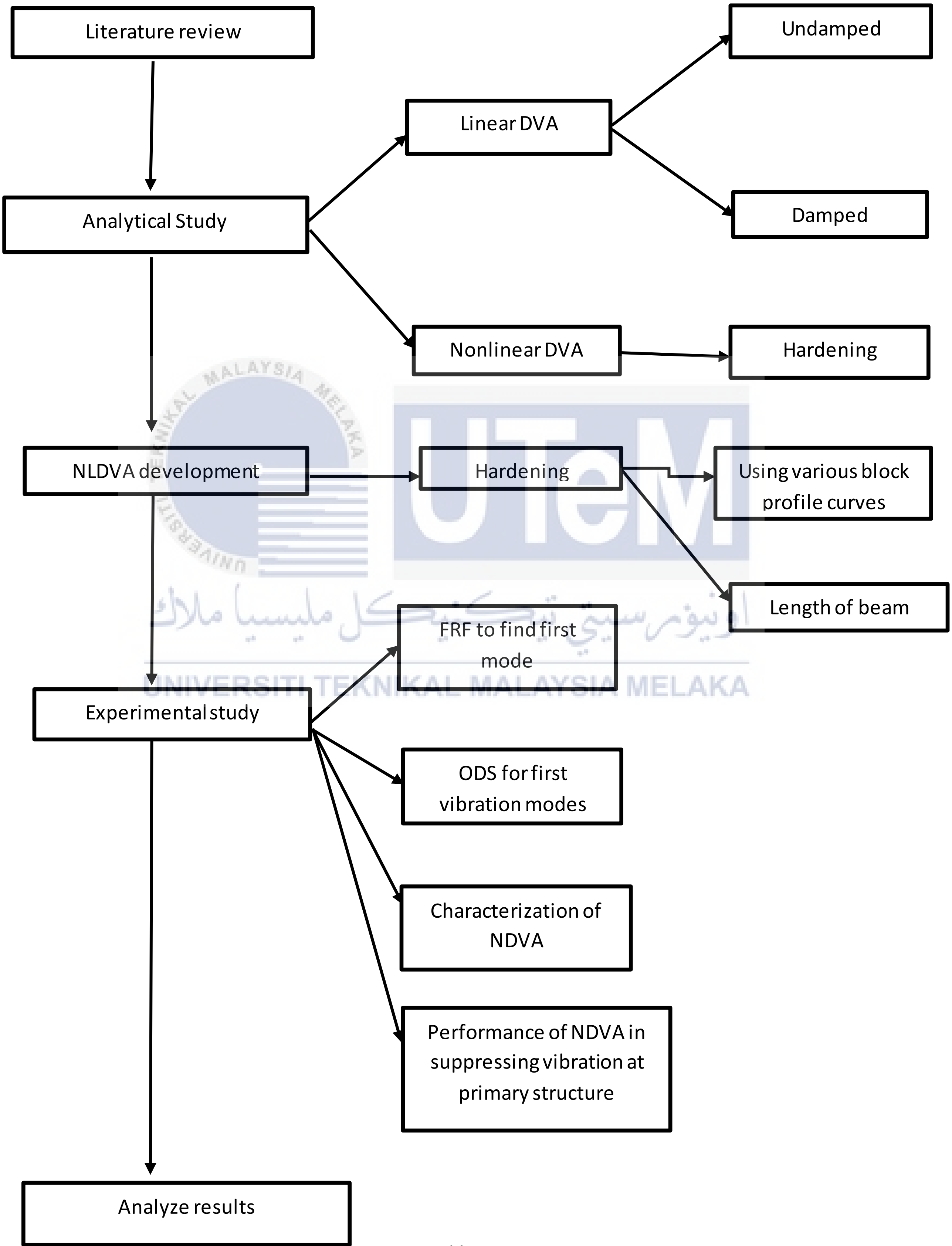
In the previous study, nonlinear absorber was used to suppress vibration and harvest energy (Zhang, 2015). Hardening by magnetic stiffness method is used to produce nonlinearity of spring. Magnets of same polarity were placed opposite to each other to produce repulsiveness. Smaller the gap, higher the nonlinearity of spring. Based on the results, nonlinear vibration absorber is useful in vibration mitigation and energy harvesting in wider bandwidth.

Furthermore in the previous research, the hardening and softening mechanism to widen the operational bandwidth of absorberwas investigated (Low, 2015). Three types of magnetic configurations were investigated. By moving the axial gaps between the magnets, stiffness of beam was altered. Altering the distance cause the stiffness either soften or harden. Both softening and hardening produce broad bandwidth of operation for absorbers. In this research hardening by profile curved block is used to produce nonlinearity of the absorber beam stiffness.

CHAPTER 3

METHODOLOGY

3.1 Flowchart



3.1 Analytical study of Linear and Non-linear Dynamic Vibration Absorber

3.1.1 Linear Dynamic Vibration Absorber (DVA)

The dynamic vibration absorber (DVA) which is an auxilary mass-spring system that is used for the purpose of mitigating or reducing the unwanted and undesired vibration. This DVA was attached to the primary structure that needs to be protected from the undesired vibration. The primary structure alone before attached with the DVA was a single degree-of-freedom (DOF) system having only one natural frequencies and as a result of attaching the DVA on the primary structure it become a 2-DOF system and thus it now have 2 natural frequencies. If all of the DVA system basic component which is spring, mass, and damper behave linearly it is known as linear vibration absorber. There is two types of DVA which is consist of undamped and damped DVA as shown in the Figure 3.1 (a) and (b) below.

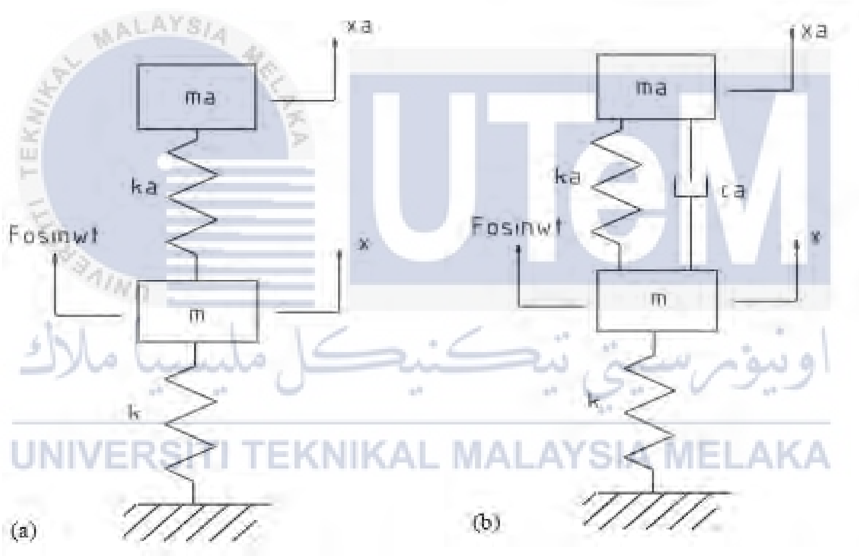


Figure 3.1 (a) The unndamped Dynamic Vibration Absorber; (b) The damped Dynamic Vibration Absorber.

From the diagram shown above primary strusture mass, m absorber mass, m_a primary structure spring constant,k absorber spring constant, k_a absorber damping coefficient, c_a horizontal displacement of primary mass, x horizontal displacement of absorber mass, x_a and $F_o\sin\omega t$ is the exciting harmonic force on the primary structure. Each of damped and undamped DVA supress the vibration amplitude of the undesired vibration in different behaviour or characteristics.

3.1.1.1 The undamped Dynamic Vibration Absorber

The undamped DVA which is only consist of mass-spring system as described before when is attached on the primary structure that need a protection from undesired vibration it will removes or supress the original resonances amplitude peak in the system frequency response providing a safe operating frequency range of the machine thus producing two new peaks as a result at once. If the system analyzed using newton's method, equation of motion of the system will be obtained. The structure and the undamped DVA equation of motion is given below as m is structure mass, m_a is absorber mass k is a structure stiffness, k_a is linear absorber stiffness, x and x_a are deflection of primary and secondary mass respectively.

$$m\ddot{x} + (k + k_a)x - k_a x_a = f_t \tag{1}$$

$$m_a \ddot{x}_a + k_a(x_a - x) = 0 \tag{2}$$

Relative motion of absorber spring can be denoted as z, where

$$z = x_2 - x_1. \tag{3}$$

Thus, equation 2 can be rewritten as,

$$m_a \ddot{x}_a + k_a z = 0 \tag{4}$$

In linear absorber spring of the absorber is a linear spring. It acts as in Hooke's Law. Hooke's Law suggest that when an elastic object is stretched, the increased length is called its extension. This extension of an elastic object is directly proportional to the force applied to it.

$$F_s = kx \tag{2}$$

In equation 2, F_s is spring restoring force, k is linear stiffness constant and x is extension of spring. Figure 3.2 shows that, if the elastic limit not exceeded, graph of force against extension produces straight line passes through the origin. Greater the value of k, the stiffer the spring becomes.

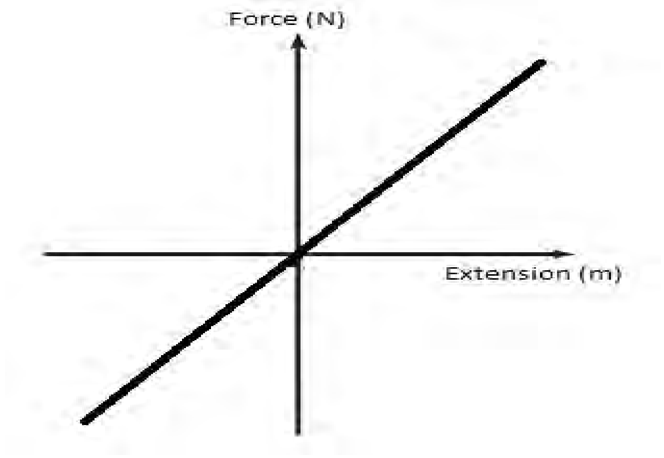


Figure 3.2: Graph of force against extension of spring.

As the machine exciting frequency passes through either these two new peaks such as when start-up or stopping the machine will experiencing large amplitude of vibration. Besides that, the DVA natural frequency is tuned to be equal to the frequency of the exciting force and thus as a result the mass of the primary structure doesn't vibrate at all but the DVA system will vibrate due to its spring forces generated is equal and opposite to the exciting force on the primary structure. Figure 3.2 below shows the frequency response of the primary structure when is attached with the undamped DVA.

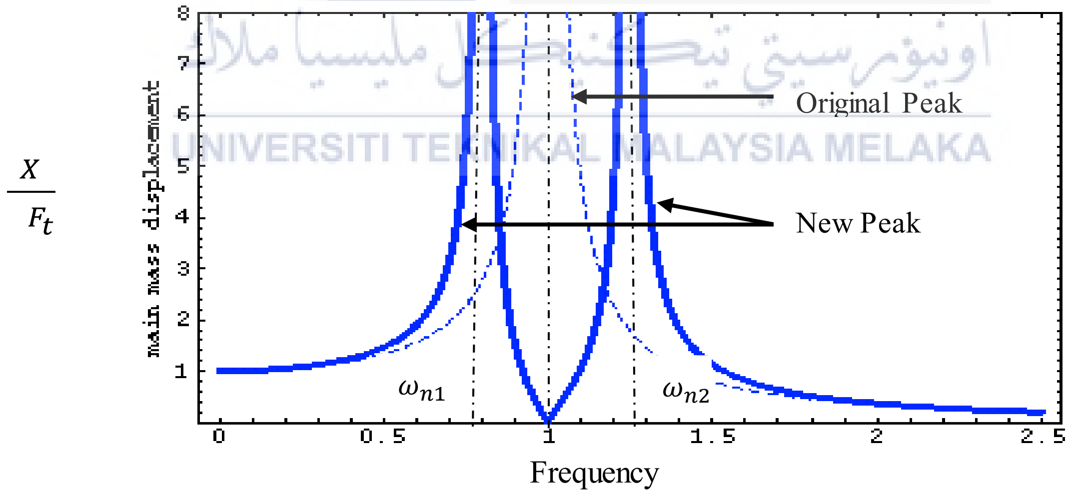


Figure 3.3: Frequency response curve of a primary structure attached with the undamped DVA.

From Figure 3.3 above the two new peaks are correspond to the two natural frequencies of the primary system attached with the undamped DVA. The frequency curve also shows that the force generated by the DVA spring is equal and opposite to the excited force on the primary structure thus reducing the original amplitude to zero.

3.1.1.2 The damped Dynamic Vibration Absorber

As been describe before the response of the vibration of the primary structure attached with the undamped DVA will provide a safe operational frequency range but at the same time having an infinite amplitude of vibration that will cause the system at resonance when at start-up and stopping. These amplitude of the system can be reduced by introducing a dashpot or damping in the DVA system thus it is called damped DVA. . Equation of motion for structure and damped DVA linear absorber is given below and c_a is damping coefficient of absorber.

$$m\ddot{x} + c_a\dot{x} - c_a\dot{x}_a + (k + k_a)x - k_a x_a = f_t \qquad (5)$$

$$m_a\ddot{x}_a + c_a\dot{z} + k_a z = 0 \qquad (6)$$

When the force of the damping doing a considerable work, the two new peak of vibration amplitude which at resonance will be reduced and remain small. However the suppression of these amplitude by the damping is depanding on the damping ratio itself. Figure 3.3 below shows that the effect of damping by the damped DVA on the purpose of supressing the amplitude in the composite system.

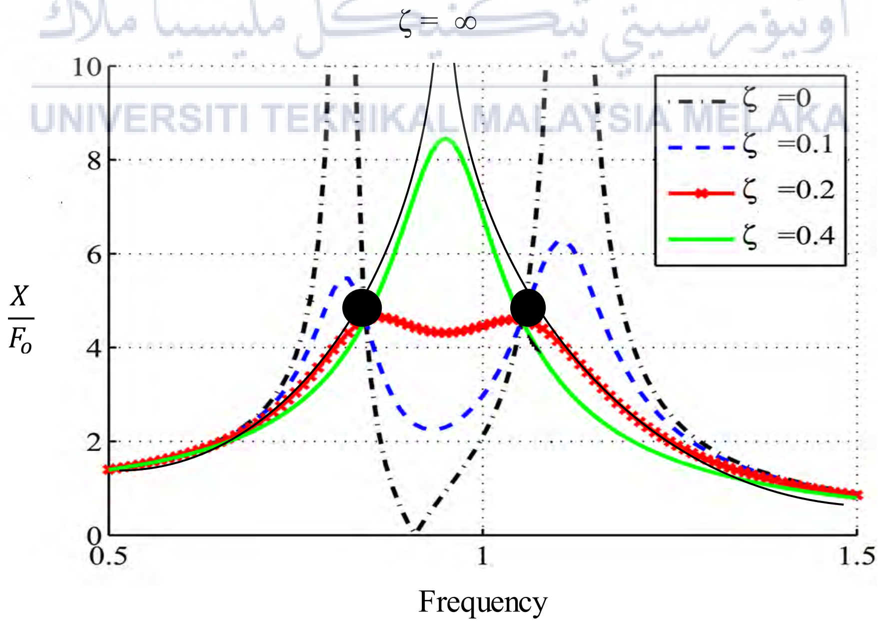


Figure 3.4: Amplitude of the main mass for various values of DVA damping.

From figure 3.4 above it is interesting to follow on what happen when the damping ratio is increased. The work done by the damping is associated by the force and its displacement. In this case, the displacement is the relative motion of the primary structure mass and the DVA mass; or also can be define as the extension of the damper spring. When the damping ratio value is zero, we will have the same result as the system with no damping. For infinite value of damping ratio, the mass of the primary structure and the DVA are virtually clamped together and it will behave like a single degree-of-freedom system. It is said between the damping ratio value of 0 and ∞ there is the value of damping ratio that will provide an optimum damping and at once resulting a small resonant amplitude which can be determine using the well known 'fix point theory' method. The two point of interception of curves shown in figure 3.3 is a fix point that indicates the point of independent of damping. The fix point theory method by Den Hartog state that the optimum tuning frequency, $f^*$and the optimum damping ζ^* can be estimated using the equation below.

$$f^* = \frac{1}{1+\mu} \tag{7}$$

$$\zeta^* = \sqrt{\frac{3\mu}{8(1+\mu)}} \tag{8}$$

Tuning frequency, f^* is used to obtained a same level of ordinate of the two fix point while the optimum damping, ζ^* is used to obtain a horizantal curve as possible passing the fix point. Both equation is used to find the optimum operation of vibration absorber.

3.1.2 Non-linear Dynamic Vibration Absorber (NLDVA)

For the case of non-linear vibration,the principle of superposition cannot be used and it is not valid, besides the technique of analysis is not well known. Due to most of the vibratory systems are more likely will behave nonlinearly when increasing amplitude of oscillation, a knowledge of the nonliear vibration become crucial in order to deal with it. Most of the cases in mechanical field the non-linearities occur in the spring or damper. A spring is said to be linear when the restoring force of the spring is increasing linearly with the spring displacement. Otherwise if the rate of force increment over a unit deflection increases the spring is said to be hard spring and if the spring's unit deflection increment over the rate of forces increases the spring is said to be a soft spring as shown in Figure 3.5 below. As mentioned, most of the spring used in practical application having a nonlinearities of force deflection relation, and more likely when there is large deflection.

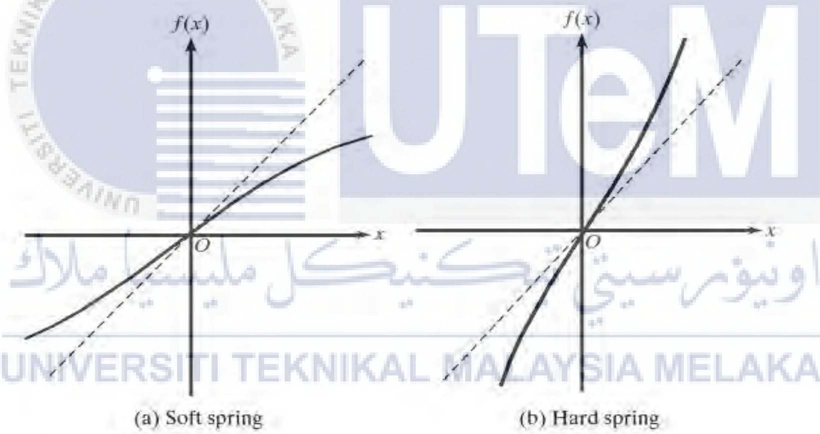


Figure 3.5: Non-linear spring characteristics (force-deflection-relation).

Figure 3.6 shows a cantilever beam having a stiffness and is constrained within a curve block. When the beam is deflected (due to vibration) ,due to it lies between the curve block which constrained the beam deflection, it will result in the shortening of its deflection thus becoming stiffer (hardening). Its force-deflection-relation will become steeper caused by the curve that constrained the beam from moving or vibrating freely.

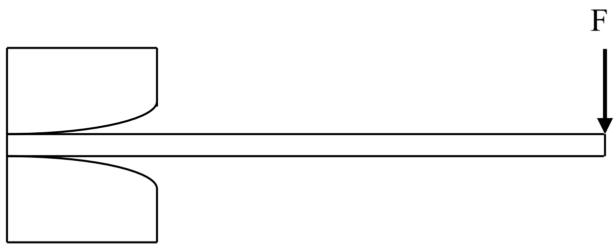


Figure 3.6: Curve constrained cantilever beam.

In the case of a non-linear DVA (NLDVA), that is attached on a primary structure for the purpose of mitigating the undesired vibration as illustrated in Figure 3.7, the resonance diagram for the NLDVA with gradually stiffening spring which differ from the linear DVA will be as shown in figure 3.8. Nonlinearity of springs can be induced by two methods, namely hardening and softening. Nonlinearity by hardening method will be investigated in this research. In a nonlinear absorber, cubic stiffness term z^3 and nonlinear stiffness constant k_3 will be added to left side of Equation 3. The equation now will be given as

$$m_a\ddot{x}_a + c_a\dot{z} + k_a z + \mathrm{k}_3\mathrm{z}^3 = 0 \tag{9}$$

Since $x_2 = z + x_1$, Equation 6 can be rewritten as

$$m_a\ddot{z} + c_a\dot{z} + k_a z + \mathrm{k}_3\mathrm{z}^3 = -\mathrm{m}\ddot{\mathrm{x}} \tag{10}$$

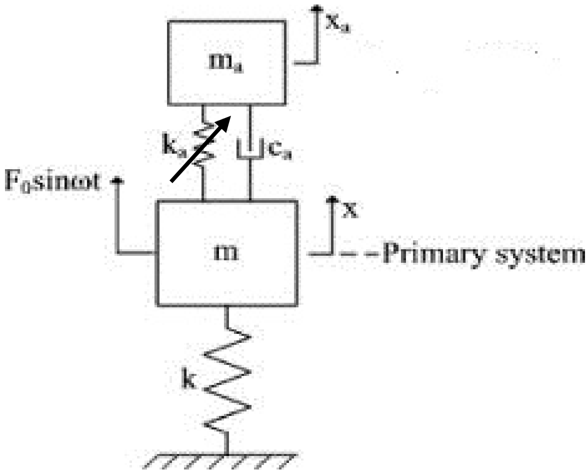


Figure 3.7: non-linear DVA with nonlinear spring attached on the primary system.

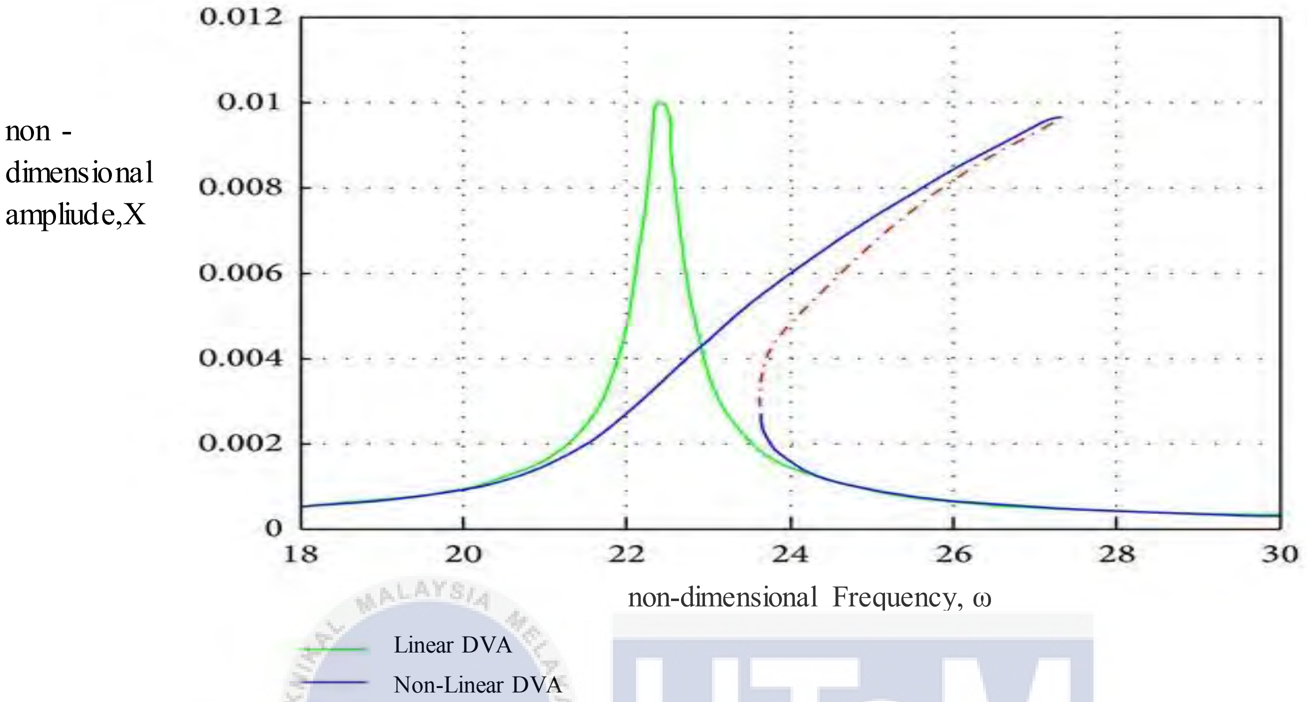


Figure 3.8: Resonance curve for the NLDVA with gradually stiffening spring.

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3.2 Setup Configuration of the design

3.2.1 Design of the Non-linear Dynamic Vibration Absorber

In this section the design of the non-linear DVA (NLDVA) that is used in this study and the method in of the hardening of the stiffness of the non-linear DVA will be shown. The NLDVA designed is from a cantilever beam having a mass at the end of the beam and the beam is lies or constrained between two curve which is called profile block. This profile block will constrained the movement of the beam when subjected to vibration from a primary structure where it is attached to, thus making the stiffness of the beam become stiffer. The design of the NLDVA used was as shown in figure 3.9 and 3.10 below.

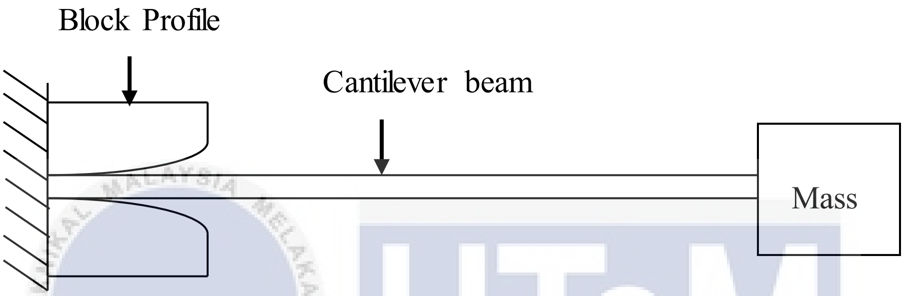


Figure 3.9: The schematic diagram of the non-linear DVA using curve constrained cantilever beam.

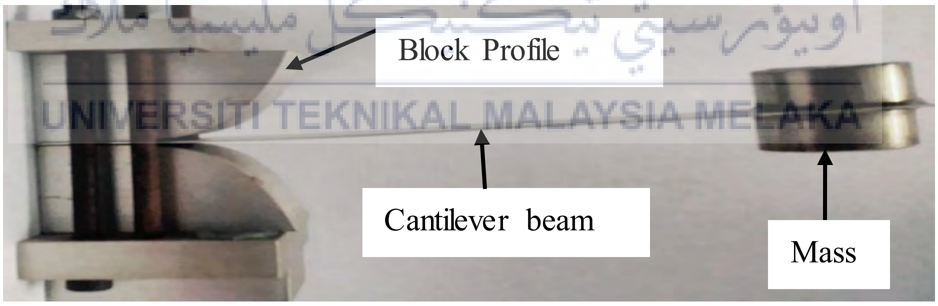


Figure 3.10: The design of the non-linear DVA using curve constrained cantilever beam.

The profile block purposes is to constrained the movement of the beam when vibrating thus making the beam become stiffer due to the hardening of beam stiffness. There will be various type of curve block profile shape from polynomial of, x^3 , x^4 , and x^6 as shown in Figure 3.11 will be used to harden the beam stiffness and it characteristics and performance on supressing the vibration on primary structure will be observe.

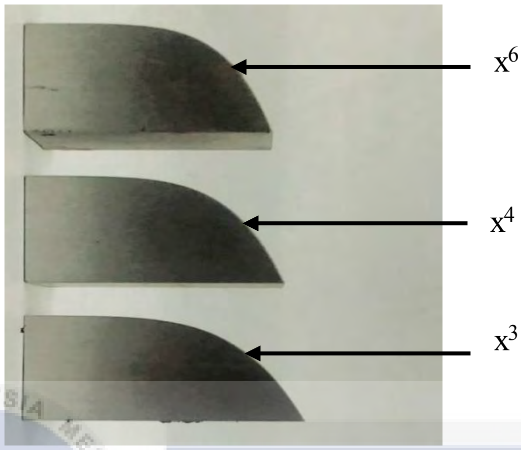


Figure 3.11 Variation of profile block for the NLDVA design used for hardening purposes.

Due to the beam is constrained with the profile block curve, and hardened the absorber beam stiffness, the restoring force of the beam is given by:

$$F_s = k_a x_a + k_3 {x_a}^3 \quad (11)$$

From the equation above, the term '$k_3 x^3$' represent the cubic stiffness due to the nonlinearity of the absorber when it is constrained by the profile block curve. Figure 3.12 shows the nonlinearity of force deflection relation of the hardening beam stiffness based on the curve profile block's polynomial number variation used.

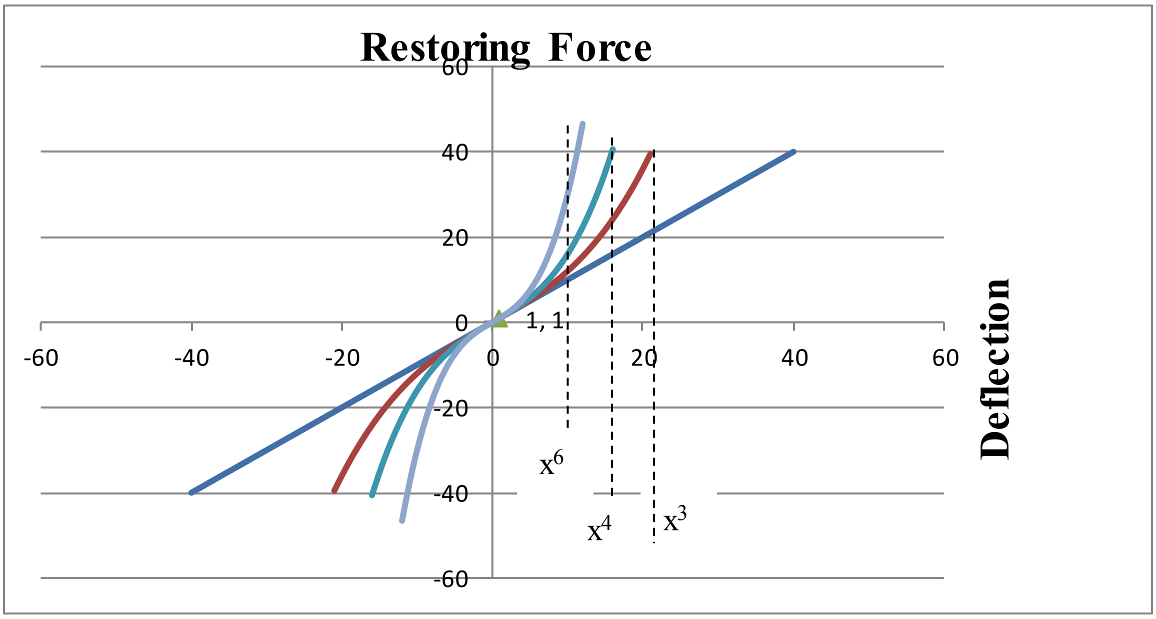


Figure 3.12 Force deflection relation of nonlinear DVA due to hardening of beam based on polynomial curve of profile block used.

If there are no profile block curve constraining the absorber beam, the overall stiffness of absorber is only depends on stiffness of beam. Stiffness of beam increases as length of beam decreases. Stiffness of beam depends on modulus of elasticity E, second moment of inertia I and length of beam L. Equation to calculate beam stiffness is given by,

$$k = \frac{3EI}{L^3} \tag{12}$$

3.3 Experimental Study

3.3.1 Mode of vibration of the beam

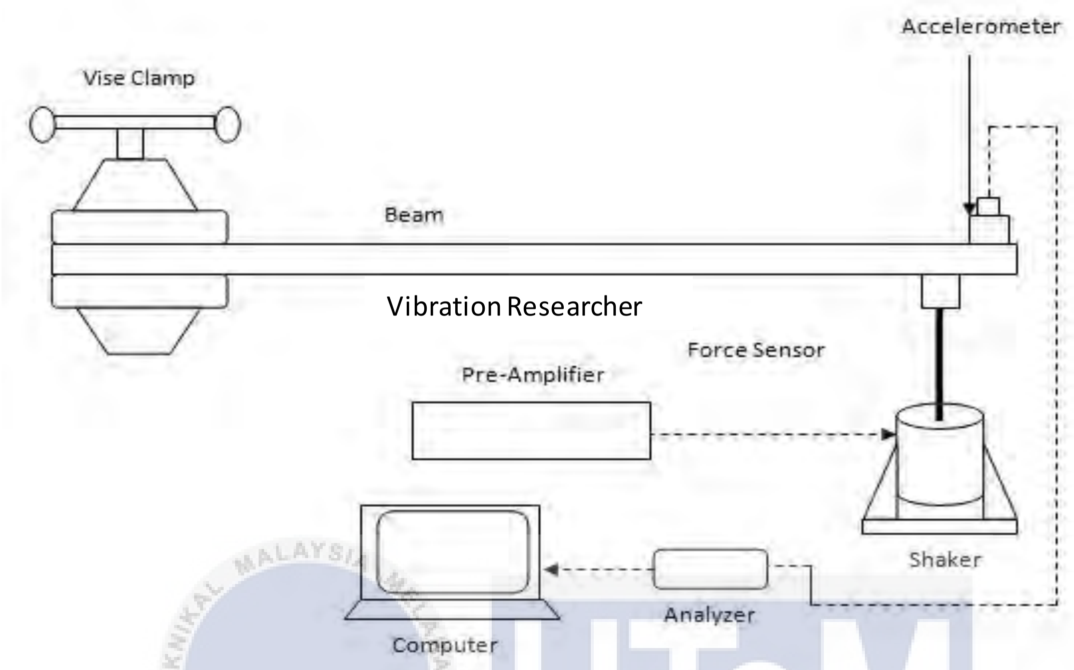


Figure 3.13: Schematic diagram of beam excited without absorber.

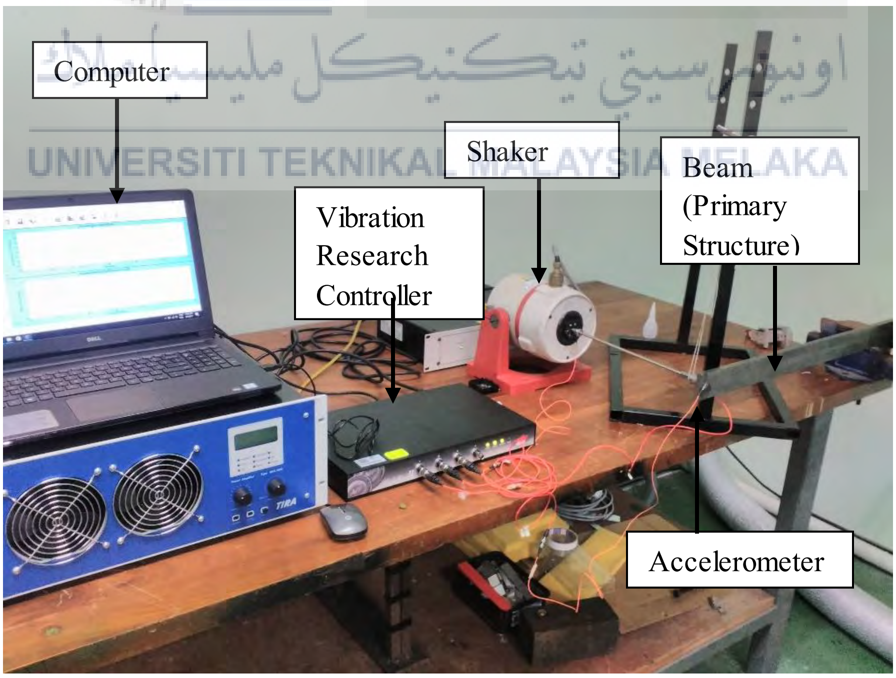


Figure 3.14: Equipment setup of beam excited without absorber.

The primary structure which is excited with random frequency without attaching the NLDVA as shown in Figure 3.13 to get the frequency response function (FRF). From the generated FRF the first first modes of vibration which is also known as natural frequency or resonant frequency will be used to characterize the performance of absorber on the primary structure. The FRF generated will have a several peaks but only the first peak will be consider and represent the first mode of vibration of the beam. The vibration mode of the beam which is modelled as a primary structure is depend on its Young's modulus, E, moment of inertia, I, mass per unit length, m, and length of beam, L. Besides retrieving from the FRF, by using the equation 13 and 14 below, the first mode of vibration is calculated first.

Equation to calculate the first mode of vibration of the primary structure beam is

$$\omega_1 = (1.875)^2\sqrt{\frac{EI}{mL^4}} \qquad (13)$$

An impact test also was conducted in order to determine the first mode of the primary structure beam. There will also several peak will appear on the FRF and only the first peak will be consider which is its first mode of vibration. The response will be compared with the theaoretical peak of a finite elements (FE) which is to validate the impact test on the primary structure beam result. Figure 3.15 shows a schematic diagram of the impact test on the primary structure beam.

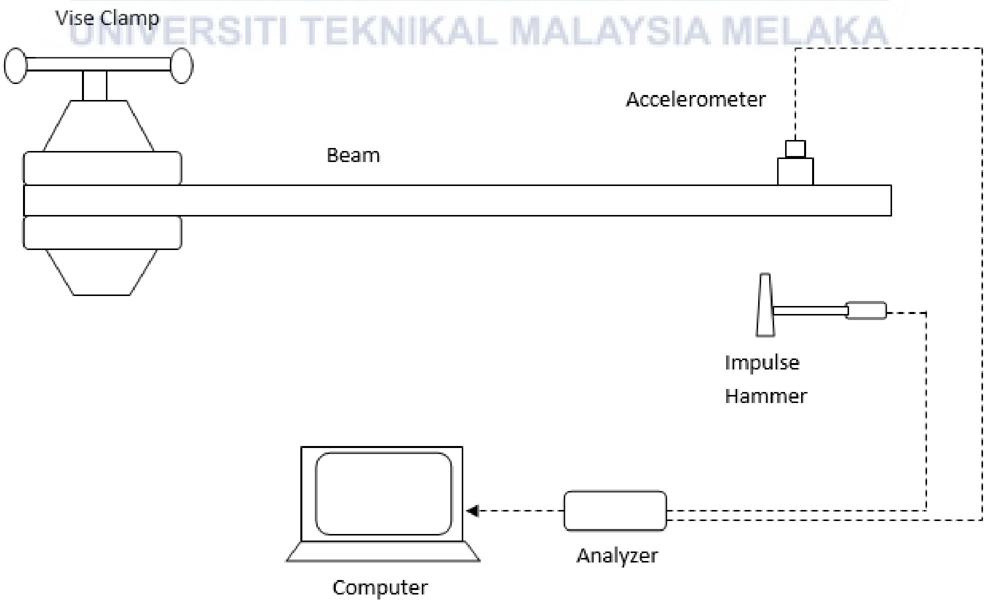


Figure 3.15: Impact test on the primary structure beam.

3.3.2 Operating Deflection Shape (ODS)

Operational deflection shape measurements was used which is the approach of the dynamic testing of structures in order to find their deformation at the critical frequency. In this method, under the operating frequency of the system the deflection shape of the beam is measured. The accelerometer is used for the measurement, which is one of it is mounted at some point on the beam as a reference and the other one which is non-stationary (fixed or mounted) is placed at several other points. The magnitude and phase differences between these non-stationary and reference accelerometers at all point under steady state operation of the system are measured. Thus the movement of the structure relative to one another can be observe by plotting these measurements.

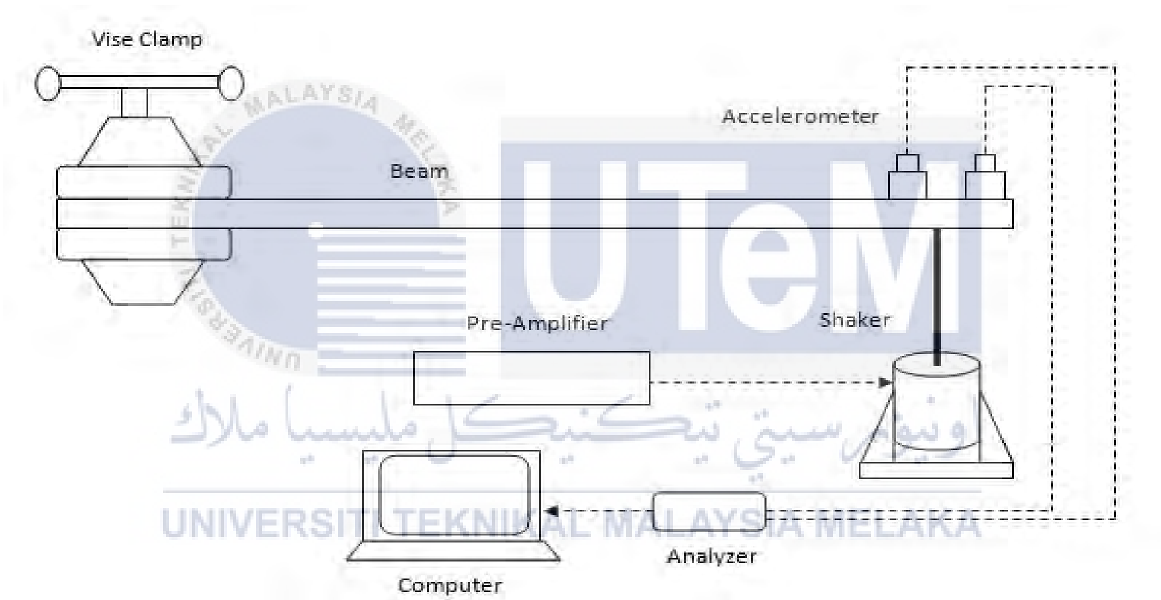


Figure 3.16 Schematic diagram to obtain ODS of the beam.

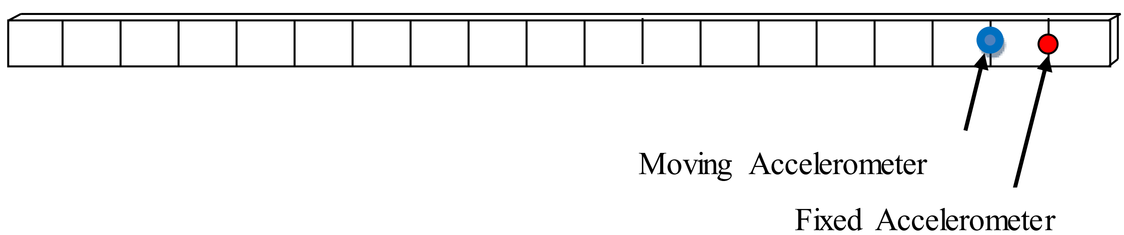


Figure 3.17 Beam marked with points to measure ODS.

After the first vibration mode been obtained, the operating deflection shape (ODS) will be define using VibShape software and the experimental set up is as shown in Figure 3.16. On the primary stucture beam, 18 point will be marked along the beam and magnitude and phase of vibration will be taken from this points using the accelerometer as shown in Figure 3.17. Shaker will pe placed at one point to excite the beam at the first mode. The reference accelerometer is fixed at one point on the beam and the another one accelerometer will be moved to each remaining 17 points marked on the beam one by one to obtained magnitude and phase of vibration at each point relative to the reference accelerometer. After all of the magnitude and phase of vibration on each 18 points is obtained for the first and second mode of vibration of the beam, then by using the VibShape software the beam deflection at each modes can be plot. When the results are simulated, operational deflection shape like Figure 3.18 will be obtained for first mode of vibration.

Figure 3.18 Mode shape for first mode of vibration.

3.3.3 Experimental Setup Configuration

The experiment will be set up as shown in Figure 3.19 which the non-linear DVA that have been designed using a curve constrained cantilever beam will be attached using a clamp on the primary structure which is a beam that will be excited with a force generated by mechanical vibration shaker. Figure 3.19 shows the schematic configuration setup of a beam as a primary structure attached with the designed non-linear DVA.

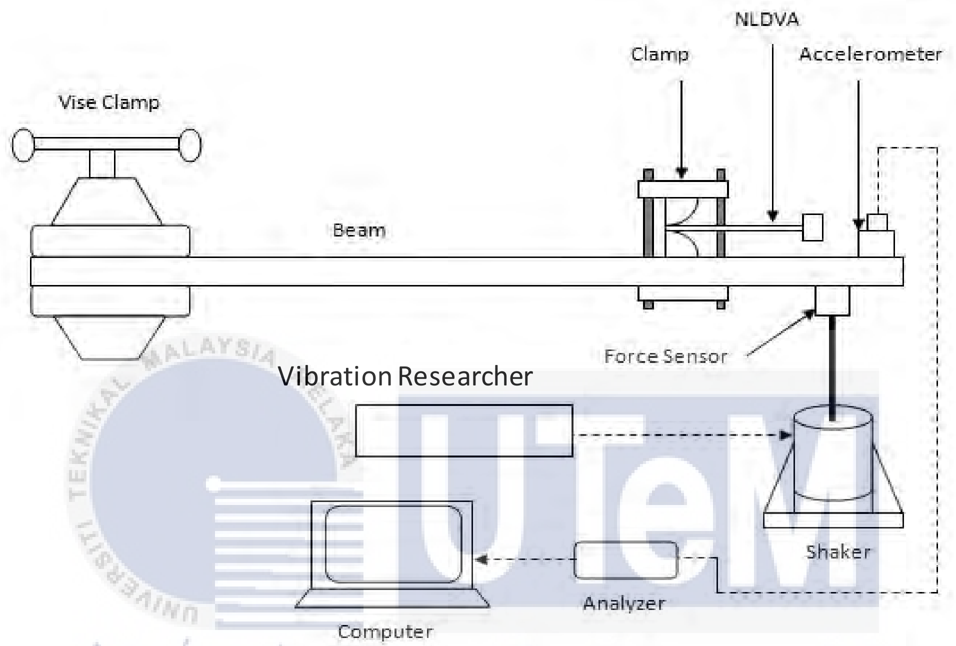


Figure 3.19: Primary structure attached with the designed NLDVA.

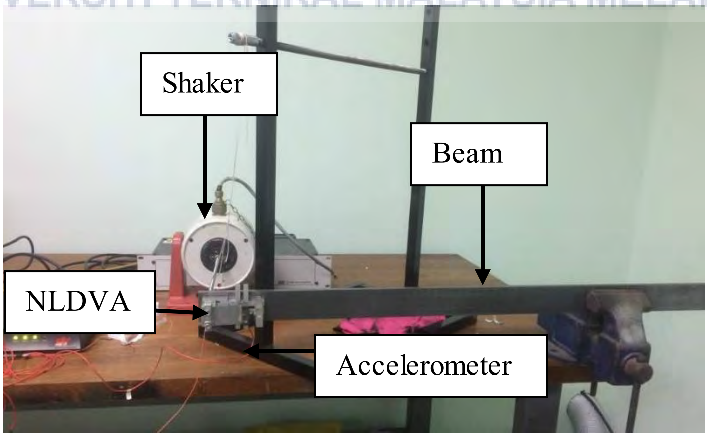


Figure 3.20: Equipment setup primary structure attached with the designed NLDVA.

The primary structure beam will be excited to its first mode of vibration by the vibration shaker. The primary structure beam is equipped with accelerometer as a sensing element which it will measure the vibration response of the system. The vibration signal sensed by the accelerometer was then transfered to the data acquisition system and encodes it into digital form. After been encoded, the computer will display the data which is the FRF by using analysis software. The performance of the NLDVA in suppressing the vibration magnitude of the primary structure at its first mode of vibrations also will be recorded and analyse.

The beam of the primary structure used in this experiment is made from Mild Steel. This type of beam is highly resistant to wear. Specification of the beam is shown in Table 1

Table 1: Specifications of beam

Characteristics	Specifications
Length, mm	670
Width, mm	50
Thickness, mm	10
Density, kg/m^3	7850
Young's Modulus, N/m^2	2^11
Moment of inertia, m^4	4.1667^-9

3.3.4 Characterization of Nonlinear Dynamic Vibration Absorber

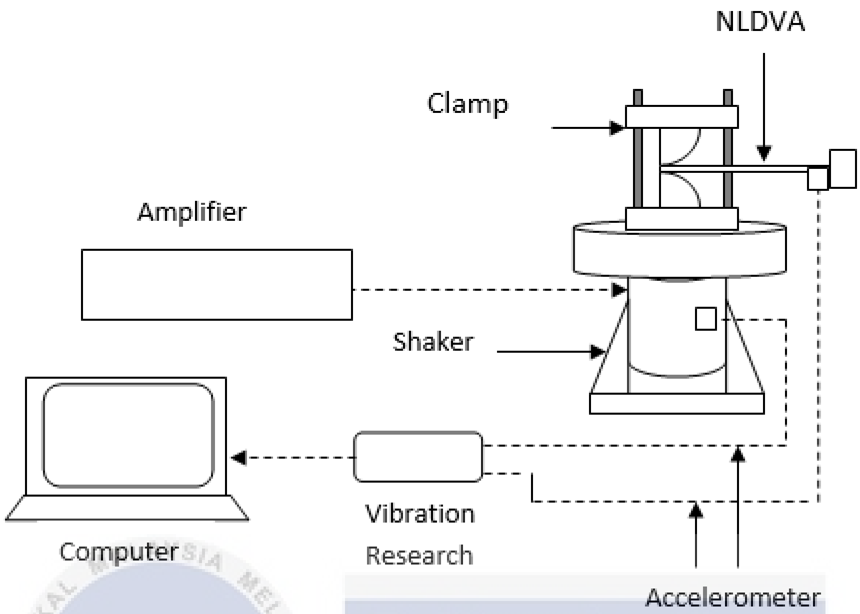


Figure 3.21: Schematic diagram of characterization of NLDVA.

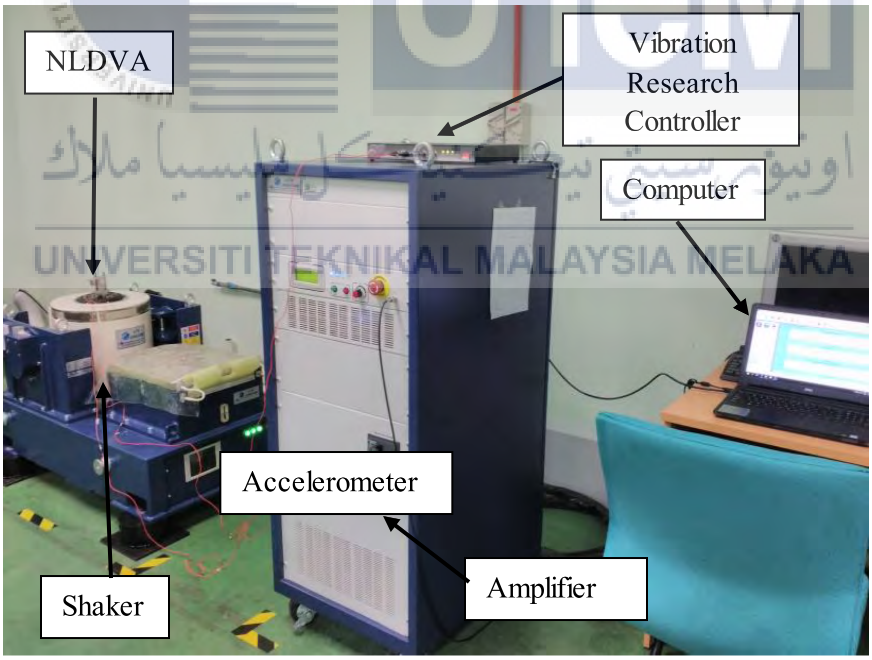


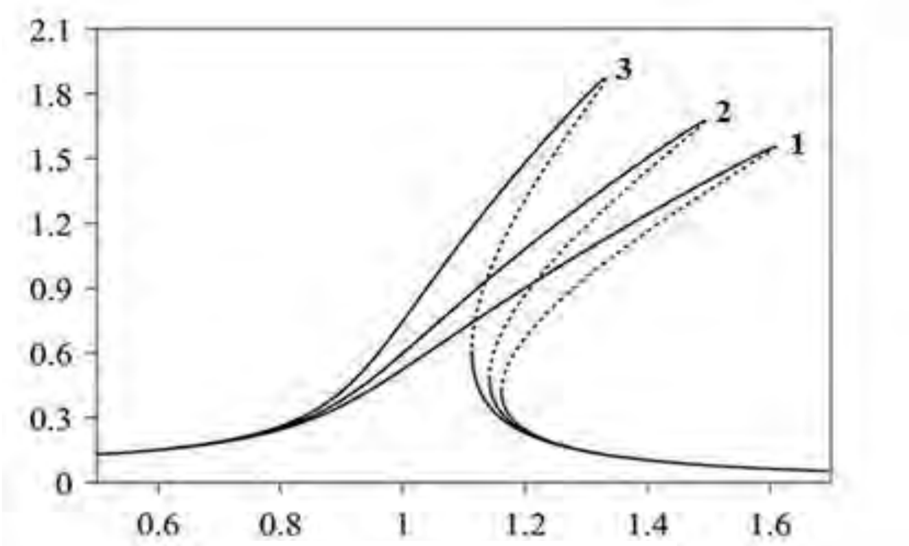
Figure 3.22: Schematic diagram of characterization of NLDVA.

Figure 3.21 shows the schematic setup for NLDVA characterization. The NLDVA will be mounted to the shaker as shown in Figure 3.20. In characterising the nonlinear vibration absorber, it is known that the stiffness of the absorber is controlled by the beam itself and the profile block which constraining the deflection of the beam when vibrating, thus stiffing up the beam which is also known as hardening. There are two parameter that will be control for the designed absorber which is the length of its beam and the variation of the profile block. For the beam, the sufficient length will be consider which is the length that will be needed to provide correct or exact stiffness as well as the variation of the profile block used, so its vibration supression response on the primary structure is around or near the frequency of interest.

Firstly, in the characterization of NLDVA the length of the absorber beam will be adjusted or varied. Shorter in length of the absorber beam will tune the absorber peak frequency response shifted to the right while longer in length of the absorber beam will tune the absorber peak frequency response shifted to the left. It is becouse when the absorber beam is short, it will vibrate at high frequency while when the absorber beam is longer it will vibrate at low frequency when excited at any given frequency.

After the length of the beam is adjusted to the sufficient length, then the nonlinearity adjustment of the NLDVA by using various type of profile block curves, x^n with different polynomial value, n is carried out. Three type of profile block curve will be used in this experiment which is x^3, x^4, x^6. The higher the value of n, the higher the nonlinearity of the absorber will be which means that the nonlinear curve of absorber will be longer and bandwidth of operation of absorber will be wider . Curve labelled 1 in Figure 3.23 shows an example of absorber response with higher nonlinearity than the curve labelled 2 and 3 with decreasing its nonlinearity respectively. Both adjustment of the absorber beam length and the block profile curve used is to ensure that the vibration suppression by the designed NLDVA on the primary structure beam is around or near the frequency of interest.

non - dimensional ampliude,X



non-dimensional Frequency, ω

Figure 3.23: FRF of nonlinear absorber for different nonlinearity.

3.3.5 Performance of Nonlinear Dynamic Vibration Absorber at Different Location.

The performance of the absorber depends on its position in the structure for the first mode of vibration. There will be three different location on the primary structure beam used to attach the absorber for each excited vibration mode which is the first mode. By observation and analysis from the FRF of the primary structure on each point, the best location of the absorber performance can be indicated for the first mode.

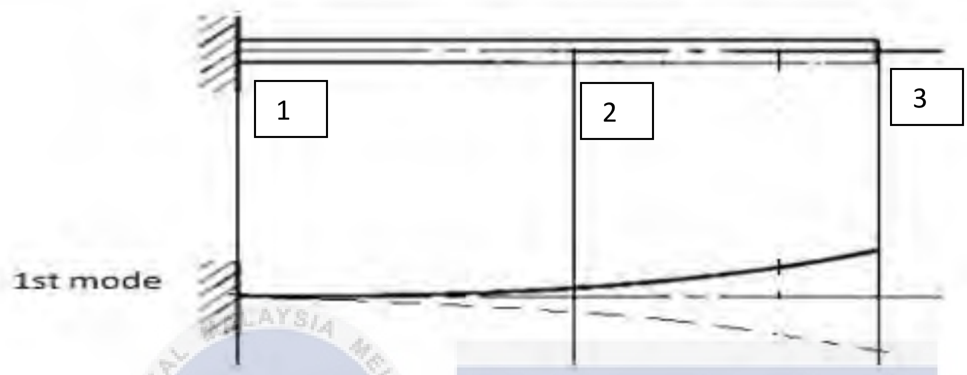


Figure 3.24: Points absorber will be placed.

From the three point on the primary structure beam used to indicate the absorber performance as shown in Figure 3.24 , one position will shows the best performance of the absorber in vibration suppression of the primary beam structure.

CHAPTER 4

ANALYTICAL CALCULATION

4.1 Vibration Modes of Primary Structure

The beam modelled as primary structure will be vibrated at random excitation frequency without attaching absorber as shown in Figure 3.11.From the FRF first modes of vibration will be used to characterize the performance of absorber on the primary structure. Modes of vibration is also known as natural frequency or resonant frequency of a structure. First mode of vibration will be analytically calculated first. The calculation of the first and mode of vibration of the primary structure beam can be calculated using equation 13 . Using equation 13, the first mode of vibration mode of the beam will be

$$1.875^2\sqrt{\frac{(200\times10^9)(4.1667\times10^{-9})}{(7850)(0.050)(0.01)(0.67)^4}} = 114.1154\,\frac{rad}{s}$$

$$114.1154\,\frac{rad}{s} = 19.0192\,Hz \approx 19\,Hz \qquad (14)$$

Then the operating deflection shape (ODS) for both mode of vibration will be obtained.

4.2 Operating Deflection Shape (ODS)

The deflection of the primary structure beam under the first mode of vibration can be observe by using the ODS. It makes easier to find a location for the placement of the absorber in suppressing the vibration purposes. The deflection of the primary structure for the first mode can be ilustrated as Figure 4 below. As shown in Figure4 the deflection at the free end marked as 1 will be the maximum thus in this case which is for the first mode of vibration, the absorber should be placed at the point marked as 1. However the testing of the absorber performance will also be tested at the middle and at the fixed end of the primary structure beam. The best placement of the absorber either at the free end, middle position, or fix end of the primary structure beam will be determined experimentally.

Figure 4.0 : ODS for the first mode of vibration.

4.3 Characterization of Nonlinear Dynamic Vibration Absorber

Figure 4.1 below is the mathematical modelling of the primary structure attached with the nonlinear dynamic absorber NLDVA. Due to the primary structure is attached with the nonlinear absorber, the system having 2 degree of freedom thus will be having two new natural frequencies.

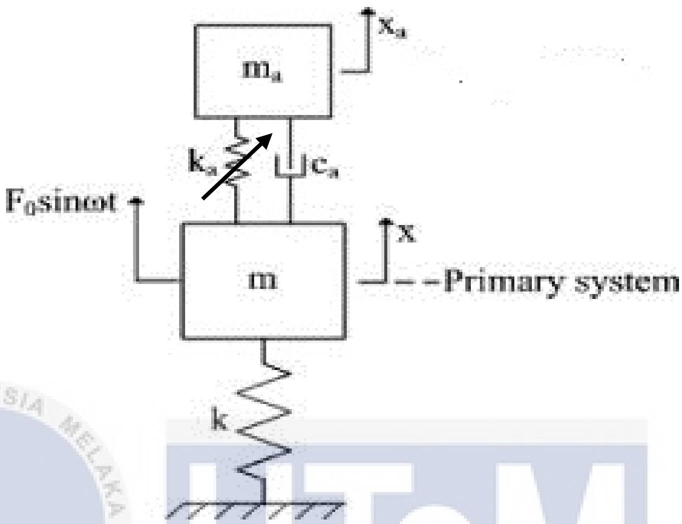


Figure 4.1 : Primary structure attached with nonlinear dynamic vibration absorber.

The stiffness of the nonlinear absorber which is $k_a x_a + k_3 {x_a}^3$ are simplifiy as $k_n = k_a + k_3$. Thus the equation of motion of the composite system can be calculated as follows

$$\begin{bmatrix} m & 0 \\ 0 & m_a \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{x}_a \end{bmatrix} + \begin{bmatrix} c_a & -c_a \\ -c_a & c_a \end{bmatrix}\begin{bmatrix} \dot{x} \\ \dot{x}_a \end{bmatrix} + \begin{bmatrix} k + k_a & -k_a \\ -k_a & k_a \end{bmatrix}\begin{bmatrix} x \\ x_a \end{bmatrix} = \begin{bmatrix} f_t \\ 0 \end{bmatrix} \tag{14}$$

Damping parameter will be neglected due to its insignificant small value. Then the system is differentiated in complex form of

$$x = Xe^{j\omega t} \tag{15}$$

$$\dot{x} = j\omega X e^{j\omega t} \tag{16}$$

$$\ddot{x} = -\omega^2 X e^{j\omega t} \tag{17}$$

Then equation 19 and 20 will obtained

$$(k + k_n - \omega^2 m)X - k_n X_a = F(t) \tag{18}$$

$$(k_n - \omega^2 m_a)X_a - k_n X = 0 \tag{19}$$

Transfer function of the output over input will be

$$\frac{X}{F} = \frac{(k_n - \omega^2 m_a)}{(k + k_n - \omega^2 m) - k_n^2} \tag{20}$$

The deflection of the beam, $X = 0$ when the nonlinear absorber is attached to the beam. Thus,

$$k_n - \omega^2 m_a = 0 \tag{21}$$

$$\omega = \sqrt{\frac{k_n}{m_a}} \tag{22}$$

As mention before, $k_n = k_a + k_3$ which mean k_n consists of linear constant, k_a and nonlinear constant k_3. The manipulation of the linear stiffness can be done by changing the length of the absorber beam. Thus it will shift the absorber peak either to the left or to the right. While the nonlinear stiffness is depends on the variation of profile block curves used.

In order to suppress the vibration of the primary structure the linear stiffness must be near or around the excitation frequency. It must be ensured that the safe operational frequency of the absorber is around the resonance frequency of the primary structure. This condition can be done by shifting the peak of the non-linear absorber to match the natural frequency of theprimary structure. Table 2 below is the design specification for the nonlinear dynamic absorber in order to suppress the vibration of the first mode of vibration of the primary structure.

Table 2: Properties of absorber to suppress first mode of vibration of primary structure

Design Specification	Value
Absorber beam heaight, mm	0.5
Absorber beam width, mm	25
Absorber beam length, mm	120
Absorber mass, kg	0.0334

For the safe operational frequency band of the absorber, it can be manipulated by changing the type of profile block curve as shown in Figure 3.11 which is consist of profile type x^3, x^4, x^6. The nonlinearity stiffness of the absorber increase as the profile block curve type's polynomial number increase from 3 to 6 and this method is called hardening the stiffness.

CHAPTER 5

RESULTS AND ANALYSIS

5.1 Characterization of Nonlinear Dynamic Vibration Absorber

The characterisation of the nonlinear absorber was basically by using various type of profile block curve which is consists of curve block with profile x^3, x^4, and x^6. But at a certain extent and limitation, a modification has been made on the previous nonlinear absorber design. This modification is done in order to get a nonlinear response of the nonlinear absorber. In order to get the nonlinear response, when the absorber is vibrated, the deflection of the nonlinear absorber's beam should be constrained by the curve block and this phenomena is not happening with the previous design of nonlinear absorber. Even though the maximum displacement amplitude of the shaker have exceeded the nonlinear absorber beam is still not constrained by the curve block and still behave linearly. As shown in Figure 5 the modified design consists of a stopper which is attached on the profile curve block in order to be able to constrained the deflection of the nonlinear absorber beam when vibrated and thus a nonlinear response will generated.

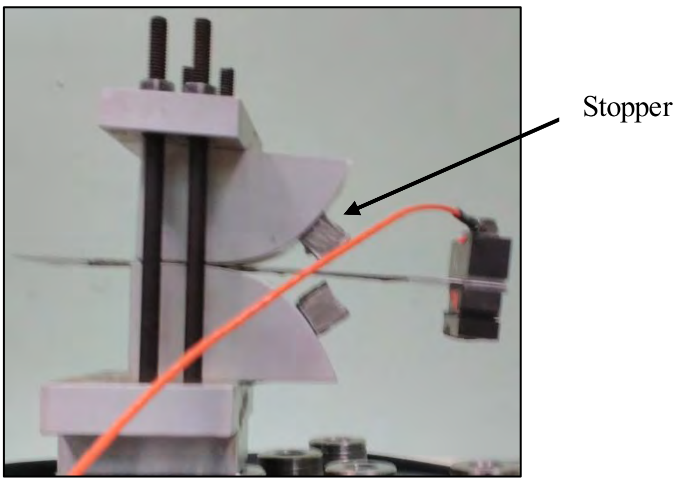


Figure 5.0 Modified nonlinear absorber.

By doing the modification on the nonlinear absorber, the absorber beam is successfully be constrained by the curve block when the absorber is vibrated. As the modification have been made, the parameter in characterising the nonlinear absorber is the effect of various effective length of absorber beam, effect of various gap between the stopper and absorber beam, effect of various level of vibration displacement input excited on the absorber, effect of various profile curve block used which is x^3, x^4, and x^6 without attaching the stopper.

The characterisation by using various effective length of the nonlinear absorber beam is to adjusting the peak weather shifted to the left or right while the other characterisation parameter is kept constant. The characterisation by using various gap between the stopper and absorber beam is to determine the nonlinearity of the nonlinear absorber with each gap used while the other characterisation parameter is kept constant. The characterisation by using various level of vibration displacement input excited on the absorber is to determine its effect on the nonlinearity of the response while the other parameter is kept constant. The characterisation by using various type of profile block curve is to determine its effect on the nonlinear absorber response weather it will generate a nonlinear response or not.

5.1.1 Effect of various effective lengths of absorber beam

For the effect of length of the nonlinear absorber beam on its response, there is two beam length was used which is 7.5cm and 9 cm. Profile curve block used is x^3 and the gap between the stopper and the nonlinear absorber beam is 1.9cm and with input level of 3mm for both beam length used. Figure 5.1 and 5.2 below shows the response of the nonlinear absorber with absorber beam length of 7.5cm and 9 cm respectively.

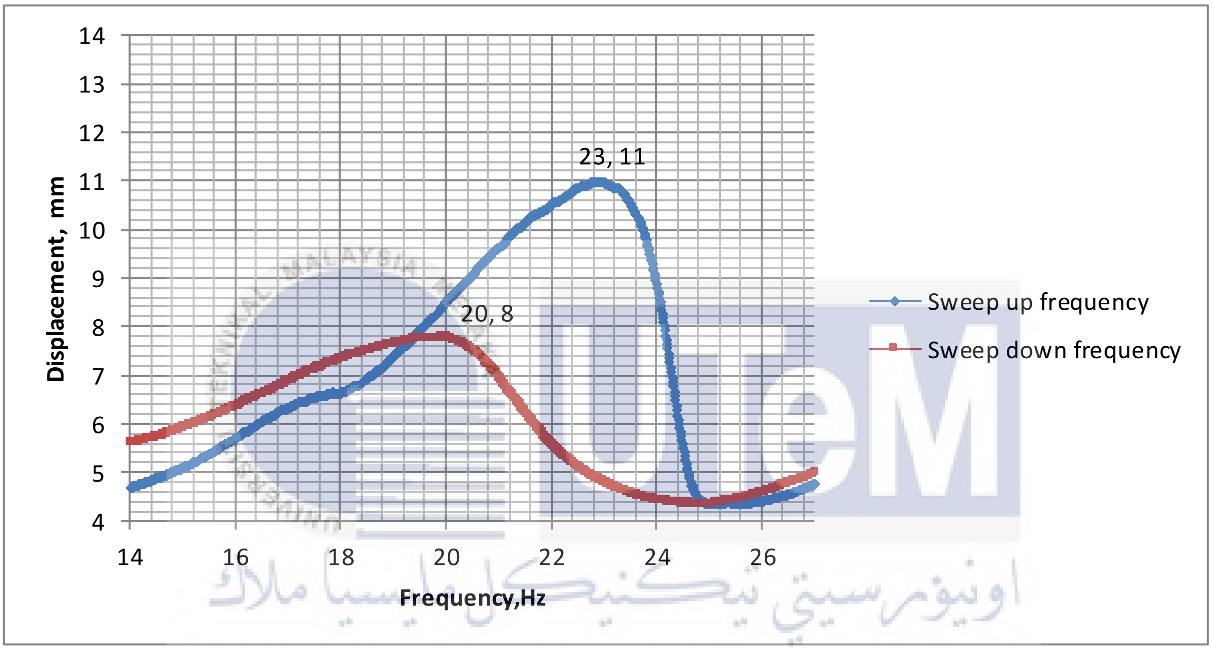


Figure 5.1 Response of the nonlinear absorber with beam length 7.5 cm.

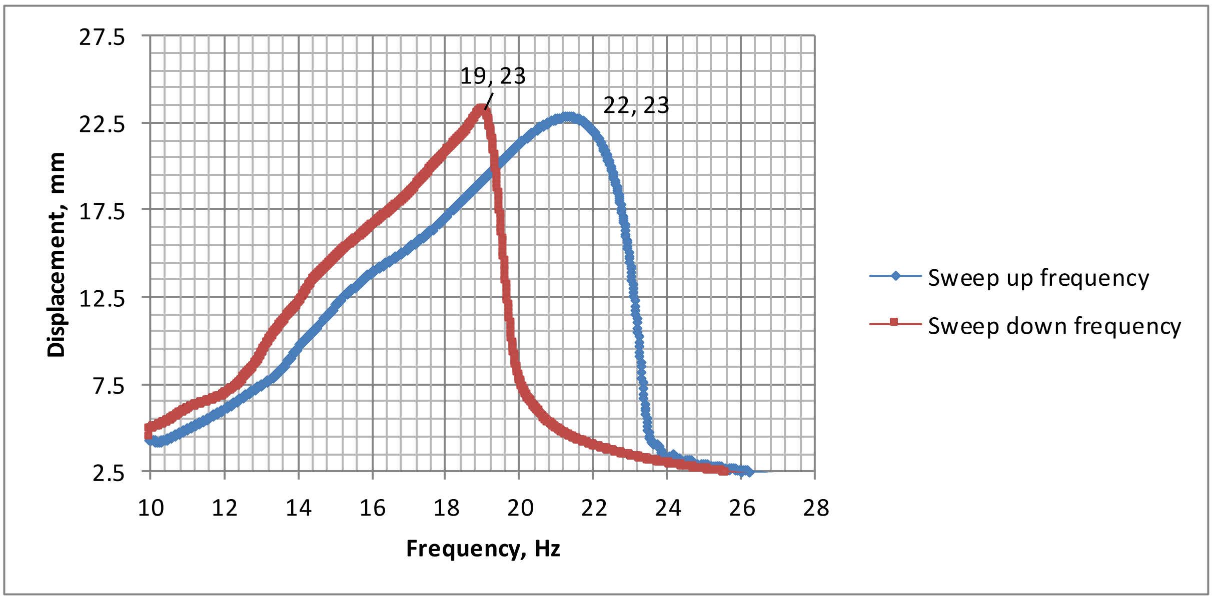


Figure 5.2 Response of the nonlinear absorber with beam length 9 cm.

From the response plotted, the ranged between the sweep up (jump down) and sweep down (jump up) frequency indicates that, the wider the range between the sweep up and sweep down frequency the larger the range of frequency of vibration can be suppressed by the nonlinear absorber. In this case which is the effect of the beam length used, from Figure 5.1 for length of the beam of 7.5 cm it shows that the range between jump up and jump down happen at high frequency compared to when the beam length used is 9cm as shown in Figure 5.2 it shows that the particular range is shifted to the lower frequency. From these result it shows that the length of the absorber beam will effect the peak of the nonlinear absorber responses. If the beam used is longer in length, the peak of the response will be shifted to the left which is to the lower frequency while for the shorter beam used the peak of the response will be shifted to the right which to the higher frequency.

5.1.2 Effect of various gap between stopper and absorber beam

For the effect of the gap between the stopper and the absorber beam, the gap of 1mm, 1.9mm and 3mm have been used while the length of the beam and the vibration input is keep constant. Figures 5.3, 5.4 and 5.5 show that the response of the nonlinear absorber with the gap between the stopper and the absorber beam of 1mm, 1.9mm and 2.5mm respectively.

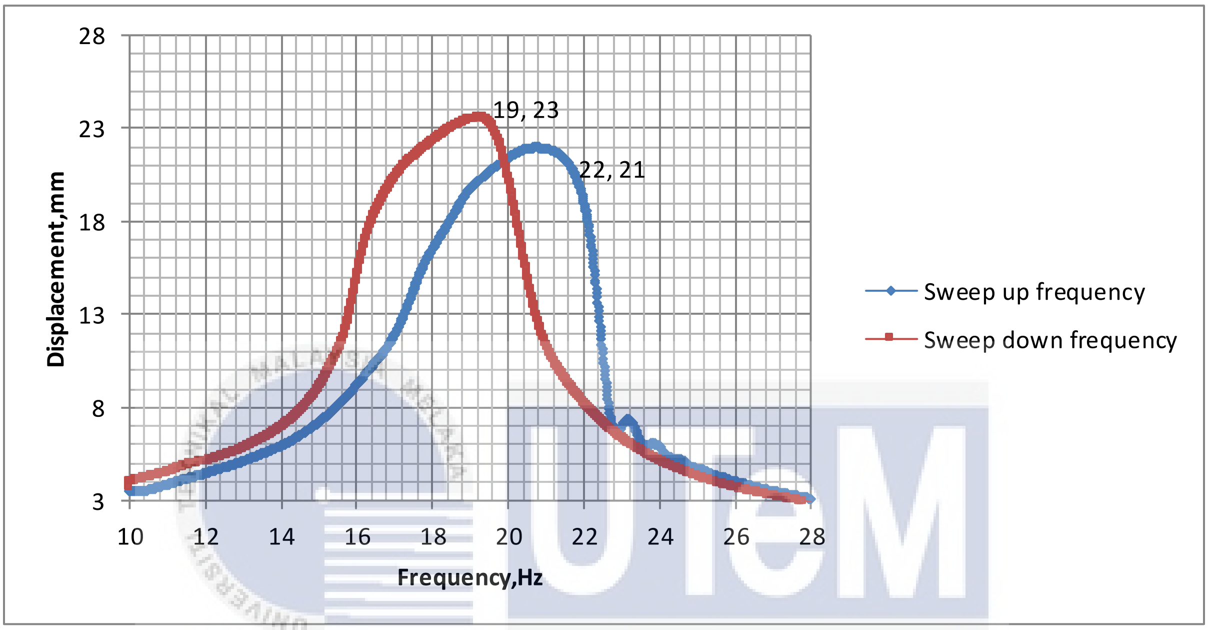


Figure 5.3 Response of the nonlinear absorber with 1mm gap.

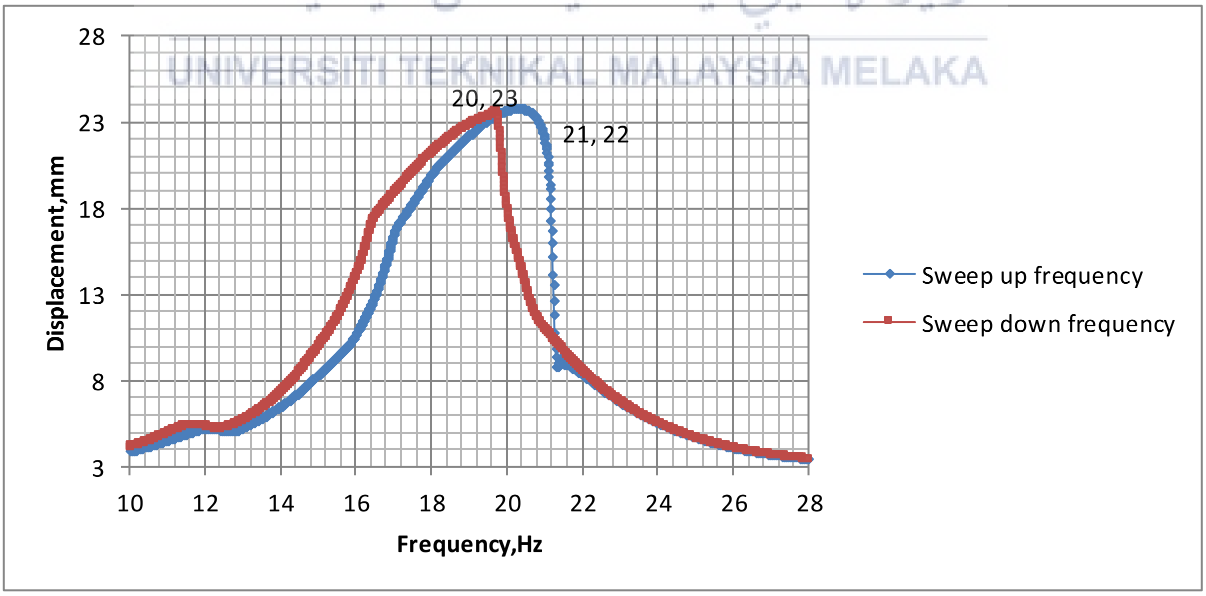


Figure 5.4 Response of the nonlinear absorber with 1.9mm gap.

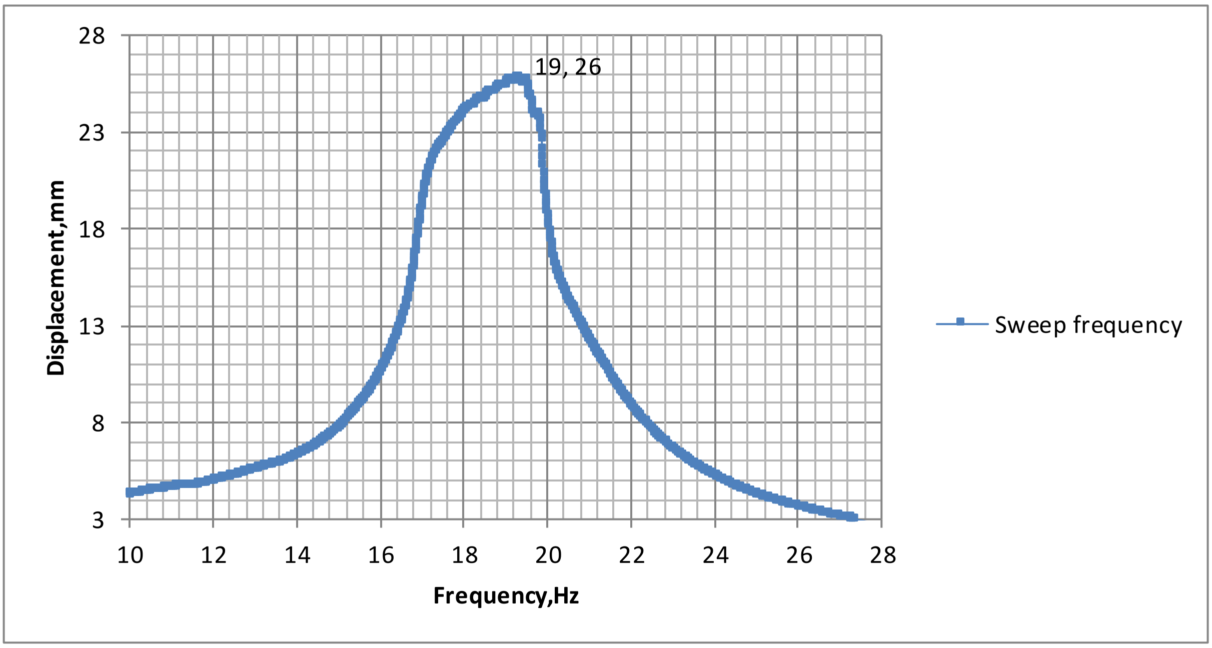


Figure 5.5 Response of the nonlinear absorber with 2.5mm gap.

The smaller the gap between the stopper and absorber beam, the higher the nonlinearity of the absorber response. As shown on Figure 5.3 which is the response of the smallest gap used which is 1mm behaves with higher nonlinearity. From the response, the nonlinearity behaviour can be indicated by the width between the sweep up and sweep down frequency. The wider these widths the higher the nonlinearity of the response of the absorber. Figure 2.5 shows that when the gap is increase to 1.9 mm the width between sweep up and sweep down frequency get a little bit smaller and thus decreasing its nonlinearity behaviour. When the gap is increase to 3mm the response behave linearly as shown in Figure 5.4. It is because the deflection of the beam is no longer constrained by the stopper which function is to harden the deflection of the beam. The response for the gap of 3mm used can be made to behave nonlinear by increasing the vibration displacement input so the the deflection of the absorber beam will again constrained by the stopper and it response will behave as nonlinear response.

5.1.3 Effect of vibration displacement input

For the effect of vibration displacement input, the input level used is 1mm and 3mm while the length of the beam and the gap between the stopper and absorber beam is keep constant which is 8.1 cm and 2.5 mm respectively.

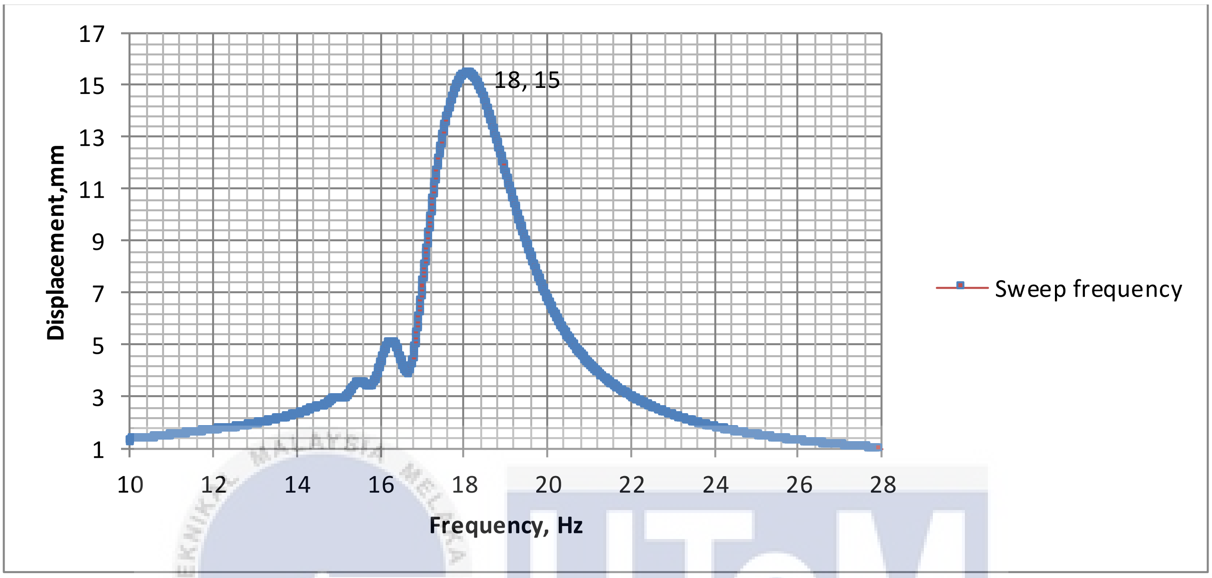


Figure 5.6 Response of the nonlinear absorber with 1mm input.

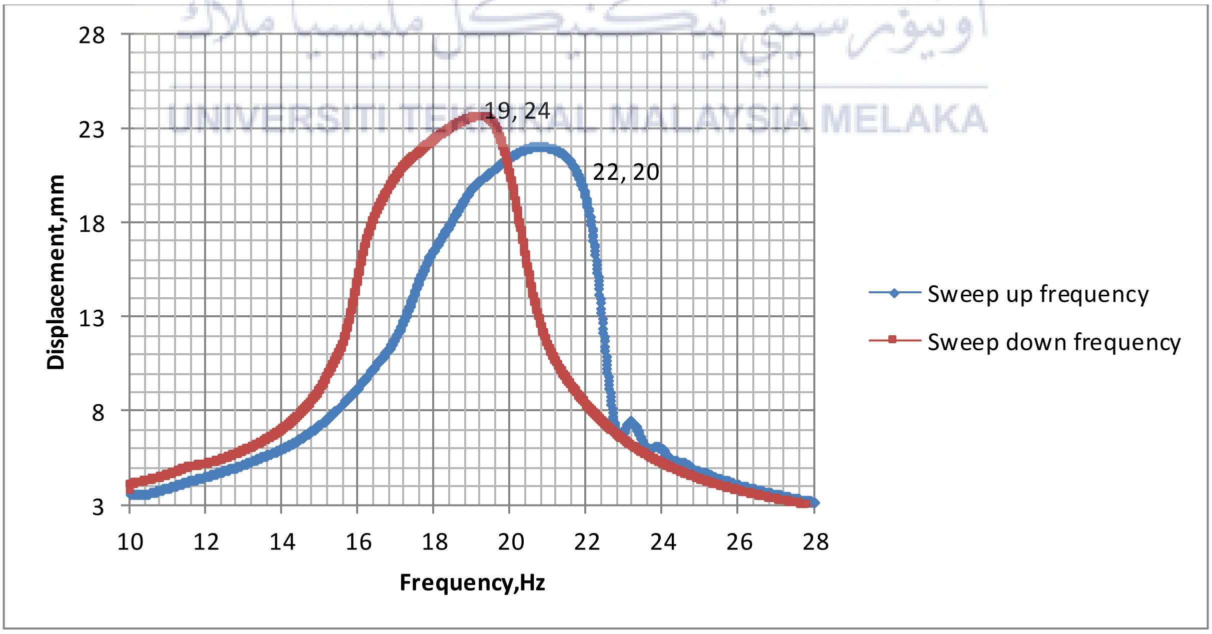


Figure 5.7 Response of the nonlinear absorber with 3mm input.

The higher the input level, the higher the peak of the response. As shown on Figure 5.6 and Figure 5.7. When the response is higher, the nonlinearity behaviour increase. As shown in Figure 5.7 when the input level is increase to 3mm, the response become nonlinear. In the Figure 5.6 when the input level is low, the vibration level is not enough to make the absorber beam to be constrained by the stopper and thus the response will behave linearly. In these case for the gap between the stopper and the absorber beam used which is 2.5mm, it might be to wide in order to get a nonlinear response with only 1mm input. However, the nonlinearity of the response appears when the input level is increased into 3mm which is a sufficient input in order to make the absorber beam been constrained by the stopper.

5.1.4 Effect of various profile block curves without stopper

For the effect of various profile block curve, block curve with profile of x^3, x^4, and x^6 have been used while the length of the absorber beam, the input level is keep constant at 9cm and 3mm respectively with no stopper attached on the block curve.

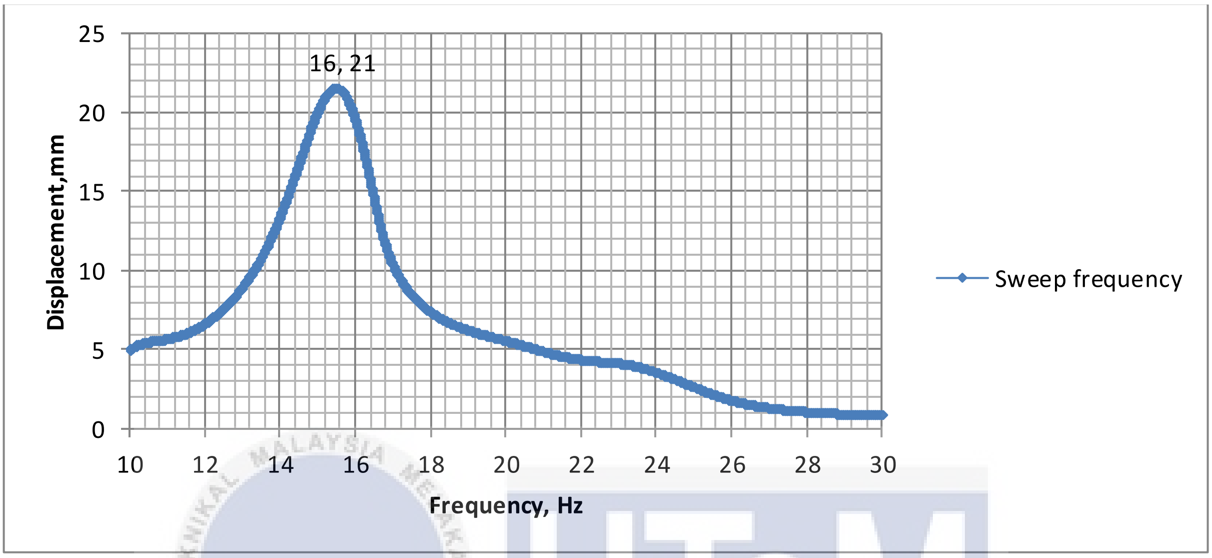


Figure 5.8 Response of the nonlinear absorber with x^3 block curve.

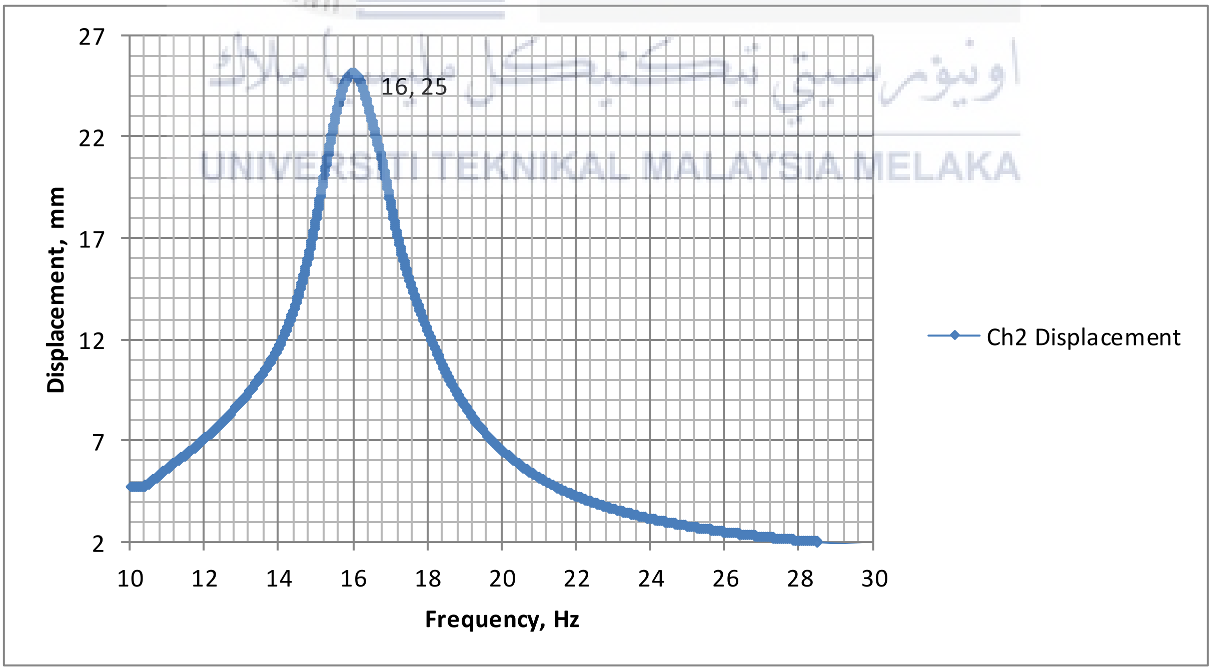


Figure 5.9 Response of the nonlinear absorber with x^4 block curve.

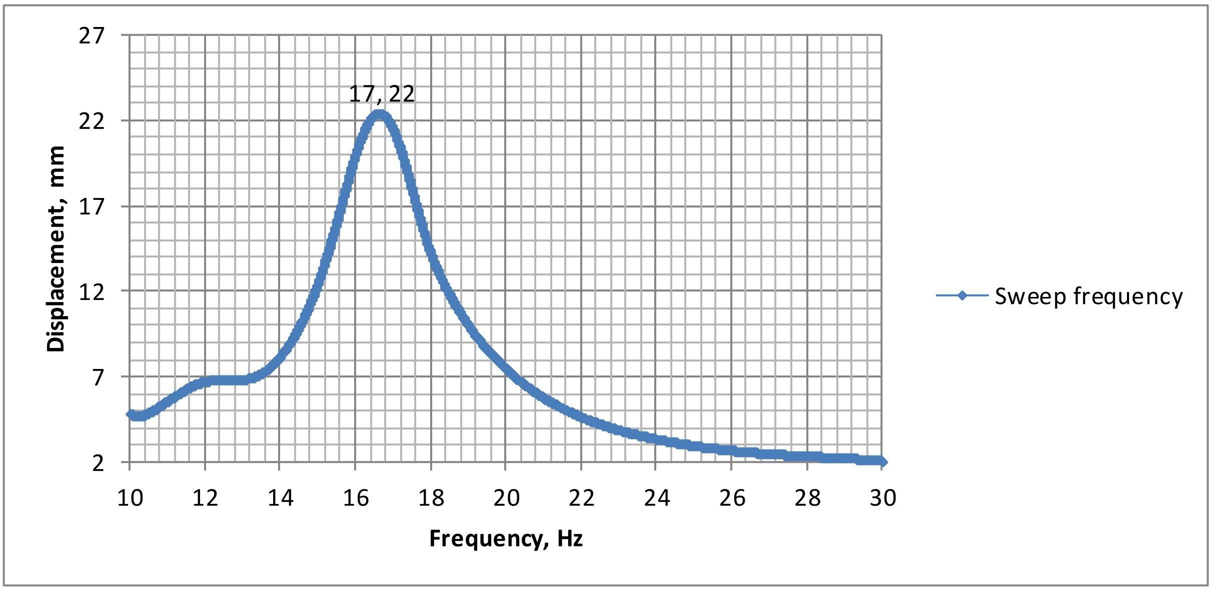


Figure 5.10 Response of the nonlinear absorber with x^6 block curve.

As shown on Figures 5.8, 5.9, and 5.10 there are all showing the linear response of the absorber. All of the three responses of the absorber of the three profile block curves having almost the same natural frequency which showing that the block curve used doesn't have a significant effect to the natural frequency of the absorber. All the response shows a linear responses due to the absorber beam is not constrained by the profile block curve. By increasing the vibration input the response might become nonlinear but it really need a high vibration input which is not suitable for the equipment used in the laboratory. Thus this is why the modification is made to the absorber which is by attaching the stopper on the profile of the block curve so as to constrained the deflection of the absorber beam.

5.2 Impact Hammer Test on Primary Structure Beam

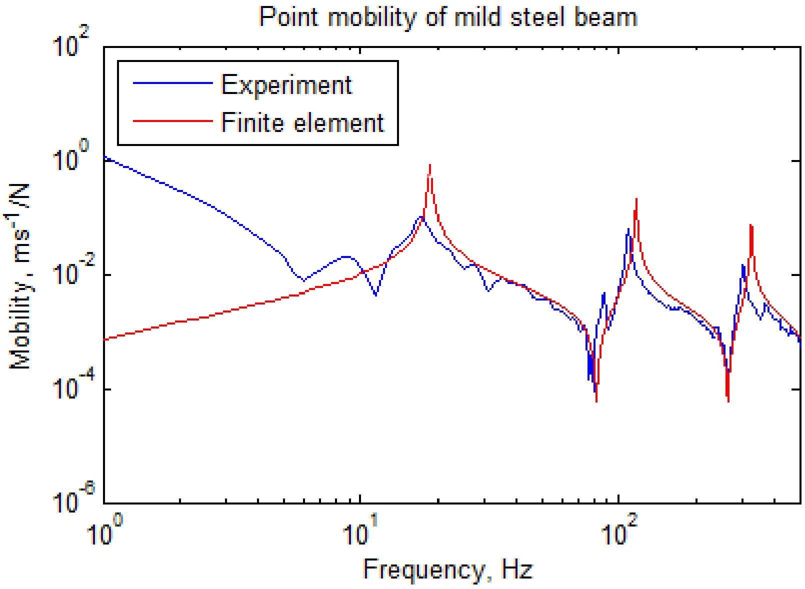


Figure 5.11 Response of primary structure beam on impact hammer test.

As shown on Figure 5.11 is a result of an impact test on a primary structure beam. The response with the blue colour represent the primary structure beam is compared with the theoretical response and the result in Figure 5.11 shows the primary structure impact test response match with the theoretical response. Thus, the impact test was validated.

5.3 ODS Results on Primary Structure

In this case a total number of 18 points were located for measurement. It is important to properly mark up and label all the points according to a fixed logical sequence to avoid confusion while taking measurement so that no points are missed out. The more the number of points, the more detail the ODS is created.

After all of the magnitude and phase of vibration on each 18 points is obtained for the first mode of vibration of the beam, then by using the VibShape software the beam deflection at each modes can be plot. When the results are simulated, operational deflection shape like Figure 5.12 will be obtained for first mode of vibration.

Figure 5.12 ODS of the primary structure beam.

From the ODS simulated from the primary structure beam, we can see that there is maximum deflection at the free end of the beam which is the end that is not been clamped. This free end will be the best position for the absorber's best performance. However, to validate it, three positions on the primary structure beam is attached with the absorber one by one to find the best position for the best performance of the absorber. These three positions is at the clamped end (nodal point), at the middle of the beam, and at the free end of the beam.

5.4 Performance of the Nonlinear Dynamic Vibration Absorber

The natural frequency of the primary structure beam is determined experimentally which is 16 Hz. The absorber used is the absorber with block curve with x^3 profile with length of the absorber beam of 8.1cm and the gap between the stopper and the absorber beam is 1mm. Figure 5.13, Figure 5.14, and Figure 5.15 below shows the response of the primary structure attached with the nonlinear absorber at the three position on the beam which is at the fix end, middle of the beam, and at the free end.

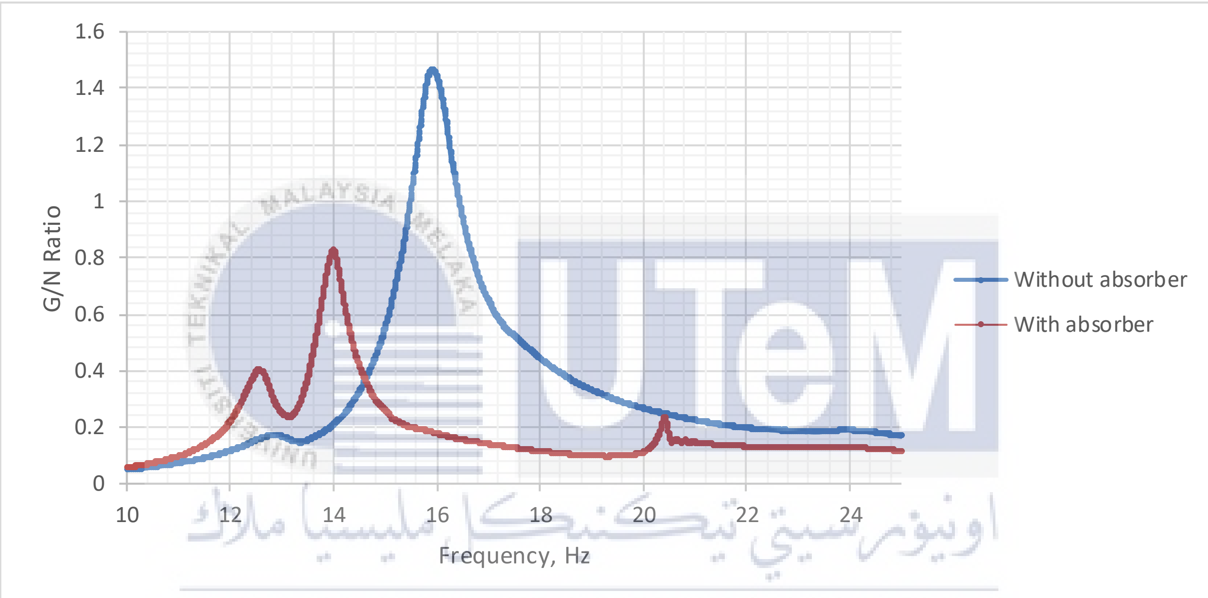


Figure 5.13 Response of the primary structure attached with the nonlinear absorber at the free end of the beam.

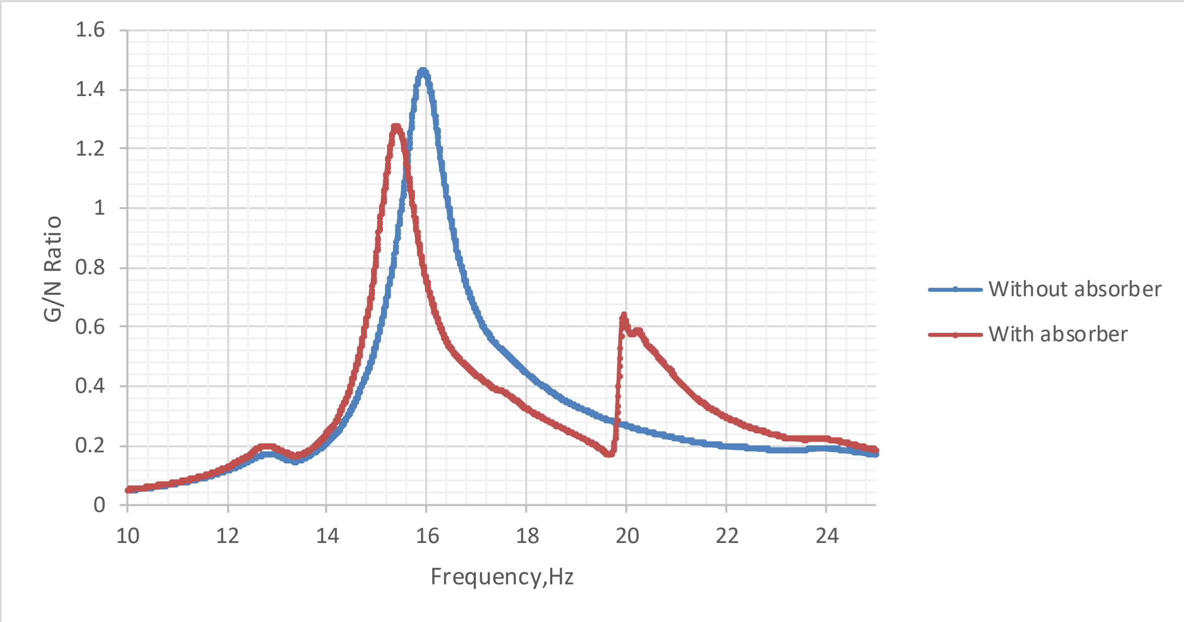


Figure 5.14 Response of the primary structure attached with the nonlinear absorber at the middle of the beam.

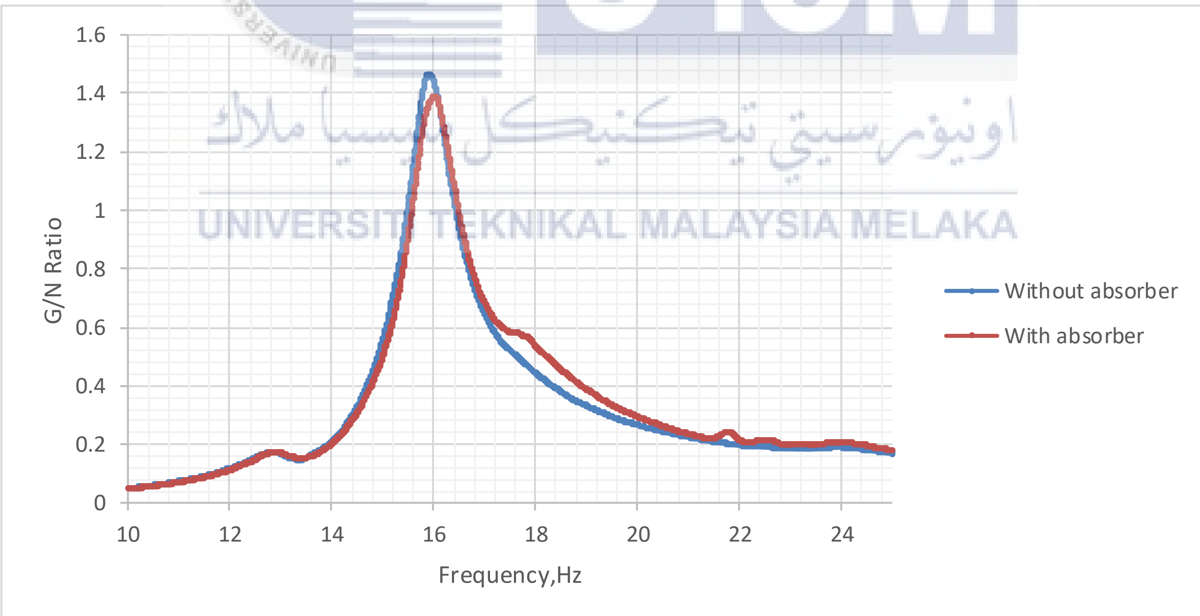


Figure 5.15 Response of the primary structure attached with the nonlinear absorber at the fix end of the beam.

From the Figure 5.12 it shows that the position of the absorber at the free end on the primary structure beam is the most effective position to attach the absorber compared at the fix end and middle position of the beam. It is because as shown in Figure 5.13 the suppression of the resonance peak of the primary structure beam after attaching the nonlinear absorber is bigger compared to when the nonlinear absorber is attached at the middle position of the primary structure beam as shown in Figure 5.14. While there is no suppression of resonance peak occur when the nonlinear absorber is attached at the fix end of the primary structure of the beam as shown in Figure 5.15 due to there is no deflection at that position. Thus, it proves that the effective position for the nonlinear absorber on the primary structure beam is at its free end.

Then, the gap between the stopper and the absorber beam at the effective position on the primary structure is adjusted to a little bit wider into 1.9mm and the Figure 5.16 below shows the effect on the response.

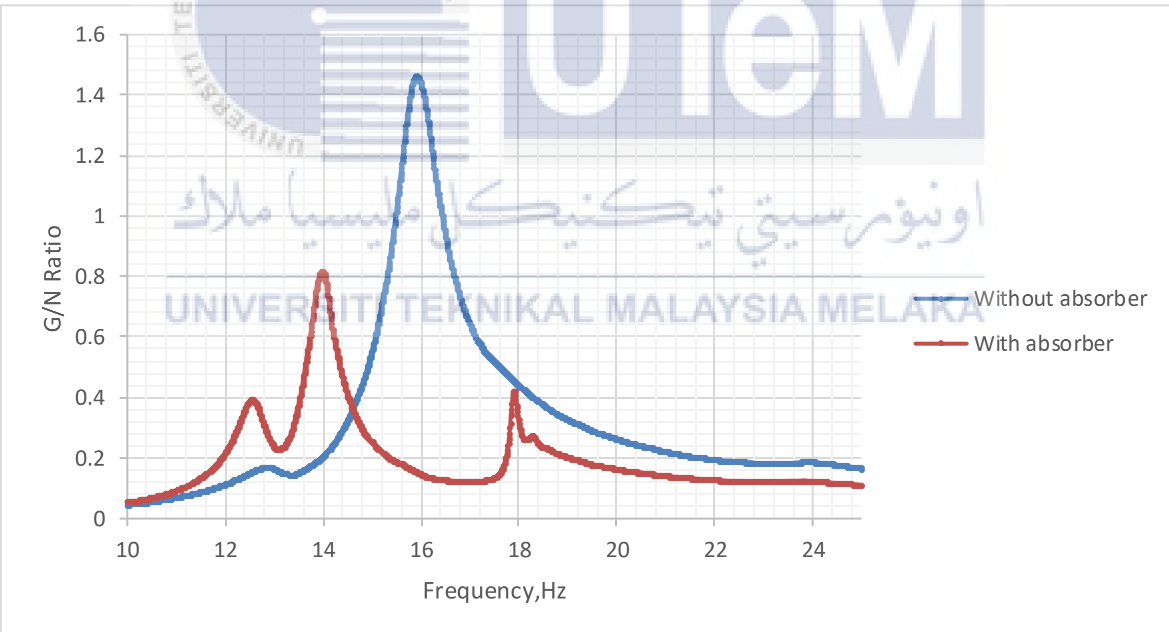


Figure 5.16 Response of the primary structure attached with the nonlinear absorber at the free end of the beam with 1.9mm stopper gap.

From Figure 5.15 which is the used of nonlinear absorber with 1.9mm gap between the stopper and the absorber beam, it shows that the response of the suppressed resonance peak having a narrow safe operating frequency band compared to the response of the application of the nonlinear absorber with 1mm gap between the stopper and absorber beam as shown in Figure 5.12. This proves that the smaller the gap between the stopper and the absorber beam the wider the safe operating frequency band of the primary structure beam.

CHAPTER 6

CONCLUSION AND RECOMMENDATION

6.1 CONCLUSION

The characterization of a wideband Nonlinear Dynamic Vibration Absorber (NLDVA) using curve constrained cantilever beam is conducted to study the ability and performance of a nonlinear absorber or NLDVA in providing a wider safe operating frequency range regarding on supressing an unwanted vibration on a structure which is modelled by a beam. The curved were supposedly providing the nonlinear characteristics to the NLDVA and as the profile of the curve block increase from x^3, x^4, to x^6 the nonlinearity of the NLDVA also will increase. But due to the limitation of the equipment and the design of the NLDVA itself, the profile curve block cannot function as it tends to be and thus the absorber behaved linearly. Modification of a NLDVA by adding a stopper is done in order to harden the deflection of the absorber beam which is what is the profile curve block supposed to do and as a result the absorber behaved nonlinearity. It means the stopper that is attached on the curve block is assisting in hardening the deflection of the absorber beam. As the absorber has behaved the nonlinearity due to the modification that've been made, the other parameter in characterizing the NLDVA is added which is the gap between the stopper and the absorber beam. The result shows that the smaller the gap the more nonlinearity the NLDVA will become. The result of the study conducted shows that the NLDVA designed was able to provide wider safe operating frequency range and it is proves that when the designed NLDVA is put on the vibrating structure which is the performance of the NLDVA is better in suppressing unwanted vibration than a linear absorber in providing a wider band of vibration suppression.

6.2 RECOMMENDATION

In future study with regard to the characterizing of the NLDVA, the investigation can be improve by redesign or used a curve block with a profile that providing a steeper characteristic to constrain the absorber beam. It means that the curve designed should be able to harden the deflection of the absorber beam easier so as to obtain a nonlinear behaviour of the absorber.

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