ACTIVE TUNING OF DYNAMIC VIBRATION ABSORBER (DVA) FOR SUPPRESING STRUCTURAL VIBRATION

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UNIVERSITI TEKNIKAL MALAYSIA MELAKA

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A report submitted
in fulfillment of the requirements for the degree of
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APPROVAL

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

Signature : ……………………………………...

Supervisor's Name : PROF. MADYA DR. ROSZAIDI BIN RAMLAN

Date : ……………………………………...

DECLARATION

I declare that his project report entitled “Active Tuning of Dynamic Vibration Absorber (DVA) for suppressing structural vibration” is the result of my own work except as cited in the references.

Signature : ……………………………

Name : ……………………………

Date : ……………………………

DEDICATION

To my beloved mother and father

ABSTRACT

The dynamic vibration absorber (DVA), also called the tuned vibration absorber (TVA), is commonly used in engineering applications. It can be used to suppress vibrations of structures, either globally for rigid structures or locally for flexible structures. DVAs can be divided into passive DVAs, active DVAs and semi-active DVAs. The passive DVA consists of a mass, a spring, and a damper. The mechanical simplicity of the passive DVA gives it good stability and it can suppress undesired vibrations of primary structures excited by harmonic forces. However, the passive DVA is only effective over a very narrow frequency range. To increase the effective frequency range, there is the invention of active DVA. The active DVA is also called the active–passive vibration absorber. It can be considered as a DVA with an active element attached. By controlling the activation force, the vibration reduction performance of the active DVA can be improved. In this project report, active tuning of dynamic vibration absorber on suppressing the structural vibration was studied. An active dynamic vibration absorber system was constructed. Frequency detection method was discussed. An electric motor was used as the active element to the system. The vibration amplitude of the primary system can reduce effectively when active vibration absorber is added to the system.

ABSTRAK

Dynamic Vibration Absorber (DVA), juga disebut sebagai tuned vibration absorber (TVA), biasanya digunakan dalam aplikasi kejuruteraan. Ia boleh digunakan untuk menekan getaran struktur, baik secara global untuk struktur tegar atau secara tempatan untuk struktur fleksibel. DVA boleh dibahagikan kepada DVA pasif, DVA aktif. DVA pasif terdiri daripada jisim, pegas, dan peredam. Kesederhanaan mekanikal DVA pasif memberikannya kestabilan yang baik dan dapat menekan getaran struktur primer yang tidak diingini oleh daya harmonik. Walau bagaimanapun, DVA pasif hanya berkesan pada julat frekuensi yang sangat sempit. Untuk menyelesai masalah tersebut, terdapat penciptaan DVA aktif. DVA aktif juga dipanggil active–passive vibration absorber. Ia boleh dianggap sebagai DVA dengan elemen aktif terpasang. Dengan mengawal daya pengaktifan, prestasi pengurangan getaran DVA aktif dapat ditingkatkan. Dalam laporan projek ini, penalaan aktif penyerap getaran dinamik untuk menekan getaran struktur telah dikaji. Sistem active dynamic vibration absorber telah dibina. Kaedah pengesanan frekuensi telah dibincangkan. Motor elektrik digunakan sebagai elemen aktif kepada sistem Amplitud getaran sistem primer dapat berkurang dengan berkesan apabila penyerap getaran aktif ditambahkan kepada sistem..

ACKNOWLEDGEMENT

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Firstly, I would like to thank and give millions of appreciations to my PSM supervisor, Prof. Madya Dr. Roszaidi Bin Ramlan who has guided me and spent a lot of time in assisting me with advices and encouragement along the way of completing this PSM report. Furthermore, I would like to thank my second reviewer and seminar panel 1, Prof. Madya Ir. Dr. Md. Fahmi Bin Abd. Samad @ Mahmood and seminar panel 2, Prof Madya Dr. Azma Putra for always giving comments and good suggestions for my final year project.

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CHAPTER 1

INTRODUCTION

1.1 Background

It is well accepted to be said that earthquakes will not stop to occur in the future, and it will cause critical social structural building and economic damage if we are not prepared. Engineer strategies need to be improved and earthquake risks need to be assessed to diminish the damage that are caused by the vibration from the earthquake. Intellectual such as geologists, seismologist and engineers are doing their best to improve the earthquake zoning maps, create reliable databases of earthquake processes and the consequences of it, increase on the understanding of the earthquake area characteristic and design the earthquake resistant buildings. Moreover, the alternative yet effective way to reduce the vibration from the earthquake is to install dynamic vibration absorber to the structure or system.

Dynamic vibration absorber (DVA), also called as tuned vibration absorber (TVA) is invented by Frahm to suppress the vibration from a system. It consists of spring-mass system, where its natural frequency is tuned to the vibration frequency of the host structure in order to reduce the vibration of the structure.

In recent research, several improvements to the traditional DVA are to be studied and mainly relate to the adjustability of DVA in order to increase the narrow effective frequency area. DVA will loses its efficiency on reducing vibration when the resonant frequency is out of the effective bandwidth. Worse come to worst, the DVA will increases the vibration to the structure and cause more damage. The improvements to the traditional DVA are introduced over the last 100 years, ranging from passive, semi-active, hybrid or adaptive to active dynamic systems.

In this paper, active tuning of dynamic vibration absorber on suppressing the structural vibration is to be studied. An actuator is attached to the structure and it will be controlled by microcontroller. The actuation force from the actuator need to be matched with the natural frequency of the primary in order to reduce the vibration.

1.2 Problem Statement

Vibration amplitude of a system can be reduced by applying vibration isolation or vibration absorber to the system. Vibration isolation is a control scheme taken to minimize the amount of force transmitted to the system. In another words, the purpose of the vibration isolator is to block the vibration energy being transfer to the system. While vibration absorber is to absorb the vibration energy of the source or directly from the target receiver. It is in the form of another vibrating system and it is attached on the target structure to absorb the vibration energy. For a forced vibration, the frequency of the attached vibration absorber needs to be same with the excitation frequency of the system in order to reduce the vibration amplitude. Installing a vibration absorber to a single degree of freedom (SDOF) system will changes the system to 2-DOF system. Thus, the system will have two new natural frequencies. If the operating speed of the system or machine is not stable, either reduce or increase, the system will suffer resonance. In another word, the effective bandwidth of the traditional vibration absorber is narrow. Therefore, the purpose of this research is to widen the narrow frequency range of DVA by active tuning system.

1.3 Objectives

The objectives of this project are as follows:

1. To design a frequency detection system.

2. To design and implement a system to control the speed of actuator.

3. To conduct the performance testing of the DVA on suppress the vibration of the primary structure.

1.4 Scopes of Project

The scopes of this project are:

1. The system is to be assumed as linear system.

2. The data measured from the accelerometer is in term of frequency spectrum and time domain waveform

3. The primary system is a rotating motor and to be assumed there is no vibration from the structure.

CHAPTER 2

LITERATURE REVIEW

2.1 Vibration Phenomenon

Structural vibration is a serious threat to our environment nowadays. According to Hartog (1985), Vibration is basically a periodic motion and this motion will be repeating even after certain of time interval, see in Figure 2.1. In most of the cases, the presence of the dynamic load is the most important and common that contributed to the vibration of structures. Human activity is a common contributor of the dynamic loading and the structures that normally be affected are stadiums, staircases, bridges, floors and any other spaces that can be occupied. (Pearce and Cross, 2011). Moreover, other sources of vibration include compressors, pumps, and engines and these equipment will generate dynamic forces thus result in structural vibration. These generated dynamic forces are then led to bigger problems such as piping failures, secondary damages, poor equipment reliability, and safety concerns. The vibration is caused by the mechanical resonant in the structure. The resonant occurs when the frequency of the dynamics forces as mentioned above coincide with the building natural. At resonance, the vibrating forces can be amplified up to 20 times and this will cause the whole structure or deck beams to vibrate beyond its safety operating limits. To solve the problem above, a suitable approach to overcome the vibration is to add a dynamic vibration absorber to the structure.

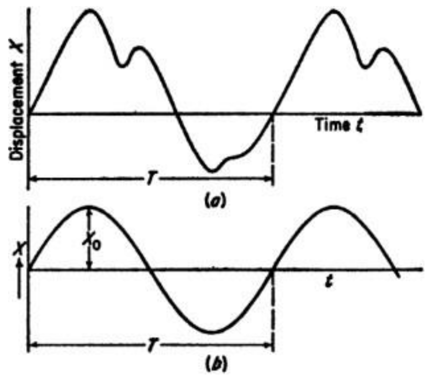

Figure 2. 1 A periodic and a harmonic function (Hartog, 1985)

2.2 Vibration Isolator

Vibration isolation is a technique commonly used in structures and machines to reduce or suppress unwanted vibrations. By inserting a resilient member or isolator, the device or system of interest is isolated from the source of the vibration (Yang, Li and Cheng, 2011). Valeev & Kharisov (2016) studied about the application of vibration isolators with a low Stiffness for the strongly vibrating equipment. The design of their vibration isolator is shown in figure 2.2. The isolator was installed between the vibrating equipment and the foundation. The steel rings are assembled to the frame of the equipment by welding to allow the isolators to hold the frame with stable position as shown in figure 2.3.

Figure 2. 2 The design of Vibration Isolator by Valeev and Kharisov (2016)

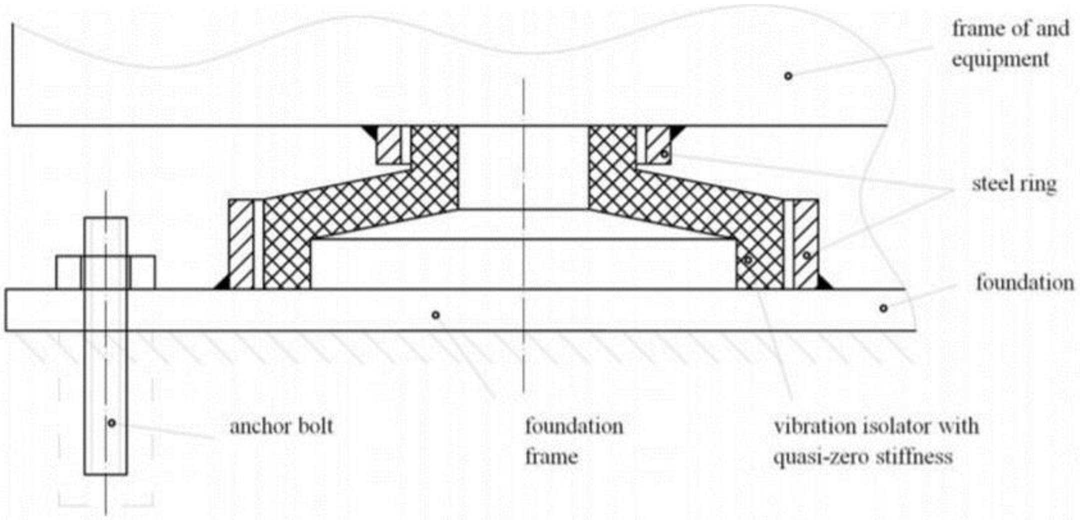

Figure 2. 3 The scheme of installation of vibration isolator (Valeev and Kharisov, 2016)

2.3 Vibration Absorber

Frahm (1911) invented the theory of undamped and damped dynamic vibration absorber (DVA). this device is also called as tuned vibration absorber (TVA). DVA consists of a mass, a damper, and a spring, and it is connected to the primary system to absorb the vibration of the primary system, see in Fig. 2.4. DVA need to be attached at the location where the maximum vibration in the primary system is acting on. The normalized amplitude of the primary system (m) can be calculated by, see in Equation 1.

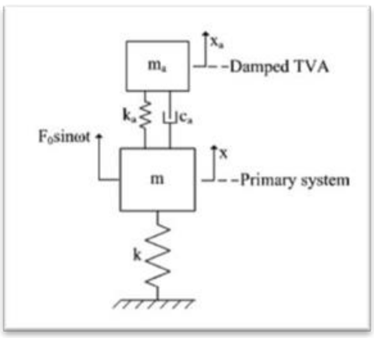

Figure 2. 4 Primary system (m) with damped tuned vibration absorber (ma) (Franchek, Ryan and Bernhard, 1996)

$$\frac{xk}{F_0} = \sqrt{\frac{(2\zeta r)^2 + (\beta^2 - r^2)^2}{(1-(1+\mu)r^2)^2(2\zeta r)^2 + ((1-r^2)(\beta^2 - r^2) - \mu\beta^2 r^2)^2}} \tag{1}$$

$$\omega_p = \sqrt{\frac{k}{m}} \qquad \omega_a = \sqrt{\frac{k_a}{m_a}} \qquad \zeta = \frac{c_a}{2m_a\omega_p} \qquad r = \frac{\omega}{\omega_p} \qquad \beta = \frac{\omega_a}{\omega_p} \qquad \mu = \frac{m_a}{m_p}$$

As noted from Equation 1, the natural frequency of the primary system, m_p, can be altered by changing the stiffness (k) or mass (m) of the primary system. The natural frequency of the primary can also be altered by adding a DVA to the system and to change its stiffness (k_a) or mass (m_a) so that the natural frequency of the whole system will be different. Naturally, the stiffness or mass of the primary system is difficult to change in the existing machine, whereas the absorber is simple enough to install to the primary system, and its stiffness can be modified so that the natural frequency of the whole system can be changed. The main purpose for adjustability of DVAs is to increase the small effective frequency area of DVAs, see in figure 2.5. The narrow effective frequency area will cause DVA loses its efficient damping characteristic easily and it can increase vibrations of the primary system and produce more damage to the system. Therefore, a lot of studies have been carried out intensively to widen the narrow frequency area of DVA or to alter the resonant frequency of the DVA as the operating conditions or parameters of the primary system may vary over time (Liu and Liu, 2005). Adjustable DVAs are also applicable to machines that operate close to the natural frequency of the primary system or machines, and it must operate without disturbing the vibrations over a wide frequency range.

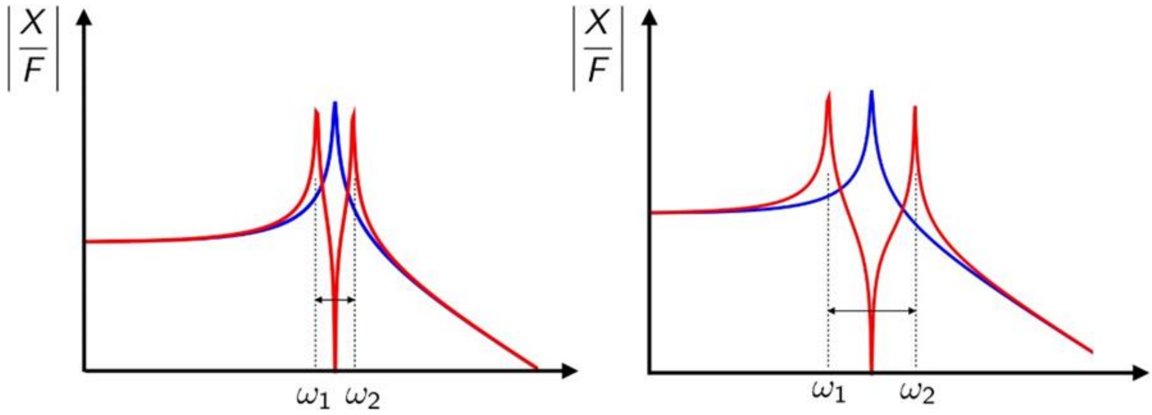

Figure 2. 5 The effective frequency area of DVAs

The mechanism of suppressing structural vibrations by attaching a DVA to the structure is to transfer the vibration energy of the structure to the DVA and to dissipate the energy in the damper of the DVA. In other words, the frequency of the damper is tuned to a structural frequency so that when that frequency is excited, the DVA will resonate out of phase with the structural motion. The ultimate performance of the DVA system is limited by the size of the additional mass, where is typically 0.25~1.0% of the building's weight in the fundamental mode.

2.4 Hybrid, Semi-Active, Passive & Active Vibration Absorbers

DVA or TVA is an over 100-year-old invention, and its adjustability has also been extensively studied by many researchers. DVAs are traditionally categorized into four classes: passive, active, hybrid, and semi-active absorbers. Nowadays, smart materials and adjustable dampers have confused the traditional classification. Resistive or reactive devices, which absorb vibration energy or load the transmission path of the harmful vibrations, are passive absorbers. Active vibration absorbers feed additional energy to the system to absorb its harmful vibrations, and at least a part of the actuator's force is used directly to damp external force. Hybrid absorbers are a combination of active and passive absorbers, and semi-active absorbers are passive absorbers, which are adjusted to optimize their effectiveness. The control parameters of the adjustable DVA includes of adjustment of different type of flexible elements: the effective coil numbers of a mechanical spring (Franchek, Ryan and Bernhard, 1996), the length of threaded flexible rods by stepping motors (Hill and Snyder, 2002), the shape of a flexible beam (Kidner and Brennan, 2001), the curvature of two parallel beams (Bonello et al., 2005), and the effective length of a flexible beam by a moving support (Brennan, 2006).

2.4.1 Hybrid Vibration Absorber

A hybrid vibration absorber is formed when an additional actuation force is added to the DVA. Its actuation force determines the performance of the hybrid vibration absorber (Yuan, 2001). Generally, a simple hybrid vibration absorber absorbs vibration at a certain frequency. Yuan (2001) widened the frequency range by inserting n pieces of simple hybrid vibration absorbers to the system. Thus, it absorbs vibrations at a specific frequency range. Simple hybrid vibration absorber represents a hybrid vibration absorber, which consumes sensor signals directly without recovering modal states (Yuan, 2001). This simple hybrid vibration absorber normally absorbs vibration at a single point; therefore, a simulation had been carried out by Yuan (2011), adding n pieces of HVAs to a cantilever beam, see in figure 2.6. The stimulation result in figure 2.7 shown that the vibration is absorbed by the HVAs n wide frequency range.

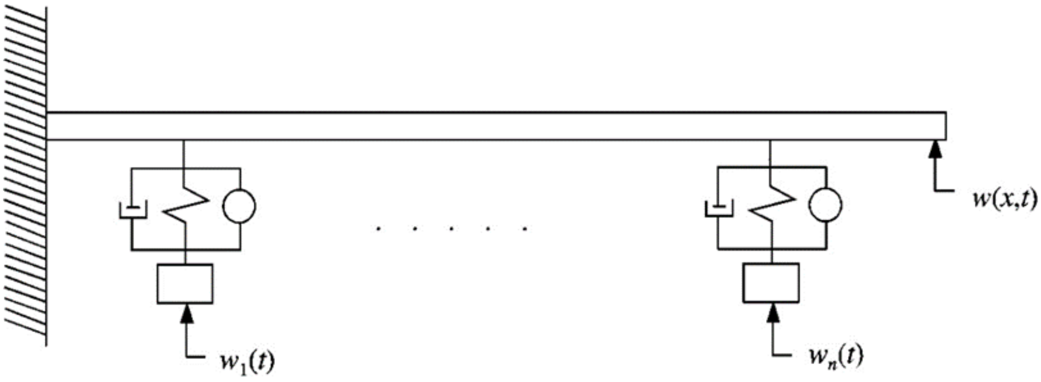

Figure 2. 6 A cantilever beam coupled with n HVAs

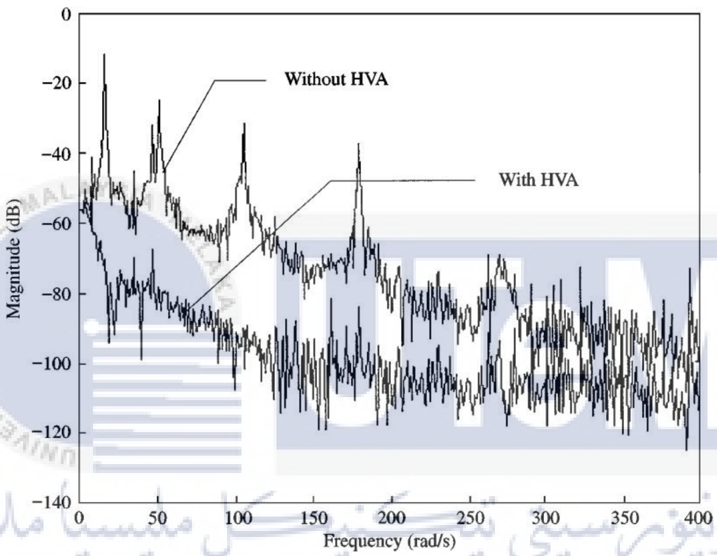

Figure 2. 7 Stimulation Result of proposed HVA by Yuan (2011)

2.4.2 Semi-Active Tuned Vibration Absorber

It has been reported that ATMDs can provide better suppression of structural vibrations than PTMDs. However, ATMDs have the disadvantages of added complexity, high operational and maintenance costs, and high-power requirements. Hence, they are considered less reliable than passive systems, limiting implementation of two special certain cases. (Setareh, 2001) proposed a semi-active tuned mass damper which is called ground hook tuned mass dampers (GHTMD), see in figure 2.8. The stiffness or damping of semi-active TVAs can be different, and they vary from original TVAs by an adjustable damper, which is located between primary system and vibration absorber. This is not an ATVA because only damping or level of amplitudes can be controlled, but not the natural frequencies of the system.

However, the damping factor can be different, for instance, with time, so that a vibrating system with GHTMD is a nonlinear dynamic system. (Setareh, 2001) also did numerical studies for the GHTMD. he noted that the primary system must be lightly damped so that the TVA worked effectively.

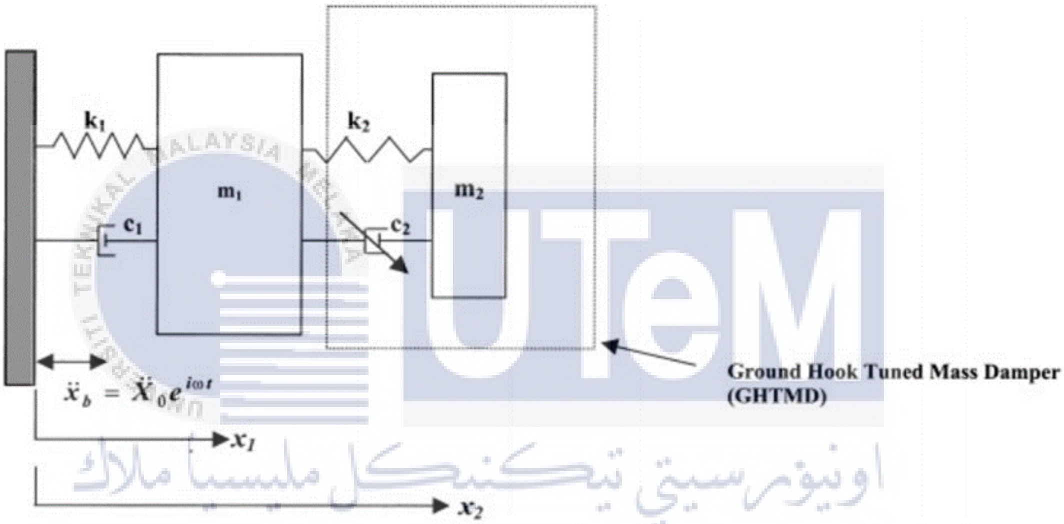

Figure 2. 8 Vibrating system with GHTMD (Setareh, 2001)

2.4.3 Passive Tuned Vibration Absorber

A passive tuned vibration absorber (PTVA) can absorbs the vibrations caused by different types of forces effectively, but the effective tuning area is narrow (Brennan, 2006). Wu & Shao (2007) used a virtual passive approach to widen a narrow frequency range of the DVA. The stiffness, inertia, and damping coefficient of the virtual DVA are easy to adjust by varying the parameters of the virtual element by control algorithms. The virtual element can be a linear actuator or a servomotor. In Wu & Shao's research, the actuator is a moving-magnet voice-coil motor equipped with an LVDT, see in Figure 2.9. They have done numerical simulations for the adaptive undamped and damped vibration absorber and experiments with damped adaptive absorber to obtain a maximum adaptation rate for the virtual stiffness (Wu & Shao, 2007).

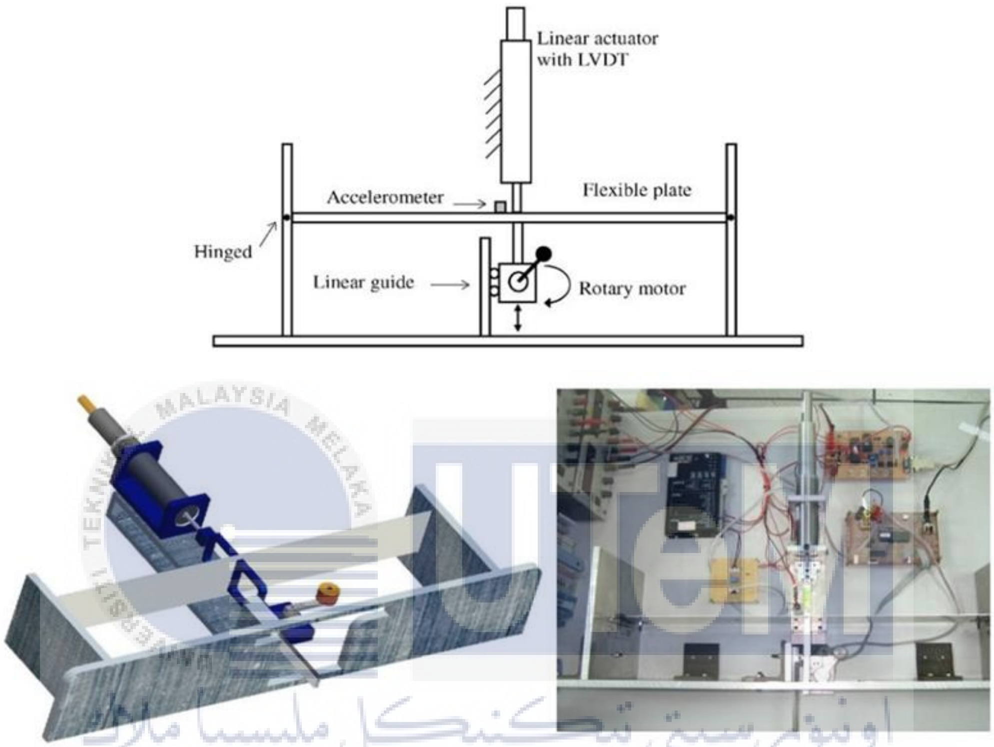

Figure 2. 9 Experimental setup: 3D schematic (bottom left), 2D sketch (top), and the photo (bottom right) (Wu & Shao, 2007)

2.4.4 Active Vibration Absorber

Active dampers are controlled by different kinds of algorithms, and the disadvantage of the active absorbers is their control, which may add vibrations and create more damages if the control fails (Jalili and Esmailzadeh, 2002). In comparison with passive control, active control of structural response is basically characterized by two features which active control requires a certain amount of external power or energy and a decision-making process based on real-time- measured data. In this regard, active control involves a wide range of technologies.

In an active control system, an external source powers control actuator that apply forces to the structure in a specified manner. These forces can be utilized to both dissipate and add energy in the structure to attain desired, optimised response. Actuators of active DVAs can be composed from mechanical mechanisms, pneumatic springs, piezoelectric actuators, electrical linear motors and electromagnetic motors (Sun et al., 2008). Electrical linear motors and electromagnetic motors have its certain benefits such as very responsive and precise, but it is difficult to control these motors and gear is needed for electromagnetic motors to convert rotational motion into translational motion. (Sun et al., 2008) proposed the use of voice coil motor (VCM) to be used as an actuator in active DVA, see in figure 2.10 and figure 2.11. The VCM is a direct drive motor having a permanent magnetic field and coil winding, thus, the produced force is proportional to the coil current. The applied construction was effectively used to damp harmful vibrations.

From the control-engineering point of view, active control systems consist of four inter-connected components or elements. These components are the plant (i.e. building), the sensors, the control computer or controller, and the actuators. Each of these elements works as a subsystem and are mutually integrated such that the output from one component is the input to other components in a closed feedback control loop. Full-scale implementation of active control systems has been accomplished in several research structures mainly in Japan. However, cost effectiveness and potential reliability considerations have limited any widespread or non-research acceptance to date in comparison to passive solutions.

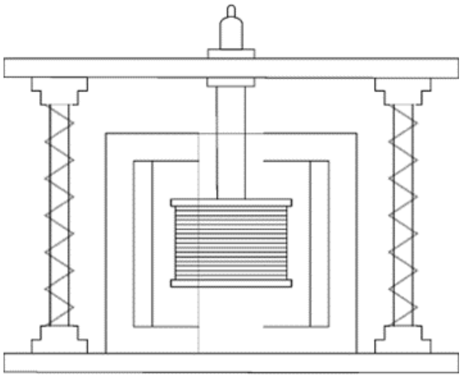

Figure 2. 10 Scheatic of VCM structure

Figure 2. 11 Experimental Platform

CHAPTER 3

METHODOLOGY

3.1 Introduction

This chapter describes the methodology to be used in this project to investigate the active dynamic vibration absorber. This project starts by studying the literature review about concept and the control method of active DVA. Through this study, its help to discover some research that have been done by past researcher that related to active dynamic vibration absorber. Figure 3.1 shows the general workflow of this paper.

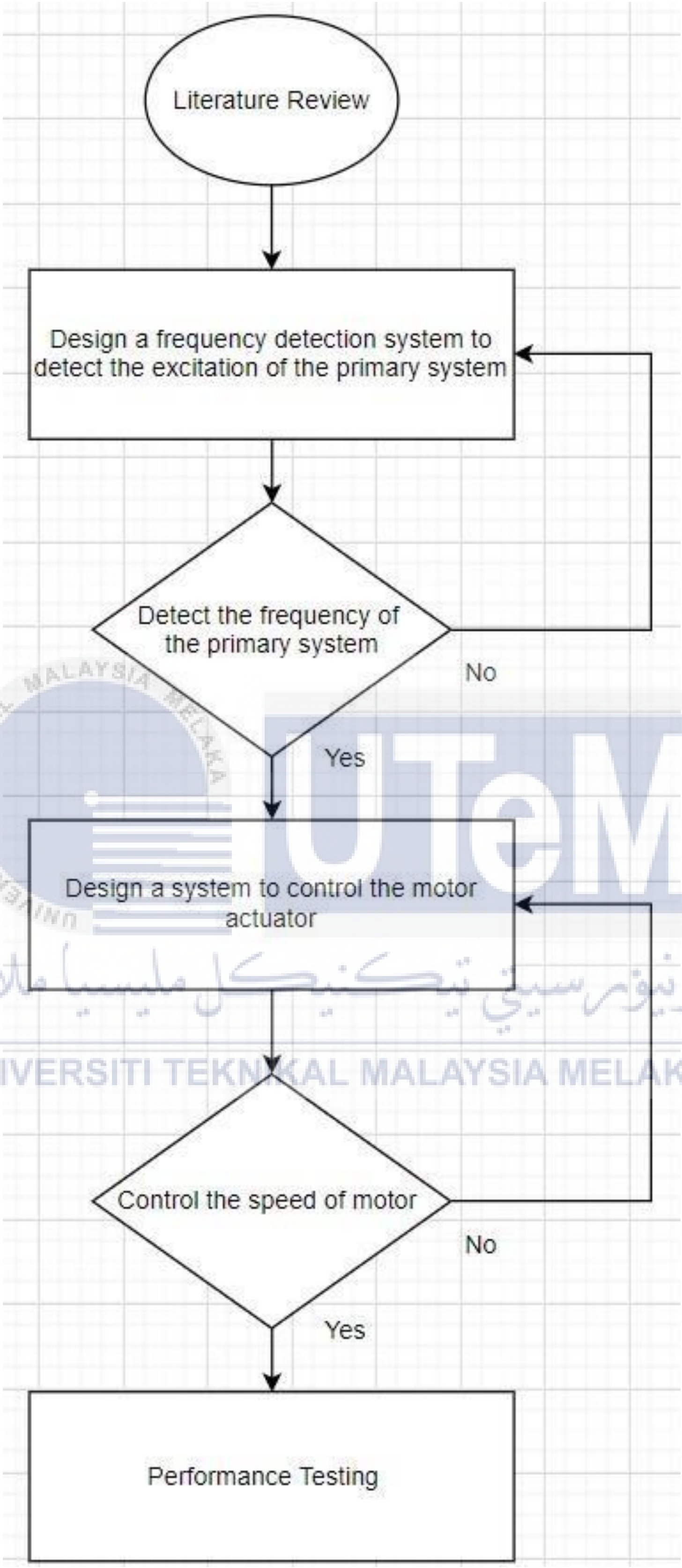

Figure 3. 1 General Workflow

3.2 Frequency Detection System

The first objective of this research paper is to determine the excitation frequency of the primary system. To achieve this objective, a frequency detection system need to be built up. The purpose of this system is to detect the excitation frequency from the primary system which is a rotating motor. In this system, few components are needed, including an accelerometer sensor, piece of microcontroller and a laptop. The experimental set-up is shown in figure 3.2. The motor is modified to increase the vibration effect by adding eccentric forces to it, the modified motor is shown in figure 3.3. The accelerometer sensor is used for measuring the vibration amplitude in term of frequency (Hz). Then, a microcontroller is needed as a medium to process all the data that collected from accelerometer sensor and output the data to the laptop. While laptop is used for compiling the programming coding into the microcontroller by using software LABVIEW. In figure 3.4, the overall workflow of the system is shown. First, the magnitude of the vibration is obtained by the accelerometer sensor and present in time-domain waveform. The data will then be processed by the LABVIEW and Fast Fourier Transform will then be performed to get the frequency spectrum of the system. Before start to measure the vibration of the system, the calibration process of the accelerometer sensor needs to be done. The calibration process will be shown in Chapter 4 along with the block diagram this frequency detection system. The motor driver, powered by the power source, is to control the speed of motor. Accelerometer sensor is attached at the top of the rotating motor to measure the vibration. Then, the obtained data from the accelerometer sensor will then be compared with rated frequency of the motor. Table 3.1 shows the components in this system and their function.

Table 3. 1 Components to be used for frequency detection system

Components	Specifications	Functions

Arduino Uno Microcontroller	• Microcontroller: ATmega328P. • Operating Voltage: 5V. • Input Voltage (recommended): 7-12V. • Input Voltage (limit): 6-20V. • Digital I/O Pins: 14 (of which 6 provide PWM output) • PWM Digital I/O Pins: 6. • Analog Input Pins: 6. • DC Current per I/O Pin: 20 mA.	To process the data that measure by the accelerometer sensor
Accelerometer sensor ADXL335	• On-board 3.3V voltage regulator • Operating Voltage: 2.5V - 6.0V • Typical Current: 300 µA • Range: ±3g • 3-axis sensing • Bandwidth adjustment with a single capacitor per axis	To measure the acceleration of the primary system
L298N Motor Driver Board	• Driver: L298N • Input voltage: +5V~+46V • Input current: 2A • Maximum power: 25W • Dimension: 60mm x 54mm • Weight: 48g • Working temperature: - 25C~+130C	To control the speed of actuator

	• Motor channels: 2	
9V DC Motor	• Rated Voltage: DC 9V • No-load Speed: 1500rpm • Rated current: 0.21A • Weight: 60g	The primary system or act as the source of vibration

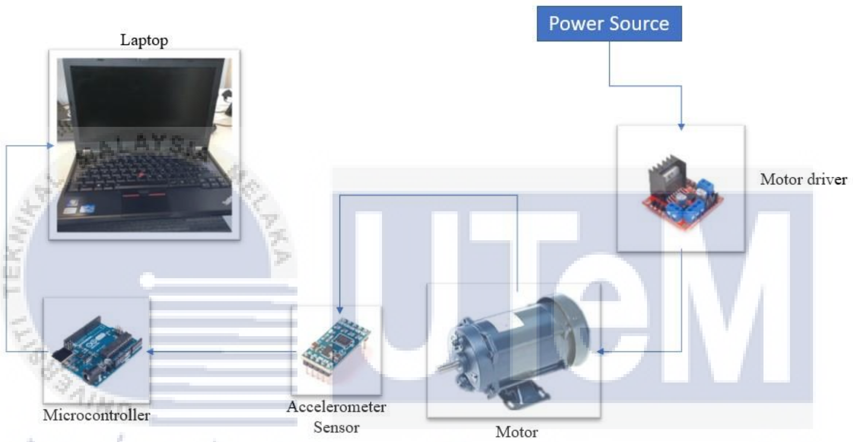

Figure 3. 2 Experimental Set up for frequency detection system

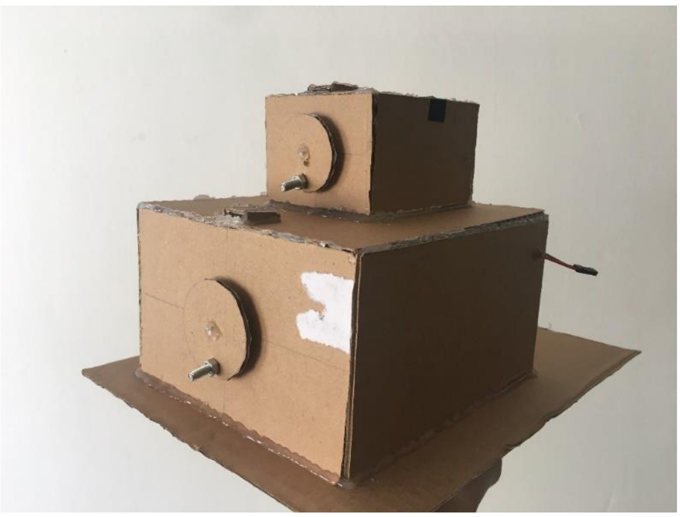

Figure 3. 3 Motor Prototype that used in this experiment

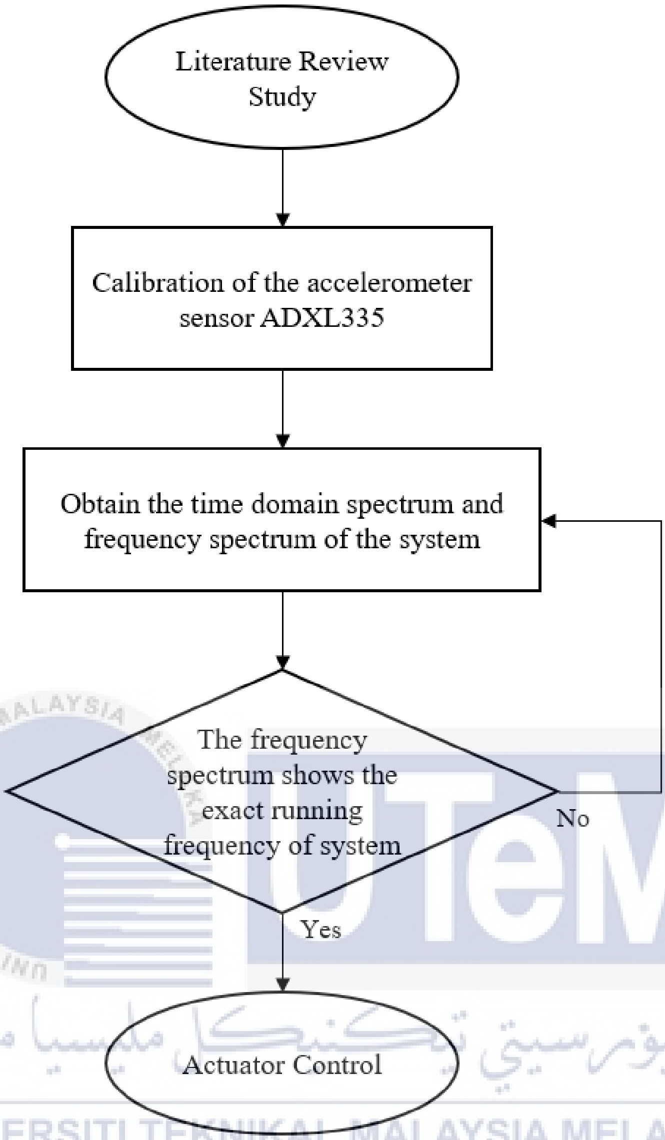

Figure 3. 4 Flow Chart of Frequency Detection System

3.2.1 Calibration of the Accelerometer Sensor ADXL335

The main purpose of the accelerometer sensor is to measure the static acceleration of gravity in tilt-sensing applications as well as dynamic acceleration resulting from motion, shock, or vibration. In this project, accelerometer sensor ADXL335 is used. It is an analog device. This sensor works on 3.3V and its outputs is ratiometric, which means 0g measurement output is nominally equal to half of the 3.3V supply voltage (1.65V), -3g is at 0V and 3g is at 3.3V. The pinout of ADXL335 accelerometer is shown in figure 3.5. VCC is the pin that provides power for the accelerometer which can be connected to 5V on the Arduino. X-out, Y-out and Z-out pin output the analog voltage proportional to acceleration exerted on X-axis, Y-axis, and Z-axis respectively. GND is the ground pin which is connected to the ground of Arduino. The purpose of the calibration process is to convert the acceleration output from voltage to acceleration. The connection of the calibration process is shown in figure 3.6. The calibration process is done by using LABVIEW. Firstly, ADXL335 accelerometer is placed on a flat surface. Next, the voltage reading is recorded. This voltage reading is the base level voltage of the ADXL335 accelerometer. To convert the output into acceleration (0 m/s^2), the output need to subtract the base level voltage and be divided by the sensitivity. The sensitivity of each axis can be found in the datasheet of the ADXL335 accelerometer which is shown in the appendix.

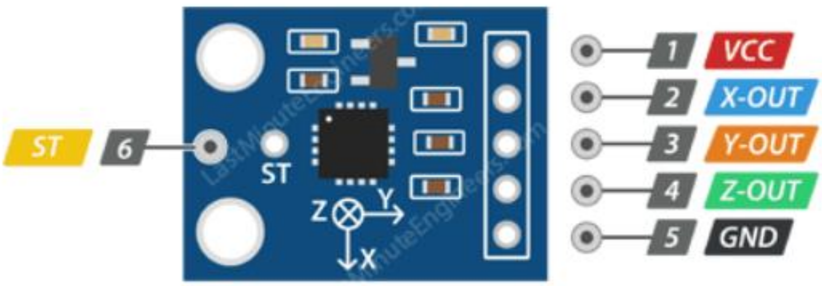

Figure 3. 5 ADXL335 Accelerometer Pinout

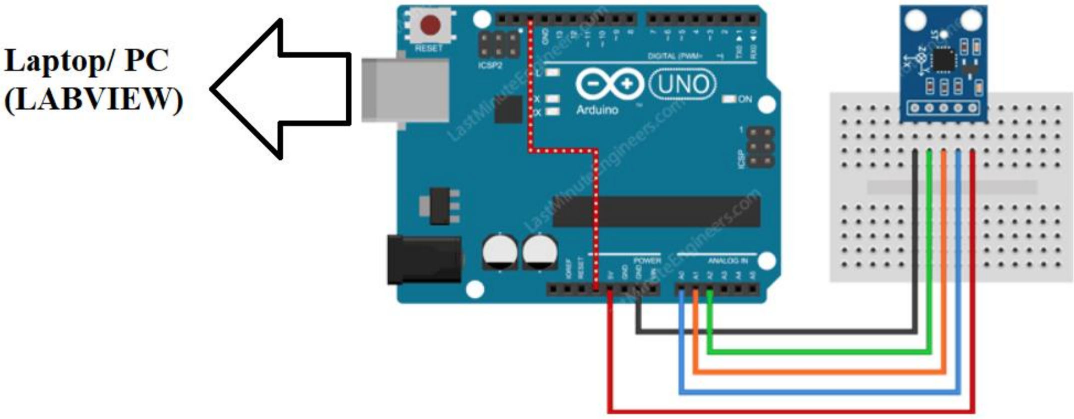

Figure 3. 6 Connection of ADXL335 accelerometer to perform calibration process.

For an example, if the voltage reading for X-axis is 2.3V, assuming the base level voltage for X-axis= 1.65V and 0.33v/g sensitivity for X-axis the acceleration is:

$$\text{Acceleration} = (2.3V - 1.65V)\ x\ \frac{1g}{0.33V}\ x\ 9.81\ \text{m}/s^2 = 19.32\ \text{m/s2}$$

3.3 Actuator Control

The second objective of this paper is to design a control system to control the speed of the motor actuator. The workflow of this system is shown in figure 3.7. In order to achieve that, few components are required in this system, the components are a microcontroller, motor driver and a DC motor. The microcontroller is to control the speed of the DC motor actuator with the motor driver by using Pulse Width Modulation (PWM) method. PWM is a method that using digital means to achieve analog results. A square wave, a signal switched between on and off, is generated by digital control. The on-off pattern will modulate voltages between full on (5 Volts or any desired voltages) and off (0 Volts) by changing the amount of time the signal spends on versus the amount of time the signal spends off. The pulse width is the length of "on time". To get varying analog values, pulse width is to be changed. The pulse width values can be changed from a scale of 0-255, 255 indicates 100% duty cycle (always on) while 127, which is half of 255, indicates 50% of duty cycle as shown in figure 3.8. The speed of the motor actuator can be controlled using the control scheme above. The purpose of the motor actuator is acting as the vibration absorber to the primary system and applying the counterbalance force to suppress the vibration. The components will be connected in the way as shown in figure 3.9. In the figure, the speed of the actuator which is the electric motor will be control by the motor driver, to check the effective of this system, the voltage readings need to be measured to make sure the PWM signal is correctly calculated and transferred to the motor. The circuit design for this system is shown in figure 3.10. Table 3.2 lists out the components in this system and their function.

Table 3. 2 Components to be used for actuator control system

Components	Specifications	Functions

Arduino Uno Microcontroller	• Microcontroller: ATmega328P. • Operating Voltage: 5V. • Input Voltage (recommended): 7-12V. • Input Voltage (limit): 6-20V. • Digital I/O Pins: 14 (of which 6 provide PWM output) • PWM Digital I/O Pins: 6. • Analog Input Pins: 6. • DC Current per I/O Pin: 20 mA.	To process the data that measure by the accelerometer sensor and control the motor driver
Accelerometer ADXL335	• On-board 3.3V voltage regulator • Operating Voltage: 2.5V - 6.0V • Typical Current: 300 µA • Range: ±3g • 3-axis sensing • Bandwidth adjustment with a single capacitor per axis	To measure the acceleration of the primary system
L298N Motor Driver Board	• Driver: L298N • Input voltage: +5V~+46V • Input current: 2A • Maximum power: 25W • Dimension: 60mm x 54mm • Weight: 48g • Working temperature: - 25C~+130C	To control the speed of actuator

	• Motor channels: 2	
9V DC Motor	• Working Voltage: DC 1.5V - 9V • Rated Voltage: DC 9V • No-load Speed: 8400 rpm • Rated current: 0.21A • Weight: 45g	The actuator or the dynamic vibration absorber
Digital Multimeter	• Battery: 9V • AC Voltage: 60mV / 600mV / 6V / 60V / 600V / 750V ± (1.0%+3) • AC Current: 60mA / 600mA / 6A / 10A ± (1.5%+3) • DC Voltage: 60mV / 600mV / 6V / 60V / 600V / 1000V ± (0.5%+3) • DC Current: 60mA / 600mA / 6A / 10A ± (1.2%+3) • Resistance: 600Ω / 6kΩ / 60kΩ / 600kΩ / 6MΩ ± (0.5%+3); 60MΩ ± (1.5%+3) • Size :130 x 65 x 32mm • Weight: 114g	To measure the voltage reading from the DC motor.

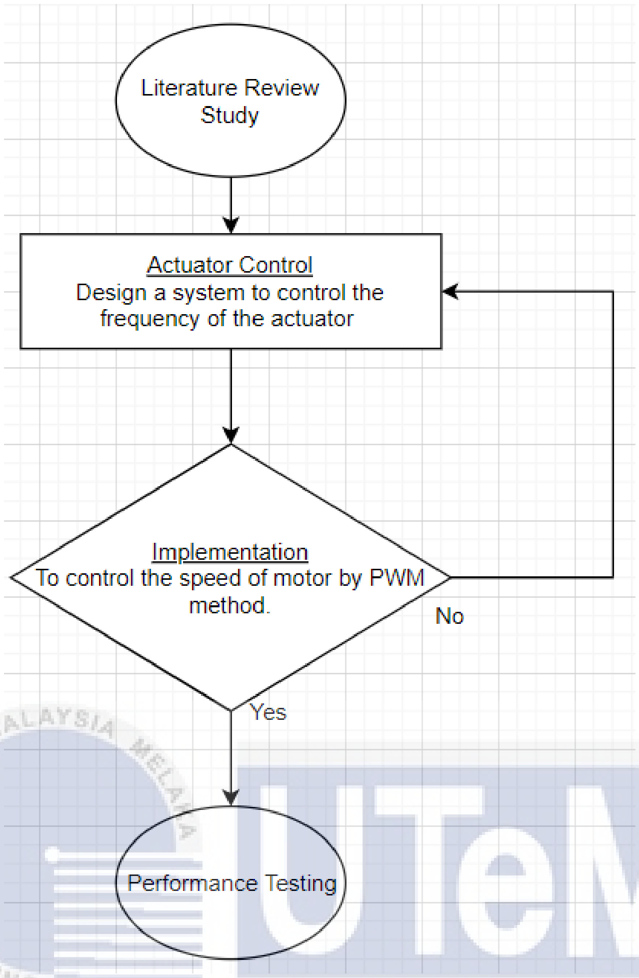

Figure 3. 7 Flowchart for actuator control

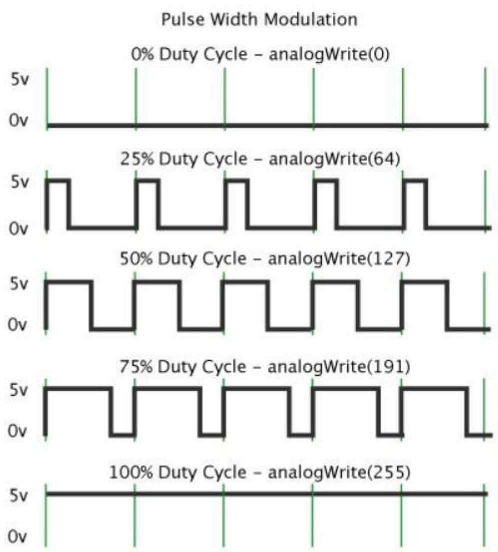

Figure 3. 8 PWM method

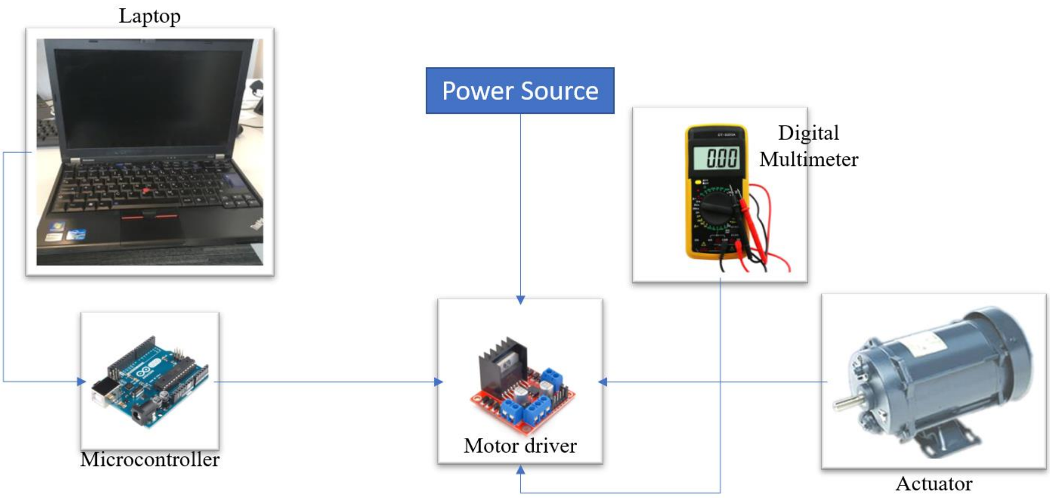

Figure 3. 9 Experimental Set up for actuator control

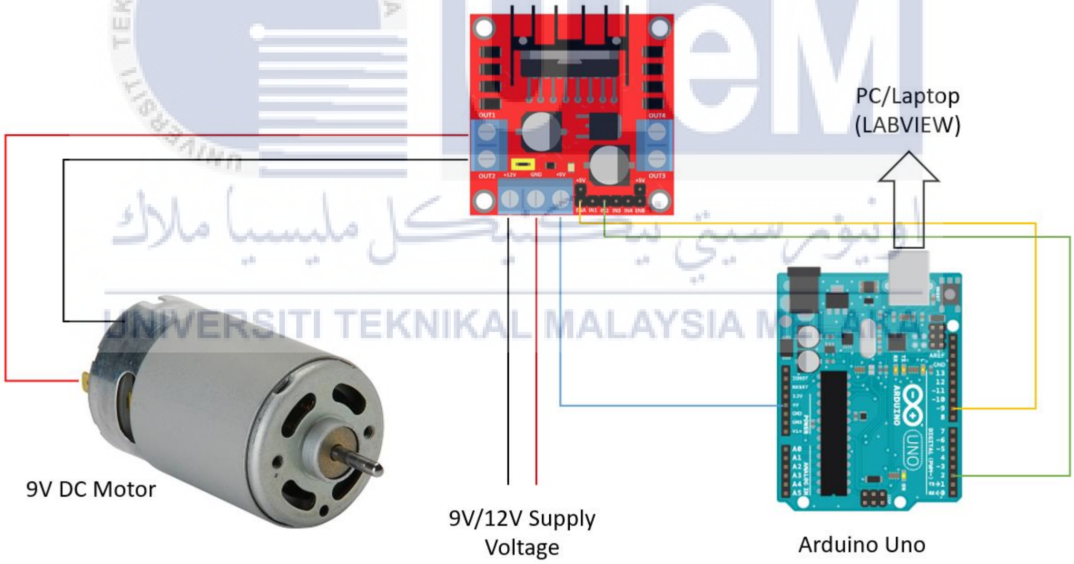

Figure 3. 10 Circuit Connection for Actuator Speed Control

3.4 Performance Testing of the Vibration Absorber

The third objective of this paper is to conduct a performance testing on the DVA on suppressing the structural vibration. The working flow chart is shown in figure 3.11. Frequency detection system and actuator control are to be combined to carry out the performance testing. The experiment setup is shown in figure 3.12. The amplitude of the primary system before adding a vibration absorber and after adding the vibration absorber are to be measured and compared for the proof of concept. Accelerometer sensor is to be used to measure the amplitude of the primary system. The data collected by the accelerometer sensor will be processed by Fast Fourier Transform (FFT) to obtain the critical frequency of the system. After obtaining the frequency of interest, the speed of the actuator is to be tuned by the motor driver to match the key frequency to obtain the frequency domain graph, our microcontroller act as a data acquisition (DAQ) device. It can interface the communications between the accelerometer and computer. The function of DAQ device is to digitize incoming analog signal so that computer can interpret the data. All the data in term time domain and frequency domain will be done by a programmable software, LabVIEW.

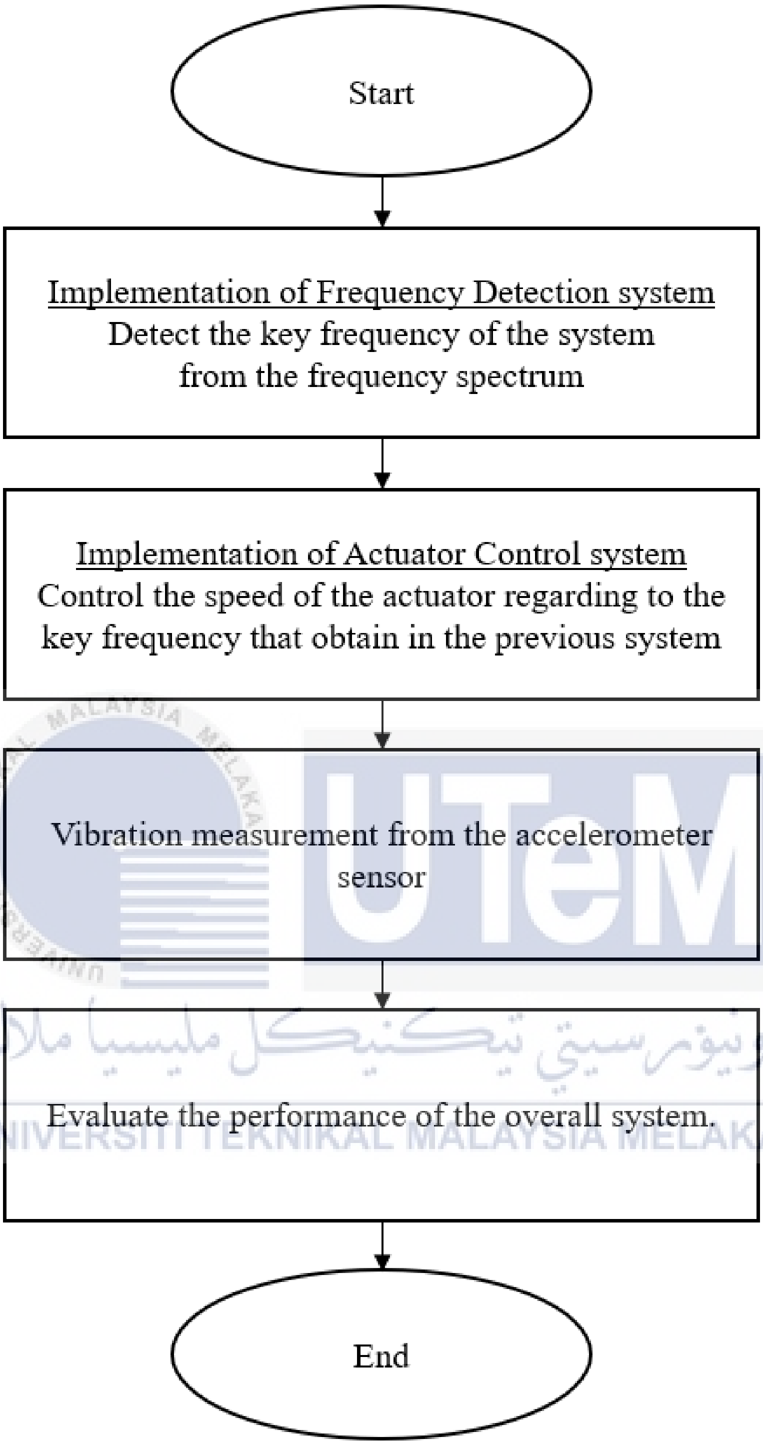

Figure 3. 11 Flowchart for performance testing

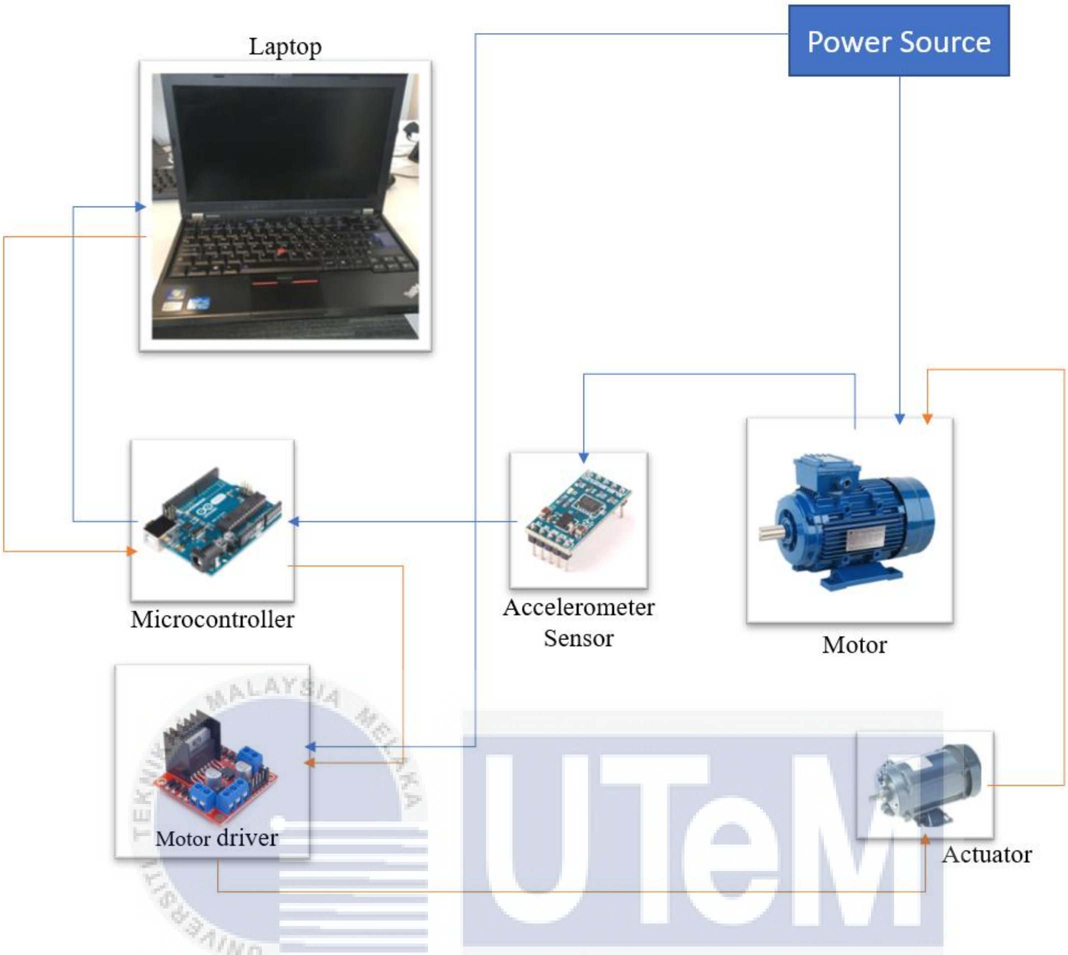

Figure 3. 12 Experimental Set up for performance testing

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Introduction

This chapter discuss on the all the 3 constructed systems from the LABVIEW and the result obtained from these constructed systems. The first system which is the frequency detection system included of calibration of accelerometer sensor ADXL335 and the Fast Fourier Transform process to obtain the frequency spectrum of the system. The next system which is the actuator control system included of speed control of the motor with the use of PWM method. Voltage readings were measured to check the function of PWM method. The performance testing system were constructed to test the performance the vibration absorber which is the actuator. The amplitudes of the vibration were measured to evaluate the effectiveness of the vibration absorber.

4.2 Results

4.2.1 Frequency Detection System

4.2.1.1 Base Level Voltage Reading of Accelerometer Sensor ADXL335

Accelerometer Sensor is the most important component in building up the system because its purpose is to obtain the acceleration information from the system and further to be processed into time domain and frequency spectrum. With most of the sensors, there is some variation in the output. Therefore, calibration was done to minimize the variation in the output and to increase the consistency of the reading. The first process was to obtain the base level voltage of accelerometer sensor. For accelerometer sensor ADXL335, it works on the principle of the Piezoelectric effect. The output of the ADXL335 is in term of Voltage (V), from the specification table in 3.1, it has a 3.3V voltage regulator which means when the ADXL335 sensor stay static on the horizontal surface, according to the datasheet of ADXL335, the reading of the base level voltage should be from the range of 1.35V-1.65V for X-axis and Y-axis, while for the Z-axis, the range should be in 1.2V-1.8V. A system was built by using LABVIEW, the block diagram is shown in figure 4.1, the purpose of this system was to obtain the voltage reading of base level of the accelerometer sensor. The result is shown in figure 4.2 and table 4.1.

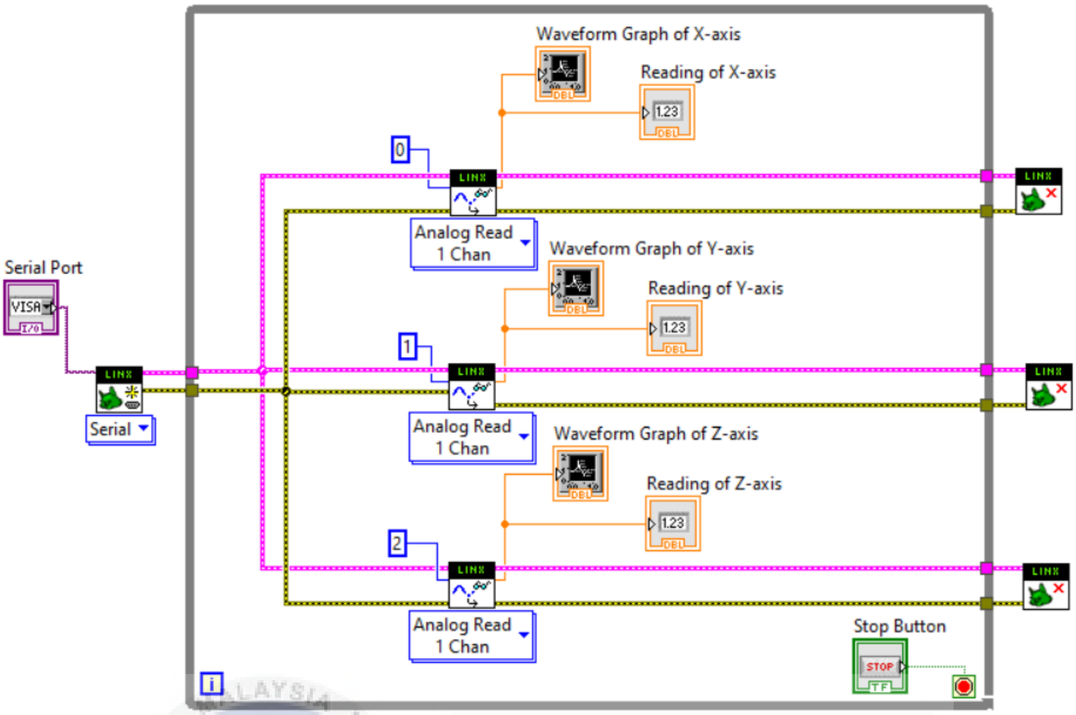

Figure 4. 1 Block diagram of measuring the base level voltage of ADXL335 in LABVIEW

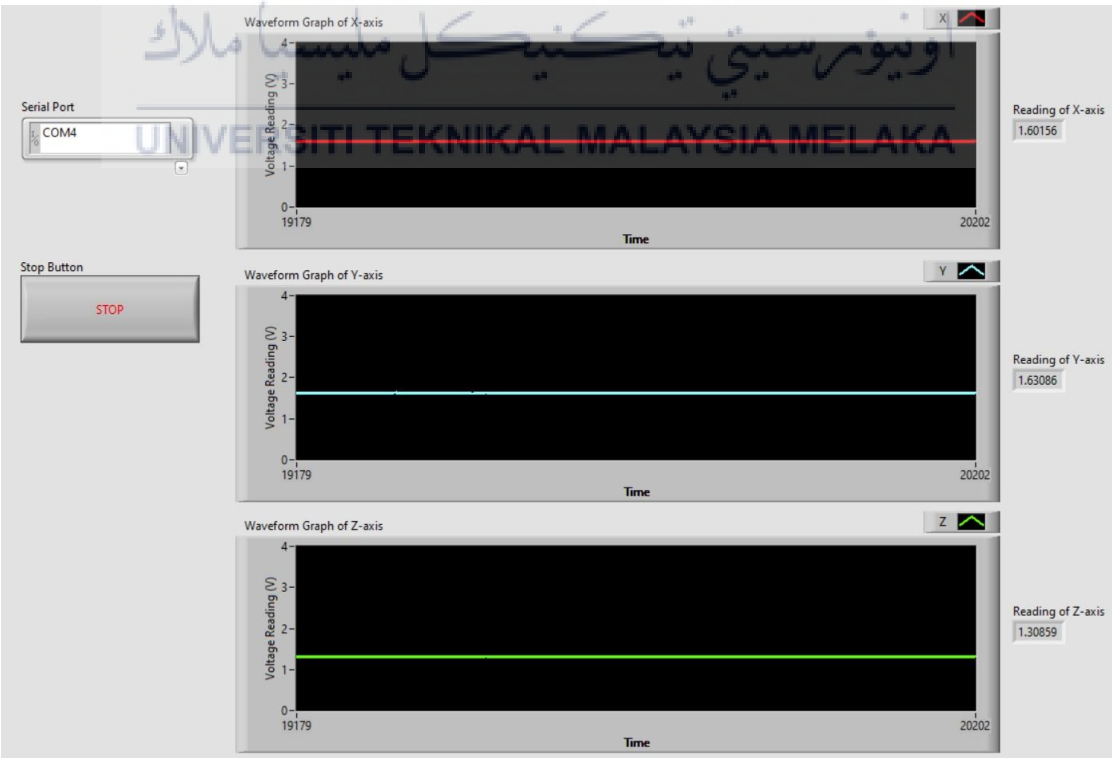

Figure 4. 2 Results of base level voltage of ADXL335 in LABVIEW

Table 4. 1 Results of Base Level Voltage Reading

Axis of Accelerometer Sensor ADXL335	Theoretical Base Level Voltage Reading (V)	Actual Base Level Voltage Reading (V)
X-axis	1.35 - 1.65	1.602
Y-axis	1.35 - 1.65	1.631
Z-axis	1.2 - 1.8	1.309

From the results, the accelerometer sensor gave 1.602V reading for X-axis, 1.631V reading for Y-axis and 1.309V Z-axis. All the readings were fall within the expected range according to the datasheet of ADXL335.

4.2.1.2 Calibration Process of the Accelerometer Sensor ADXL335

The base level voltages of the accelerometer sensor were used to carry out the calibration. The desired output of the accelerometer sensor for this system was in term of acceleration (m/s^2). A calibration system for the sensor was built to convert the output of the sensor from voltage (V) to acceleration (m/s^2). The block diagram of the system is shown in figure 4.3. The new output unit, accelerations (m/s^2) were transform by the subtraction between the current output (V) with the base level voltage (V) and followed by the division with sensitivity of the accelerometer sensor (V/g). The sensitivity of each axis was taken from the datasheet of the accelerometer sensor. The outputs were change from voltage (V) to acceleration in unit of gravitational force (g). To obtain acceleration (m/s^2), multiplication of gravitational acceleration (9.81 m/s^2) was done. The new outputs units were in acceleration (m/s^2). The calibrated results were recorded and shown in figure 4.4 and table 4.2.

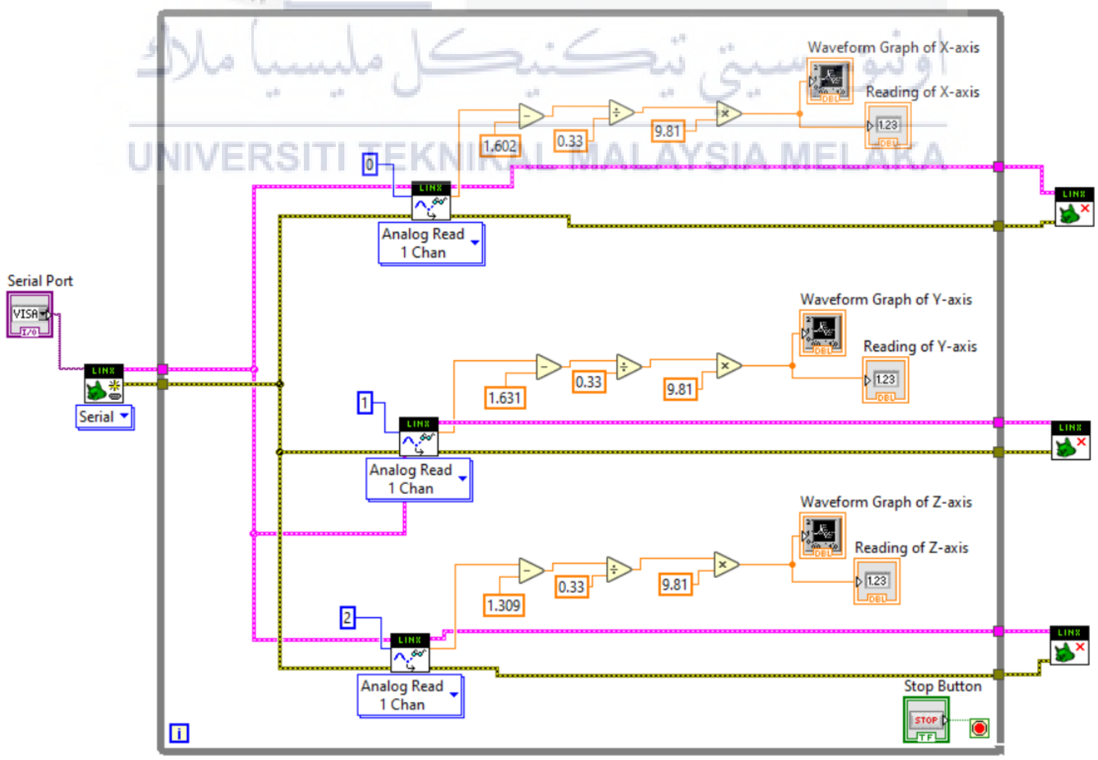

Figure 4. 3 Block diagram of Calibration of Accelerometer Sensor ADXL335

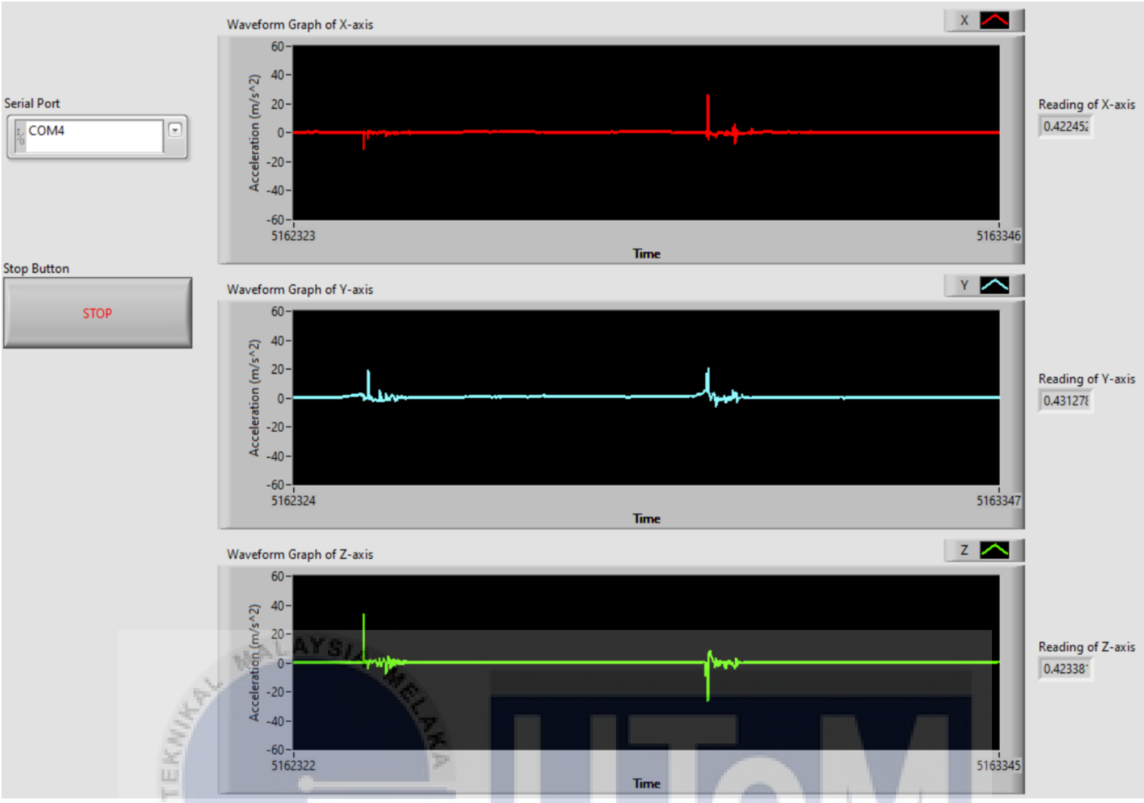

Figure 4. 4 Results of Calibrated Accelerometer Sensor ADXL335

Table 4. 2 Results of Calibrated Accelerometer Sensor ADXL335

Axis of Accelerometer Sensor ADXL335	Base Level Voltage (V)	Sensitivity (V/g)	Theoretical Calibrated Output (m/s^2)	Actual Calibrated Output (m/s^2)
X-axis	1.602	0.33	0	0.422
Y-axis	1.631	0.33	0	0.431
Z-axis	1.309	0.33	0	0.423

After the calibration process, the accelerometer sensor was now giving 0.422 m/s^2 for X-axis, 0.431 m/s^2 for Y axis and 0.423 m/s^2 for Z-axis when the accelerometer stayed in static condition. This calibration process is important to be done before constructing other system because the accelerometer sensor will highly affect the output of the time domain waveform and frequency spectrum.

4.2.1.3 Fast Fourier Transform (FFT)

Calibration of the accelerometer was completed, and it offered the real-time based time domain waveform of a system. However, it only showed the magnitude of the vibration of a system, but it did not give any frequency information of the system which is the most important criteria in constructing a vibration absorber. Therefore, Fast Fourier Transform was added into the system to acquire the frequency spectrum. The block diagram of the system was shown in figure 4.5 while the front panel of the system was shown in figure 4.6. From figure 4.6, there were 3 parameters that can be adjusted for the frequency spectrum, which were the sampling rate or sampling frequency, F_s (Hz), number of samples (N), and types of windowing. The parameters were adjusted based on the Nyquist Theorem, it stated that the waveform must be sampled at a rate greater than twice the input signal. The frequency range, F_{max} was set at 256 Hz. While the number of samples determined the bandwidth or resolution in the spectrum. Hanning window was used in the system. Figure 4.7 showed the frequency spectrum of the motor at its maximum voltage which is 9V. While figure 4.8 demonstrated the frequency spectrum of the actuator at its maximum voltage (9V). The comparison of the measured frequency and the rated frequency was showed in table 4.3.

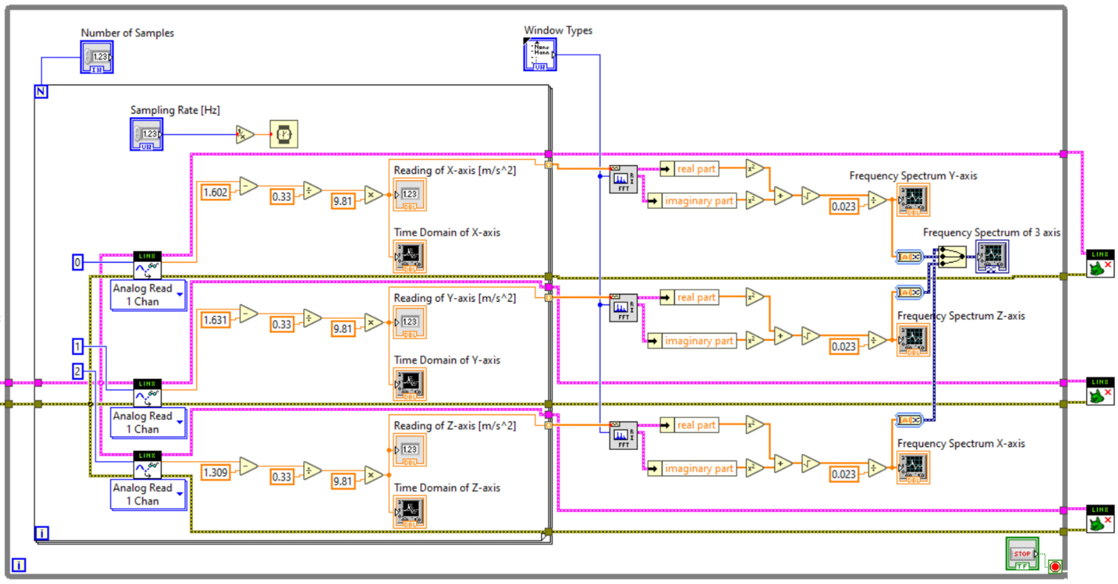

Figure 4. 5 Block diagram of the frequency detection system (Fast Fourier Transform)

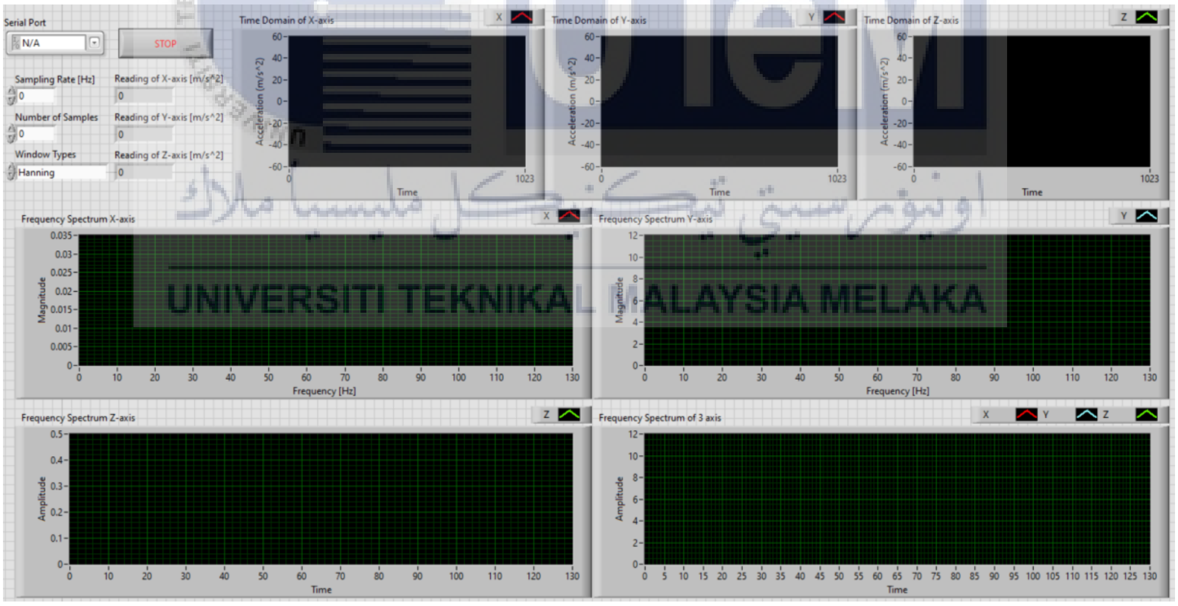

Figure 4. 6 Front Panel of frequency detection system (Fast Fourier Transform)

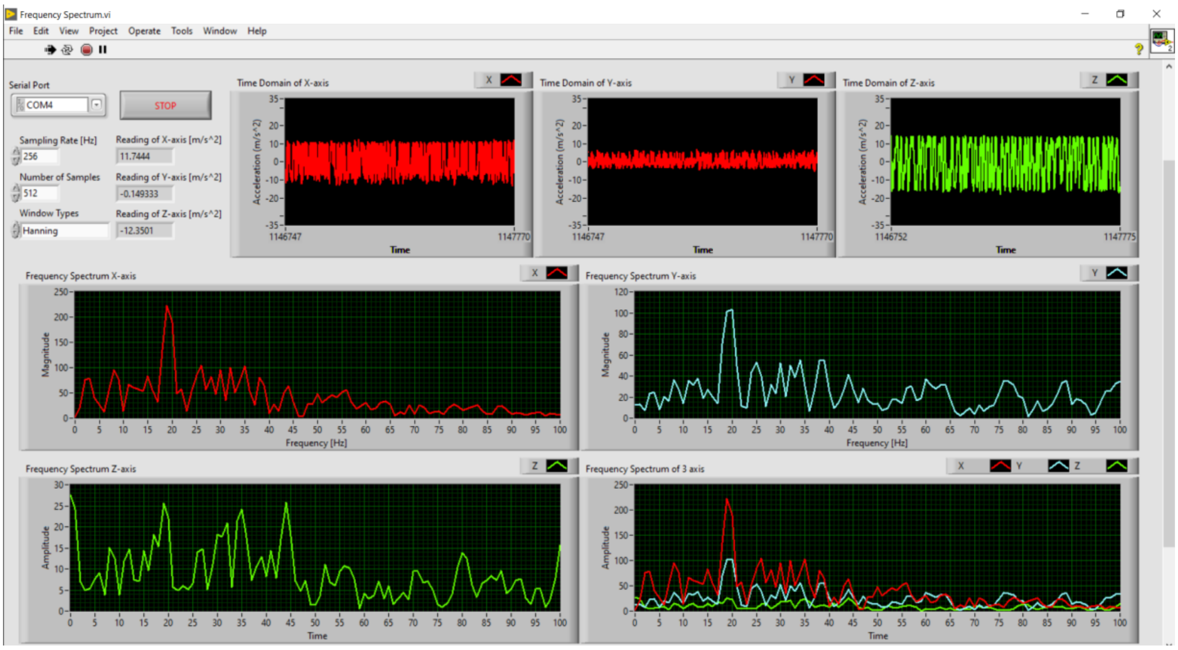

Figure 4. 7 Time Waveform and Frequency Spectrum of Motor at maximum voltage

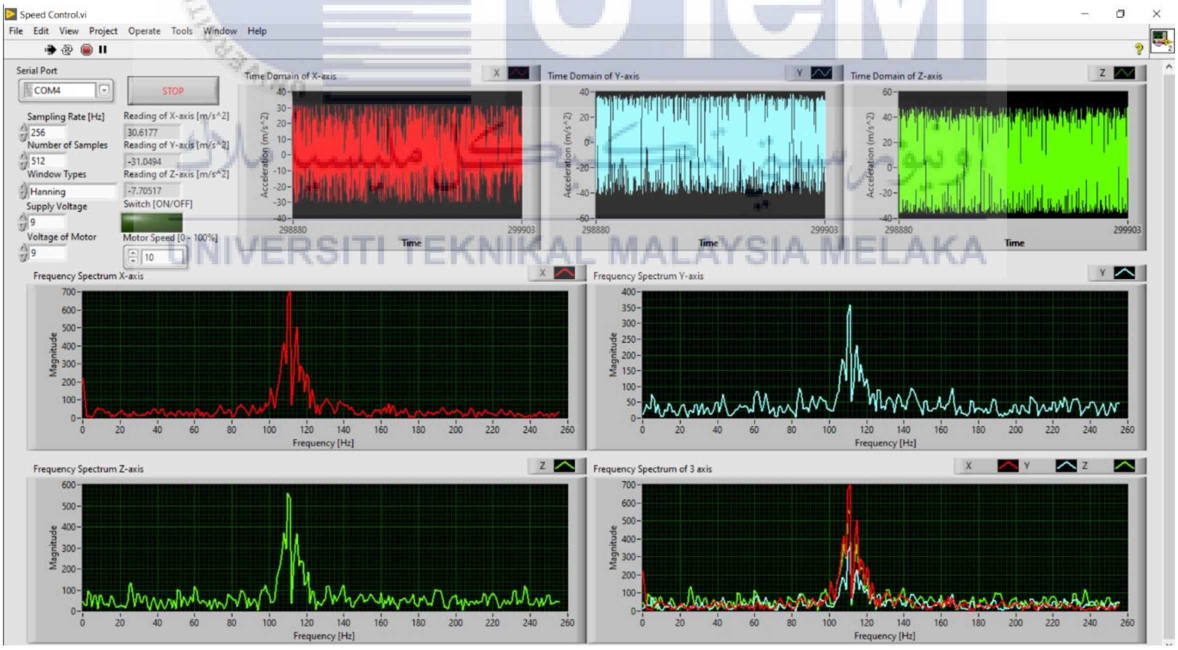

Figure 4. 8 Time Waveform and Frequency Spectrum of Actuator at maximum voltage

Table 4. 3 Table of comparison between Rated Frequency and Measured Frequency of Motor and Actuator

Components	Frequency Range (Hz)	Number of Samples	Rated Frequency (Hz)	Measured Frequency (Hz)
Motor	256	512	25	19
Actuator	256	512	140	110

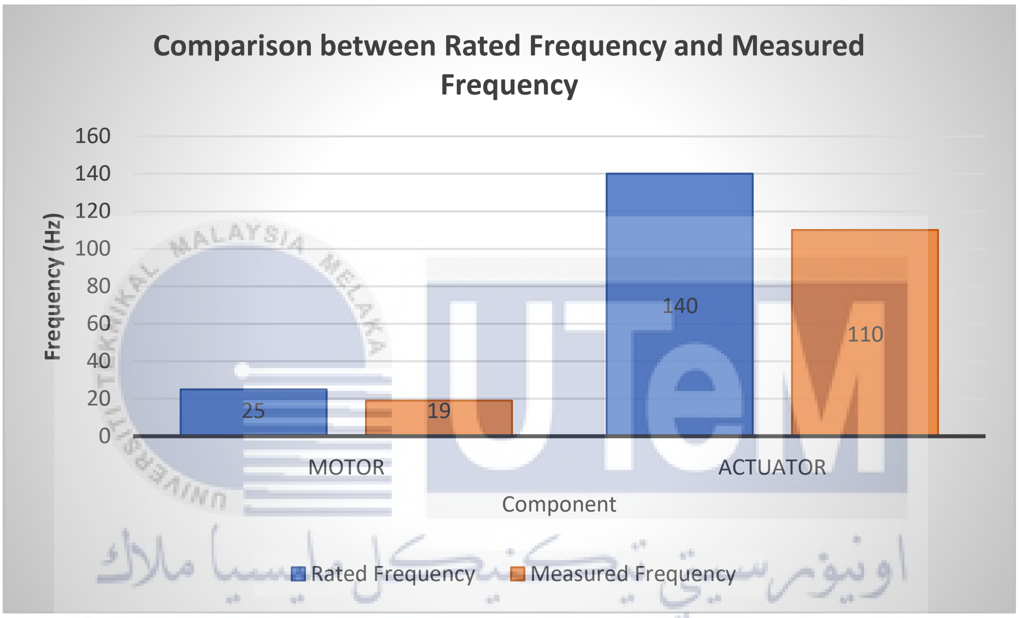

Figure 4. 9 Graph of comparison between Rated Frequency and Measured Frequency of Motor and Actuator

From the results, this system measured 19 Hz in the motor while the rated frequency of the motor was 25 Hz (1500 rpm). It showed an 76.0% accuracy on detecting the frequency of the motor. Moreover, this system measured 110 Hz in the actuator while the rated frequency of the actuator was 140 Hz (8400 rpm). It demonstrated an 78.6% accuracy on detecting the frequency of the actuator. This accuracy can be improved by using a higher sensitivity accelerometer.

4.2.2 Actuator Control

4.2.2.1 PWM Method

Speed control is one of the important criteria to build up an active vibration absorber which is actuator in this project, because the actuator need to be able to change its turning speed to match the frequency of the primary system. Therefore, a speed control system was constructed by using PWM method in LABVIEW. The block diagram of this system was shown in figure 4.8 while the front panel of the system was shown in figure 4.9. There were 4 new manipulating variables added to the system to complete the speed control task. The variables were on/off switch, supply voltage, motor voltage and motor speed. The on/off switch was to turn on the speed control function, the Arduino board sent a TRUE value to the motor driver when the switch is on. The supply voltage was set according to the rating of power source, 9V power supply was used. The voltage of the motor is set regarding to the maximum voltage of the motor, 9V motor was used. The motor speed was tested from 0% to 100%. The speed was control by duty cycle in PWM. While the duty cycle describes the amount of time the signal is in a high (on) state as a percentage of the total time of it takes to complete one cycle. For example, 4V signal can be generated from 5V supply voltage using PWM with a duty cycle of 80%. This PWM method was able to control the speed of the actuator by changing the output voltage from the motor driver to the motor. Table 4.4 showed the voltage reading of the actuator with the use of PWM duty cycle when supply voltage was 9V. Table 4.5 showed the voltage reading of the actuator with the use of PWM duty cycle when supply was 12V.

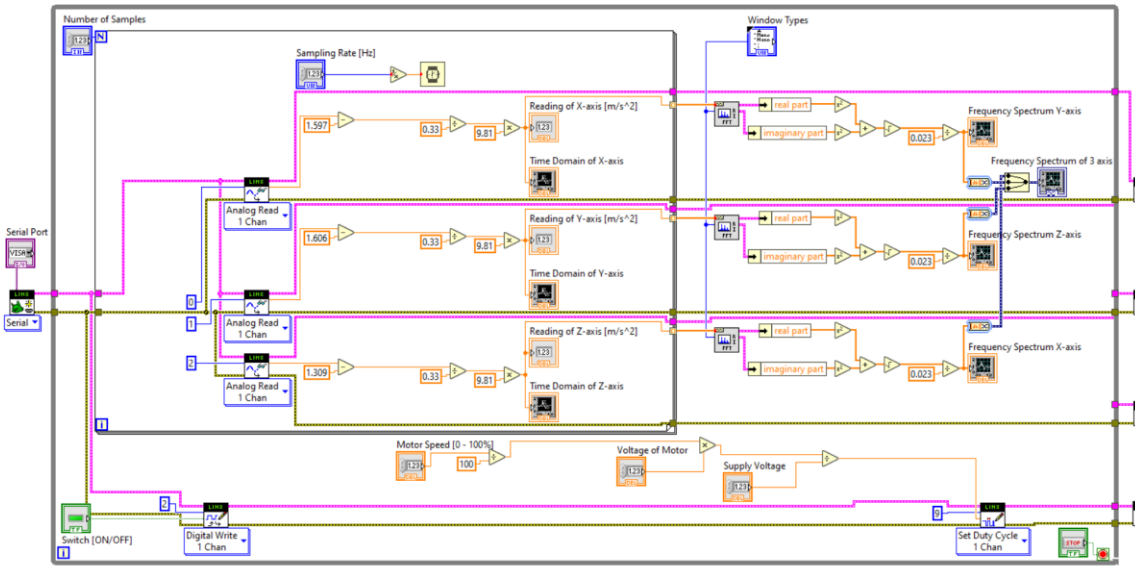

Figure 4. 10 Block diagram of actuator speed control system

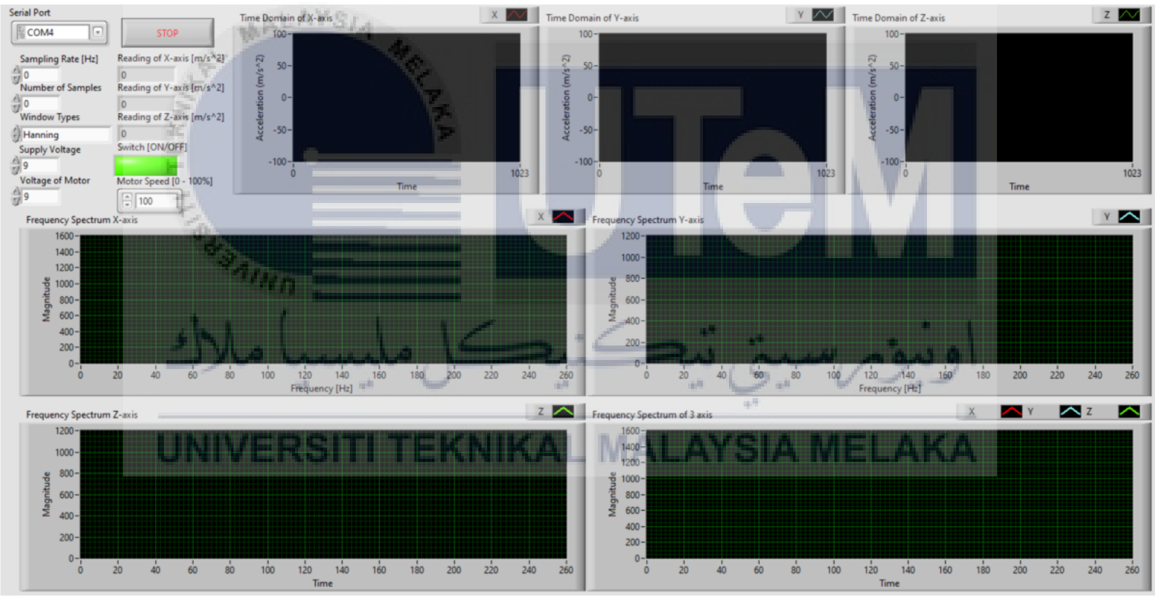

Figure 4. 11 Front Panel of actuator speed control system

Table 4. 4 Voltage Reading of the actuator when Supply Voltage = 9V

Motor Speed (%)	Theoretical Voltage Reading (V)	Actual Voltage Reading (V)
10	0.9	0.13
20	1.8	1.84
30	2.7	3.04
40	3.6	3.82
50	4.5	4.31
60	5.4	4.84
70	6.3	5.28
80	7.2	5.73
90	8.1	6.03
100	9.0	6.93

Table 4. 5 Voltage Reading of the actuator when Supply Voltage = 12V

Motor Speed (%)	Theoretical Voltage Reading (V)	Actual Voltage Reading (V)
10	0.9	0.21
20	1.8	1.63
30	2.7	3.01
40	3.6	4.11
50	4.5	4.88
60	5.4	5.56
70	6.3	6.07
80	7.2	6.71
90	8.1	7.03
100	9.0	7.51

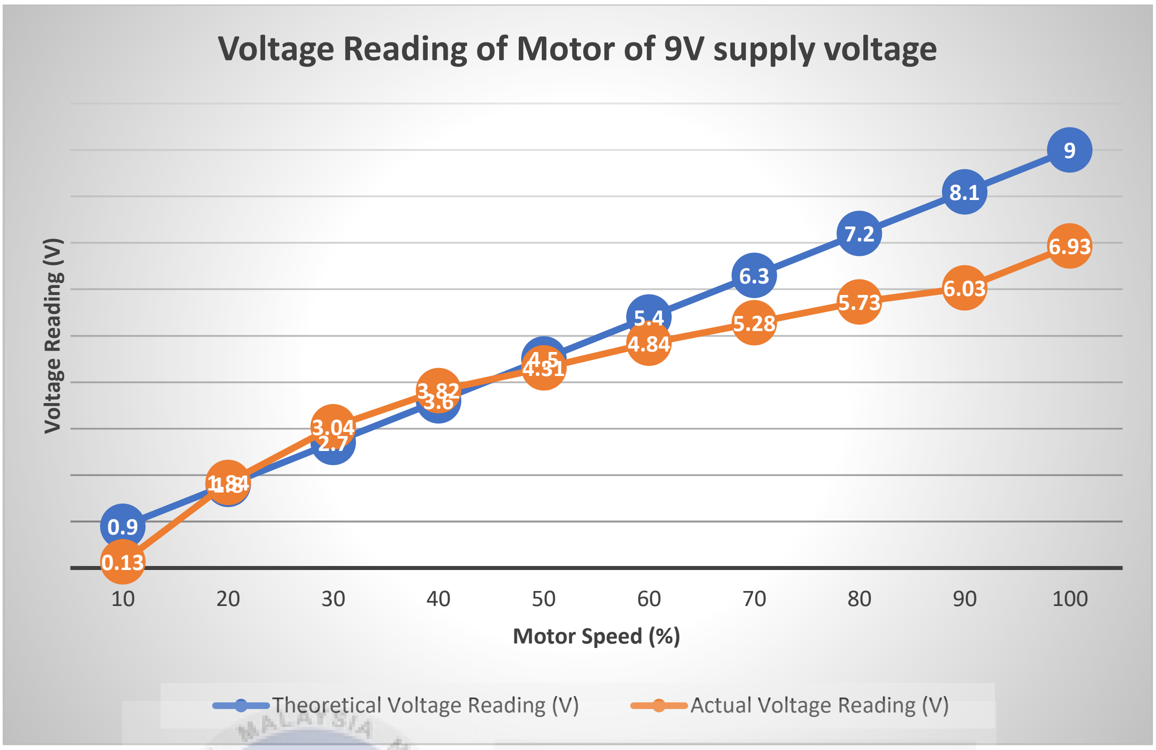

Figure 4. 12 Graph of Voltage Reading of the Motor of 9V Supply Voltage

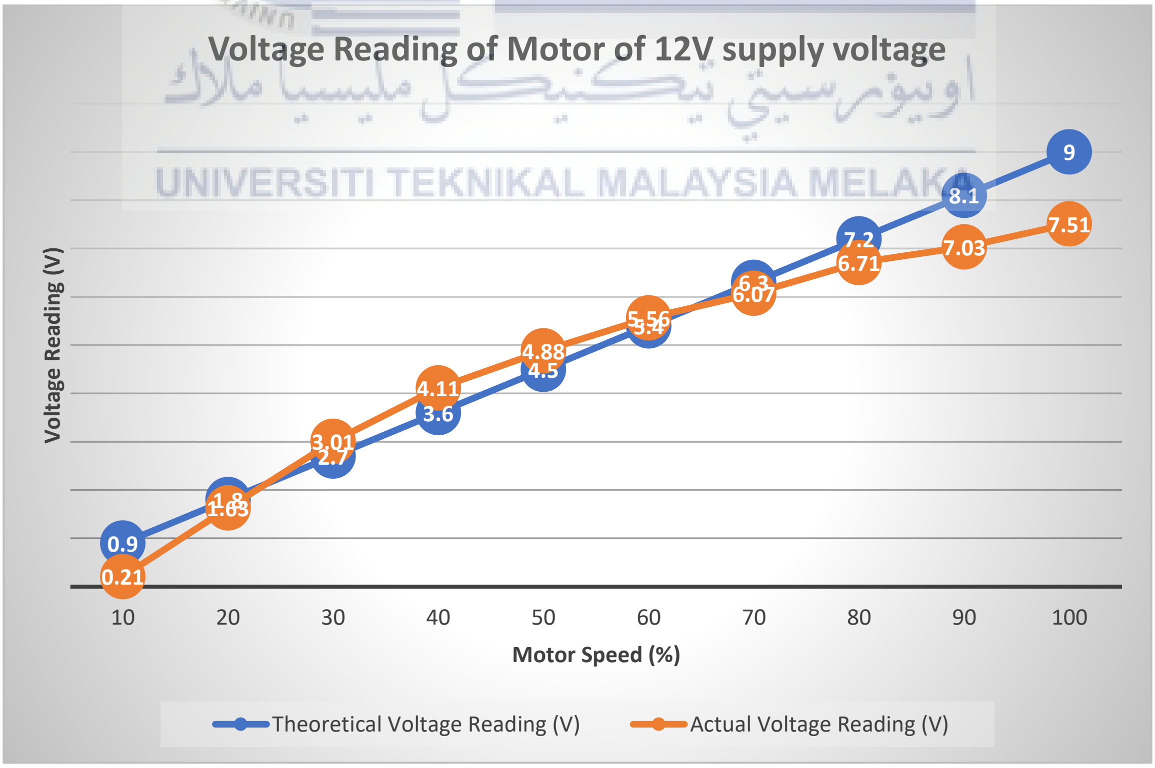

Figure 4. 13 Graph of Voltage Reading of the Motor of 9V Supply Voltage

From figure 4.12 and 4.13, 12V supply voltage offered a better accuracy on controlling the speed of the 9V actuator compared to 9V supply voltage. Yet, when the motor was controlled at 80% speed, the voltage did not reach 7.2V yet it dropped to 6.71V, it indicated that the speed of the motor was not exactly 80% of its maximum speed. The voltage differences were getting larger when the speed was controlled at 90% and 100%. The drop in voltage was due to the voltage loss, which was increased resistance or increased load in a circuit or system. Therefore, this actuator speed control system is only effective in controlling the speed of an actuator from 0% up to 70% of its maximum speed.

4.2.3 Performance Testing

4.2.3.1 Effectiveness of the DVA on suppressing the vibration

The frequency detection system was constructed to find out the key frequency of the primary system while the actuator speed control system was designed to change the turning speed of the motor to match the operating frequency of the primary system. The effectiveness of the DVA was tested by combining both systems by inspecting the changes in magnitude of the frequency spectrum with the DVA and without the DVA. The comparison between the frequency spectrum of a vibrating motor without the DVA and the frequency spectrum of the vibrating motor with DVA were shown in figure 4.14 and figure 4.15. Table 4.6 showed the magnitude of the frequency spectrums.

Figure 4. 14 The Frequency Spectrum of the motor without DVA

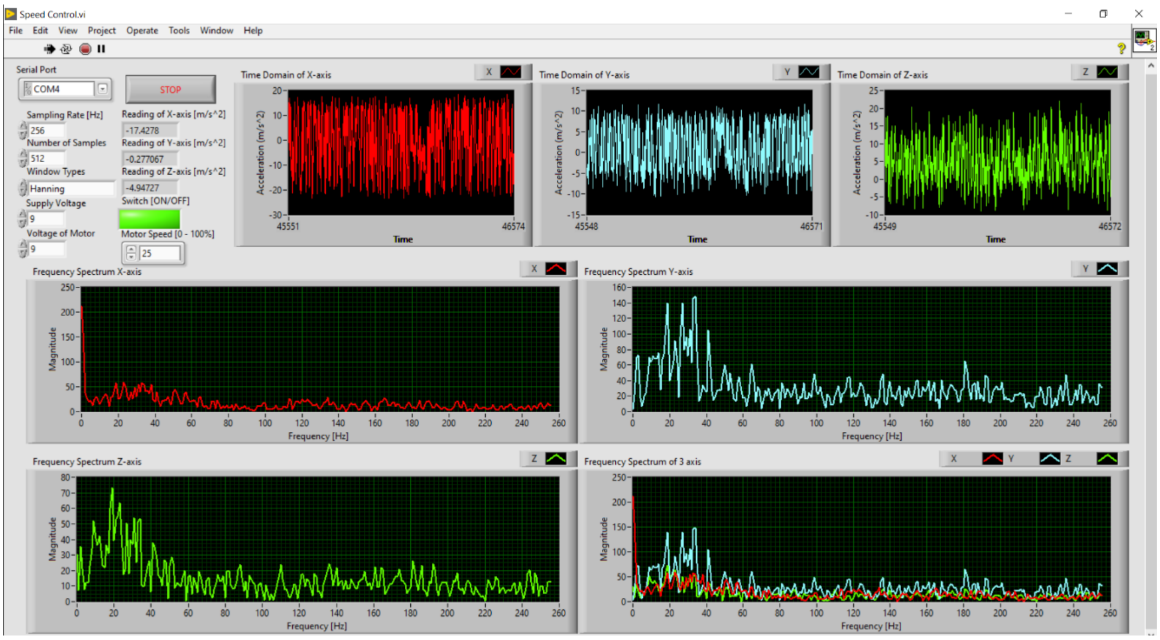

Figure 4. 15 The Frequency Spectrum of the motor with DVA

Table 4. 6 The amplitude of the motor with DVA and without DVA

Axis	Amplitude of the motor without DVA (dB)	Amplitude of the motor with DVA (dB)	Reduce in vibration (%)
X-axis	130	60	53.8
Y-axis	225	150	33.3
Z-axis	120	73	39.2

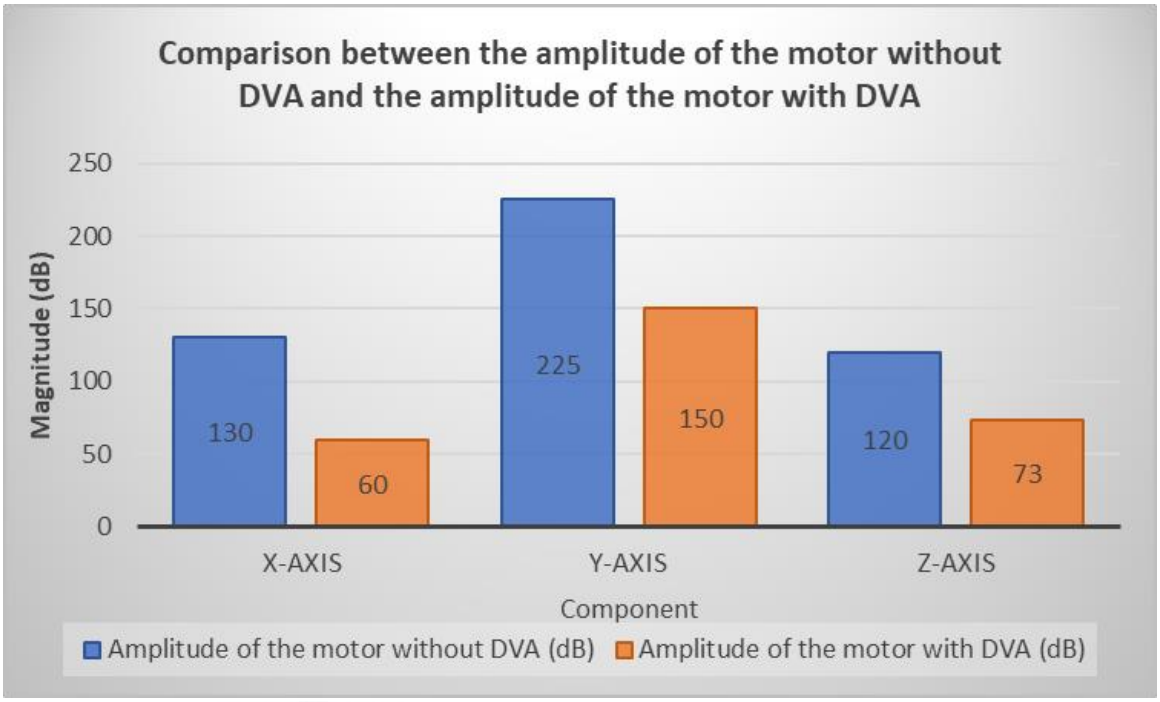

Figure 4. 16 The comparison graph of system amplitude without DVA and system amplitude with DVA

From figure 4.16, the amplitudes of the system with the use of DVA were decreased in all axis. The amplitude was decreased from 130 dB to 60 dB, it reduced 53.8% vibration in the X-axis. While the amplitude of Y-axis was decreased 33.3% from 225 dB to 150 dB. Moreover, there was a 39.2% drop in vibration in the Z-axis, the amplitude was decreased from 120 dB to 73 dB. The effect of suppressing vibration can be further improved by using a better motor driver board which can handle higher voltage rating motors. The tuning of frequency of the actuator is important because if the speed is not tuned precisely, it will be having a small variation in frequency with the operating frequency of the primary system. This will cause beating problem to the system.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

In this research, the performance of dynamic vibration absorber on suppressing the structural vibration was investigated. A dynamic vibration absorber was added into a source of vibration and the amplitude of the source was measured and to be compared. A frequency detection system was constructed to obtain the frequency spectrum of the system by using Fast Fourier Transform in LABVIEW. The key frequency of the source was shown in the frequency spectrum. Moreover, a speed control system for actuator was constructed to control the frequency of the actuator. PWM method was used in the system and it changed the speed of the actuator by controlling the duty cycle (0-255 or 0% to 100%), which its purpose was to control the average power delivered by an electrical signal. Furthermore, both systems were them combined to carry out the performance testing to evaluate the effectiveness of this DVA system.

5.1.1 Frequency Detection System

In this research, the frequency detection system was constructed by using programmable software, LABVIEW. An Arduino Uno board and accelerometer sensor ADXL335 was used in this system. The system was started with the calibration of the ADXL335 because its default output was in Voltage (V). The output was calibrated by subtracting the base level voltage (V), divided by its sensitivity (V/g) and multiplied with the gravitational acceleration 9.81 m/s^2. The calibrated output of the accelerometer sensor was now approximate 0 m/s^2. After calibration, Fast Fourier Transform function was added into the system. The time domain waveform was transformed into frequency spectrum and the frequency of the system was found. The first objective of this research was achieved.

- The base level voltage of X-axis, Y-axis- and Z-axis of the accelerometer sensor was 1.602V, 1.631V and 1.309V, respectively.
- The calibrated output of the accelerometer sensor was 0.422m/s^2 for X-axis, 0.431m/s^2 for Y-axis and 0.423 m/s^2 for Z-axis.
- The system detected 19Hz frequency on the motor at its maximum voltage while the rated frequency was 25Hz. It showed a 76% accuracy on detecting the frequency of the motor at its maximum voltage.
- The system detected 110Hz frequency on the actuator at its maximum voltage while the rated frequency was 140Hz. It showed a 78.6% accuracy on detecting the frequency of the actuator at its maximum voltage.

5.1.2 Actuator Speed Control

In this system, PWM method was used to control the speed of the actuator. The duty cycle was controlled from 0% to 100% to test the performance of the L298N motor driver board on controlling task. Before the test, the supply voltage and motor voltage were input into the LABVIEW to ensure the correct duty cycle was calculated. The voltage of motor was measured when the duty cycle was set from 0% to 100%. 12V power supply and 9V power supply were tested separately. From the data obtained from digital multimeter, it showed a more stable output with 12V power supply compared to 9V power supply. The system was only effective in controlling the speed of motor from 0% to 70% of its maximum voltage. Beyond 70%, there was a significant drop in voltage which was due to the increased resistance or increased load in the circuit. The second objective was reached.

5.1.3 Performance Testing

To test the effectiveness of this DVA system, the amplitude of the system before and after the application of DVA was recorded. The key frequency of the system was 19Hz, therefore the actuator was tuned to match the operating frequency of the system. The running frequency of the actuator was tuned to 19Hz and the frequency spectrum and amplitude was recorded. This system can be concluded as a reliable and

- The amplitude of the system in X-axis was decreased 53.8% from 130dB to 60dB
- The amplitude of the system in Y-axis was decreased 33.3% from 225dB to 150dB
- The amplitude of the system in Z-axis was decreased 39.2% from 120dB to 73dB

5.2 Recommendations

In this research, there are some recommendations and future works are proposed:

- The accelerometer sensor could be replaced with a higher sensitivity to further improve the accuracy of the data.
- The experiment could be carried out in a flatter table to reduce the calibration error.
- The experiment could also be conducted on a isolated surface where there are no vibration will affect the data taken from the primary system.
- The motor could be rebuilt with a different material to investigate the relationship between the vibration and the type of material used in the prototype.
- The motor driver board could be replaced by a higher rating one, thus the speed of the motor can be controlled more precisely.
- The frequency spectrum could be improved by using other advanced programmable software such as MATLAB and Python to further increase the quality of the frequency spectrum.
- The piezoelectric accelerometer could be used in this experiment to crosscheck the frequency that obtained from the accelerometer sensor ADXL335.

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APPENDIX

APPENDIX A – ADXL335 Data Sheet

ADXL335

SPECIFICATIONS

T_A = 25°C, V_S = 3 V, C_X = C_Y = C_Z = 0.1 µF, acceleration = 0 *g*, unless otherwise noted. All minimum and maximum specifications are guaranteed. Typical specifications are not guaranteed.

Table 1.

Parameter	Conditions	Min	Typ	Max	Unit
SENSOR INPUT	Each axis				
Measurement Range		±3	±3.6		*g*
Nonlinearity	% of full scale		±0.3		%
Package Alignment Error			±1		Degrees
Interaxis Alignment Error			±0.1		Degrees
Cross-Axis Sensitivity[1]			±1		%
SENSITIVITY (RATIOMETRIC)[2]	Each axis				
Sensitivity at X_{OUT}, Y_{OUT}, Z_{OUT}	V_S = 3 V	270	300	330	mV/*g*
Sensitivity Change Due to Temperature[3]	V_S = 3 V		±0.01		%/°C
ZERO *g* BIAS LEVEL (RATIOMETRIC)					
0 *g* Voltage at X_{OUT}, Y_{OUT}	V_S = 3 V	1.35	1.5	1.65	V
0 *g* Voltage at Z_{OUT}	V_S = 3 V	1.2	1.5	1.8	V
0 *g* Offset vs. Temperature			±1		m*g*/°C
NOISE PERFORMANCE					
Noise Density X_{OUT}, Y_{OUT}			150		µ*g*/√Hz rms
Noise Density Z_{OUT}			300		µ*g*/√Hz rms
FREQUENCY RESPONSE[4]					
Bandwidth X_{OUT}, Y_{OUT}[5]	No external filter		1600		Hz
Bandwidth Z_{OUT}[5]	No external filter		550		Hz
R_{FILT} Tolerance			32 ± 15%		kΩ
Sensor Resonant Frequency			5.5		kHz
SELF-TEST[6]					
Logic Input Low			+0.6		V
Logic Input High			+2.4		V
ST Actuation Current			+60		µA
Output Change at X_{OUT}	Self-Test 0 to Self-Test 1	−150	−325	−600	mV
Output Change at Y_{OUT}	Self-Test 0 to Self-Test 1	+150	+325	+600	mV
Output Change at Z_{OUT}	Self-Test 0 to Self-Test 1	+150	+550	+1000	mV
OUTPUT AMPLIFIER					
Output Swing Low	No load		0.1		V
Output Swing High	No load		2.8		V
POWER SUPPLY					
Operating Voltage Range		1.8		3.6	V
Supply Current	V_S = 3 V		350		µA
Turn-On Time[7]	No external filter		1		ms
TEMPERATURE					
Operating Temperature Range		−40		+85	°C

[1] Defined as coupling between any two axes.
[2] Sensitivity is essentially ratiometric to V_S.
[3] Defined as the output change from ambient-to-maximum temperature or ambient-to-minimum temperature.
[4] Actual frequency response controlled by user-supplied external filter capacitors (C_X, C_Y, C_Z).
[5] Bandwidth with external capacitors = 1/(2 × π × 32 kΩ × C). For C_X, C_Y = 0.003 µF, bandwidth = 1.6 kHz. For C_Z = 0.01 µF, bandwidth = 500 Hz. For C_X, C_Y, C_Z = 10 µF, bandwidth = 0.5 Hz.
[6] Self-test response changes cubically with V_S.
[7] Turn-on time is dependent on C_X, C_Y, C_Z and is approximately 160 × C_X or C_Y or C_Z + 1 ms, where C_X, C_Y, C_Z are in microfarads (µF).