DEVELOPMENT OF VIBRATION TEST RIG FOR TEACHING AND LEARNING

NOOR HAKIMI BIN NOOR ROZALI

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

DEVELOPMENT OF VIBRATION TEST RIG FOR TEACHING AND LEARNING

NOOR HAKIMI BIN NOOR ROZALI

A report submitted
in fulfillment of the requirements for the degree of
Bachelor of Mechanical Engineering with Honour

Faculty of Mechanical Engineering

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

2019

DECLARATION

I declare that this project report entitled "Development of Vibration Test Rig for Teaching and Learning" is the result of my own work except as cited in the references

Signature : ……………………………………….

Name : ……………………………………….

Date : ……………………………………….

APPROVAL

I hereby declare that I have read this project report 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 with Honour.

Signature	:	…………………………….
Supervisor's Name	:	…………………………….
Date	:	…………………………….

DEDICATION

To my beloved mother and father

ABSTRACT

Vibration is very important to be learned by the student in order to grasp the understanding of how vibration may bring harm to a system and machine and how it can be controlled and surpassed to get the desire condition. The mass-spring-damper simple model made the learning easy as the system and structure is simplified. In this study, the vibration test rig is developed in order to make the teaching and learning simpler and can be understand easier. In order to do that, the test rig will be design from its frame to make it ergonomics and the parameter used in the test rig will be carefully selected so that the test rig experimental test will be the same as theoretical calculation. The gained information from the past studies and research bring a lot of help in developing the vibration test rig. The imbalanced motor is attached at the centre of the beam with both end of beam is attached to the spring to the top of the frame. The rotation of imbalance motor will causing the beam to periodically move up and down from its equilibrium position which is known as vibration. Accelerometer will be used to determine the displacement of the beam. Later, the absorber will be used which will be attached directly under the motor to be compare to the test without absorber. The result will shows that using an absorber will be able to control the unnecessary vibration to prevent harm to the system. The testing result and theoretical calculation will shows almost the same in their graph shape which represent the vibration of the system when forced vibration with and without absorber is tested. The result shows that the test rig is capable to generate such achievement which will help the student to study vibration and applied the knowledge to the system as they learned in class.

ABSTRAK

Getaran adalah satu perkara penting yang perlu dipelajari oleh pelajar untuk memastikan kefahaman tentang bagaimanana getaran mampu untuk memberi kesan buruk kepada sesuatu sistem dan mesin dan bagaimana getaran dapat dikawal dan diatasi untuk mendapatkan keadaan yang diinginkan. Model mudah mass-spring-damper *dapat membantu untuk memahami dengan lebih mudah kerana sistem dan struktur sudah dipermudahkan. Kajian ini bertujuan untuk membangunkan rig ujian getaran untuk membantu memudahkan proses belajar dan mengajar dan lebih mudah untuk difahami. Penghasilan rig ujian getaran dimulakan daripada mereka bentuk bingkai untuk memudahkan penggunaannya dan penggunaan parameter dipilih dengan berhati-hati untuk mengesahkan rig ujian getaran sama dengan pengiraan teori. Semua maklumat yang diperoleh daripada kajian dan penyelidikan terdahulu telah membantu untuk membangunkan rig ujian getaran. Motor tidak seimbang telah diikat pada tengah rusuk dengan kedua-dua hujungnya disambungkan dengan pegas ke bahagian atas bingkai. Pusingan daripada motor tidak simbang akan menyebabkan rasuk untuk bergerak secara berkala ke atas dan ke bawah dari kedudukan keseimbangannya yang dikenali sebagai getaran.* Accelerometer *akan digunakan untuk menentukan anjakan oleh rasuk. Kemudian, penyerap akan disambung pada bahagian rasuk di bawah kedudukan motor untuk dibandingkan dengan ujian tanpa penyerap. Keputusan ujian akan menunjukkan penggunaan penyerap akan membantu untuk mengawal getaran yang tidak diingini untuk mengelakkan kemudaratan pada sistem. Keputusan ujian dan pengiraan berdasarkan teori*

akan menunjukkan bentuk graf yang hampir sama yang mana mewakili getaran pada sistem apabila ujian dilakukan ke atas sistem untuk getaran yang tiada penyerap dan ada penyerap. Keputusan perbandingan menunjukkan rig ujian getaran mampu untuk menghasilkan keputusan yang memberangsangkan yang mana akan membantu pelajar untuk mempelajari getaran dan menggunakan ilmu dipelajari pada sistem tersebut sepertimana yang mereka belajar di dalam kelas.

ACKNOWLEDGEMENTS

First of all, I would like to say thank you to Allah S.W.T. for give me the opportunity to finish up my research and project. Thanks to Him, I am able to give my full attention during the process of finishing the project. Next, I take this chances to express my sincere acknowledgement to my mother and father for all the time they are supporting me mentally and physically throughout the development of this project.

I would like to express my greatest gratitude to my supervisor, Professor Madya Dr. Azma Putra from the Faculty of Mechanical Engineering for his essential supervision, support and encouragement towards the completion of this entire project. This project help me gain more knowledge on vibration and process development of project.

Lastly, I would like to thank my beloved friends who help and support me throughout this project and helping me finalizing this project and report within the limited time. Thank you.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF APPENDICES

CHAPTER 1

INTRODUCTION

1.0 Background

Vibration is an oscillation of a mechanical system about an equilibrium position. The principle of vibration to work is when an inertia or force element is given onto the mechanical system from its equilibrium position due to an energy transmitted that act as external force to the system (Graham, 2012). There are many small branches under the vibration to be listed such as harmonic excitation, free vibration, force vibration, transmissibility and absorber. Every one of the branches is working directly under the vibration.

Everything that works under the principle of motor will produce the vibration. Even the sound produced as a person is talking is due to a vibration of the sound. However, it is depend on the machine or some parameter to ensure the necessity of the vibration on the system as it has an advantageous and also disadvantageous. As the system is vibrating, it will vibrate at certain set of frequency force. If this frequency corresponds with the system or structure natural frequency, it will cause a resonance that result in fatal big oscillations which is the structure will start to vibrate extravagantly (Nikhil, 2015; Jaini 2014). Some system needs a machine to vibrate at high speed frequency such as screening machine to filter something according to its size and some other system need to reduce the vibration for the machine to work smoothly and reduce the risk for broken machine.

Force vibration occurs when an external force is given or transmitted to the system rather than coming from the system itself which is the natural frequency. As the mechanical system is at rest, the started motion of unbalance motor will give inertia to the system that will result in vibrational motion. The vibration frequency is started due to the external force known as force vibration.

Generally, transmissibility is a ratio between the amplitude of the force transmitted to the base and the amplitude of the excitation force (Lage, 2014). It also can be known as the ratio when an external force is applied to the system and the corresponding excited by the base. Usually, the transmissibility is control by three parameters which are mass, spring and damper. Each one of the parameter will result in different value and different effect on the system.

Next, absorber is working as a function to absorb a certain force applied to the system and transmit remaining force to the system (Chaudhari, 2017). The absorber also known as the isolator which it absorbs the unnecessary vibration transmitted to the system and reduce the frequency on the system.

As a student learnt about the vibration in the class, student will need a certain way to comprehend the knowledge of vibration for better understanding. Even by looking into the picture and watching the video of the working principle of vibration could not give the student a better understanding as they need something to implement their knowledge to make sure they are truly understand of the vibration. This is where a vibration test rig comes in the hand to help the student.

1.1 Problem Statement

Student has been exposed to the theory of the vibration in the classroom. There are the basic of vibration which is mass, spring and damper before moving on to free and forced vibration, transmissibility and absorber at the end of the chapter. Most student can only see the effect of the variable of vibration as it work in place without knowing how to implement the theory into real working world as they cannot relate the theory as they cannot see the simple version of those things.

Hence, the objective of this project is to develop the vibration test rig to study the concept of force vibration, transmissibility and absorber as to assists the student to add into their lack of knowledge of the real version of the vibration process and how it work when some parameter is change and the result of the change. Therefore, this project will able the student to be more understands to the vibration subject.

1.2 Objective

The objectives of this project to be achieved are:

I. To design and construct a vibration test rig.
II. To test the performance of the test rig to produce good data of vibration.

1.3 Scope

The scopes to be covered in this project are:

I. To design the test rig by using the CAD software.
II. To develop the test rig according to the design.
III. Test and analysis on the vibration test rig to ensure the functionality of the test rig is the same as the theory.
IV. To assist the student on teaching and learning of the vibration.

CHAPTER 2

LITERATURE REVIEW

2.0 Introduction

This chapter describes the aspect that related to development of vibration test rig. This review method is used to gather and collect the data and any information about the product. All the information will be analyse and review in order to get the better understanding in the development of the product.

2.1 Concept of Vibration

An alternation of physical phenomena where it takes place such as repeating itself as respected to the time are defined as vibration. To put it simply, a repeated motion for an interval amount of times is called oscillation or vibration. Thus, vibration is dealing with a study of oscillatory motion of a bodies and any other force that associated to it.

A machine or its component as it produced a cyclic or oscillating motion from its equilibrium state or its position of rest is known as vibration (Krunal, 2017). Thus, the repetitive motion of a machine which it is repeating the motion from its nominal position is defined as vibration. Vibration is usually dealing with the displacement of the oscillating motion of object in respect to the time (Graham, 2012).

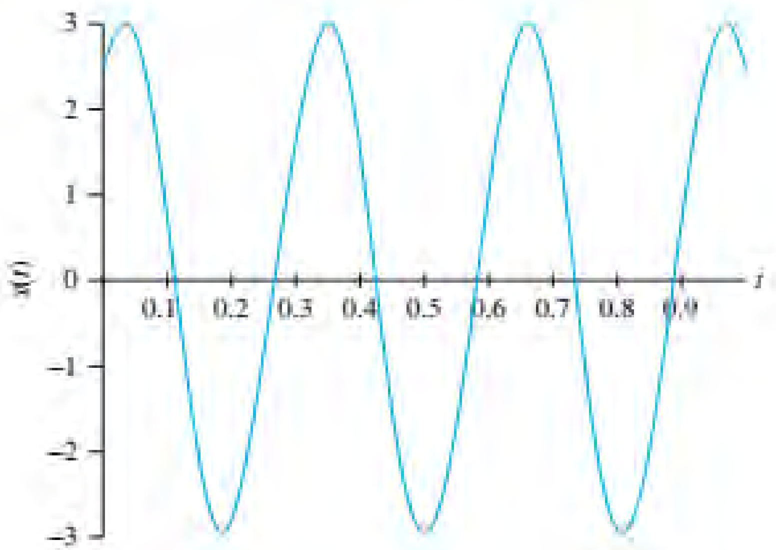


Figure 2.1: Periodic simple harmonic displacement of motion against time

The most common source of vibration that can be seen in everyday life is pump, compressor, vibrator and fans. The excitation of vibration occurs when the centre of mass of the rotating driven part is misaligned with the centre of motor's rotation that causing the centrifugal force to excite outward. The unbalance force that is going outward from its axis of rotation will transfer the excitation force to the other structure connected to the excitation sources causes the vibration. Thus, the eccentric mass and the speed of the rotation of the motor is the keys to determine the characteristic of the vibrations.

Vibration takes place in many structural and mechanical systems that if its unnecessary uncontrollable vibration appears, it could lead to harmful and calamitous situations. So, the vibrations need to be study and understood fully in order to prevent any unnecessary situation as its role is too important in engineering in order to develop a safe design, construction and operation of machine and structure that generally associated with the vibrations (Singiresu, 2007).

By studying the simple mass-spring-damper model, a person should be able to understand the basic of vibration analysis. A complex structure such as bridges could be modeled as an addition of a lot of mass-spring-damper into one complex system. Vibration cases can be classified into several types such as (Graham, 2012; Jaini, 2014);

2.1.1 Damped and undamped

Damped system is where the force is continuously depletes over time as it leads to continual decrease in the kinetic and potential energy with the help of damper. Undamped system is where the vibration oscillated about the rest position and as it reaches equilibrium, the energy is less than the previous reading. Generally, damped system dissipate its energy faster than undamped system.

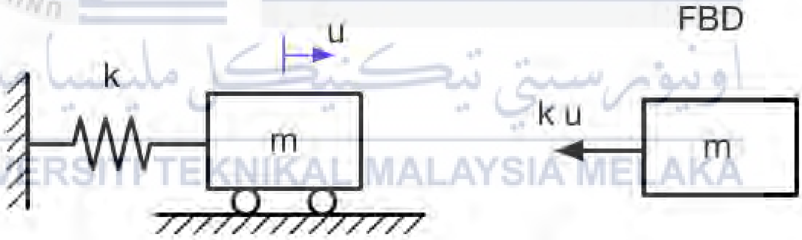


Figure 2.2: Undamped system

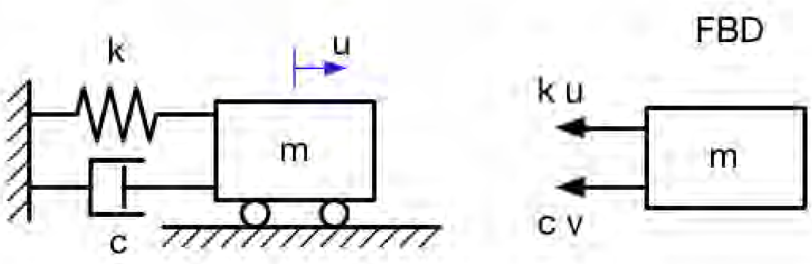


Figure 2.3: Damped system

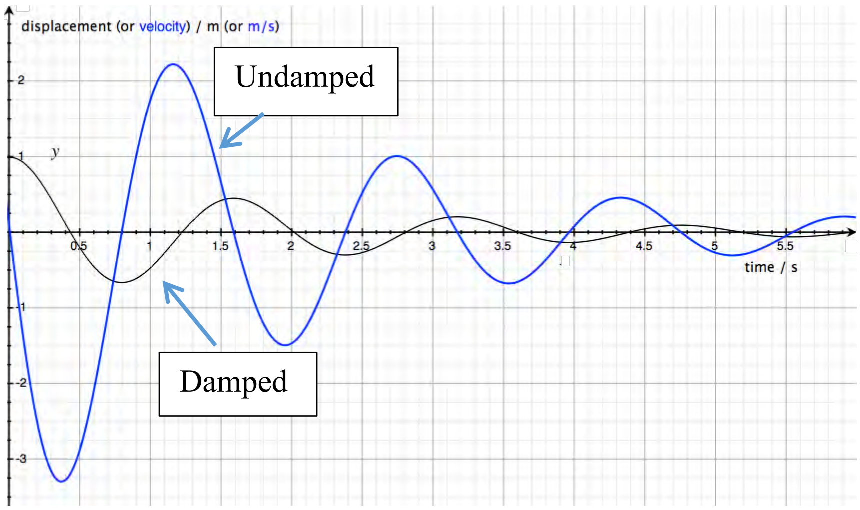


Figure 2.4: Comparison between damped and undamped

2.1.2 Free and force vibration

A system which is oscillated from the equilibrium position in the absence of external force is called as free vibration. However, when the external force is imparted to the system that excited it to cause an oscillation from its rest state is known as force vibration. The difference between both is the present of external force (Graham, 2012).

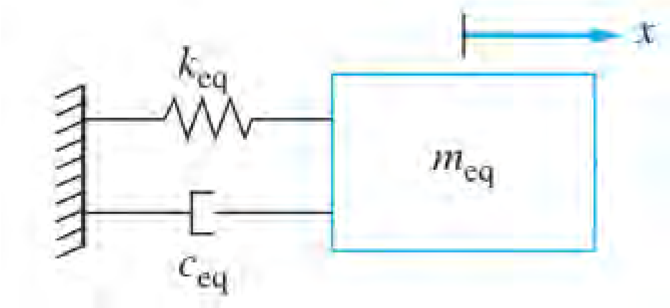


Figure 2.5: Free vibration

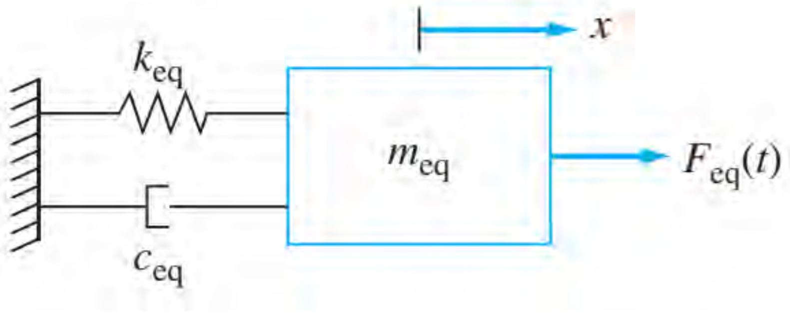


Figure 2.6: Force vibration

2.1.3 Transmissibility

Transmissibility can be defined as the ratio of the amplitude of the response displacement to the amplitude of the displacement forced at the base (Lage, 2014). In other words, it is the ratio of the transmitted force to the force of the base. When the force is transmitted to the base, it will result in the excitation force from the base.

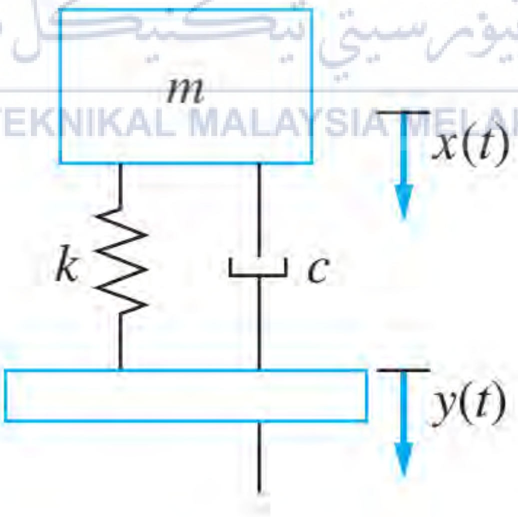


Figure 2.7: Transmissibility system excite the mass and base

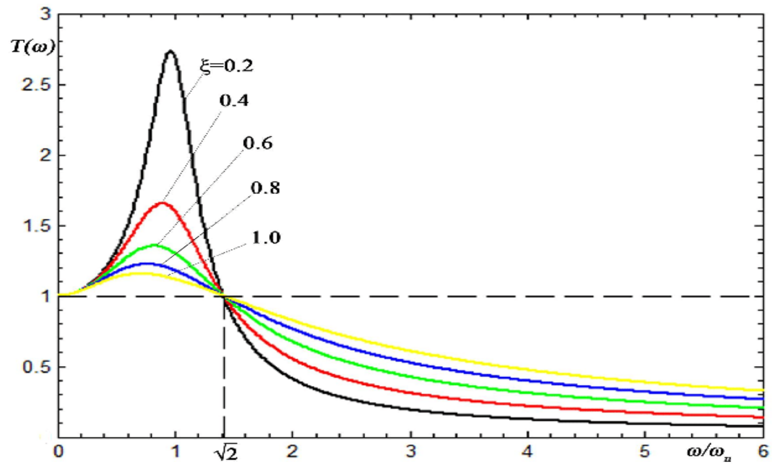


Figure 2.8: The effect on damping factor on transmissibility graph

Usually, transmissibility goes by the formula of;

$$T = \left|\frac{x}{y}\right| = \frac{\sqrt{1+\left(2\zeta\frac{\omega}{\omega_n}\right)^2}}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left(2\zeta\frac{\omega}{\omega_n}\right)^2}} \tag{2.1}$$

The formula can be simplify into;

$$T = \left|\frac{X}{Y}\right| = \sqrt{\frac{1-4\zeta^2r^2}{\left(1-r^2\right)^2+4\zeta^2r^2}} \tag{2.2}$$

Where ζ is damping factor, ω is operating frequency and ω_n is natural frequency of the system and r is ratio of the rotating speed of the motor to the natural frequency (ω/ω_n).

2.1.4 Absorber

Absorber is a function of a system which absorbs unnecessary force excited by the system and transfer the rest directly to the system. When a machine is working at a frequency that are near its natural frequency, it will result in large amplitude that could severe the machine. One of the countermeasure is to alter the properties of the system which will result in the natural frequency is away from the excitation frequency (Graham, 2012).

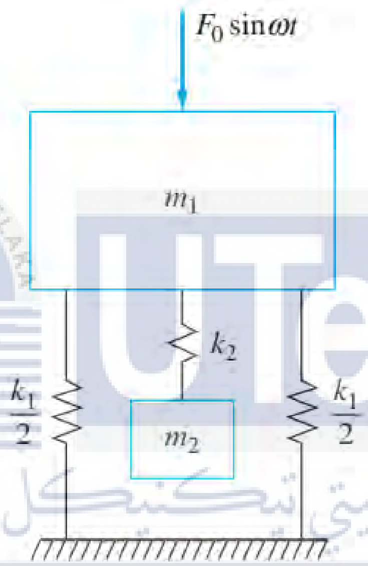


Figure 2.9: Absorber under the mass

The absorber can be tuned so that it can eliminate unwanted vibration which is also known as isolator. It will isolate or eliminate steady-state vibration of a system where the absorber is attached if it is adjusted to the excitation frequency.

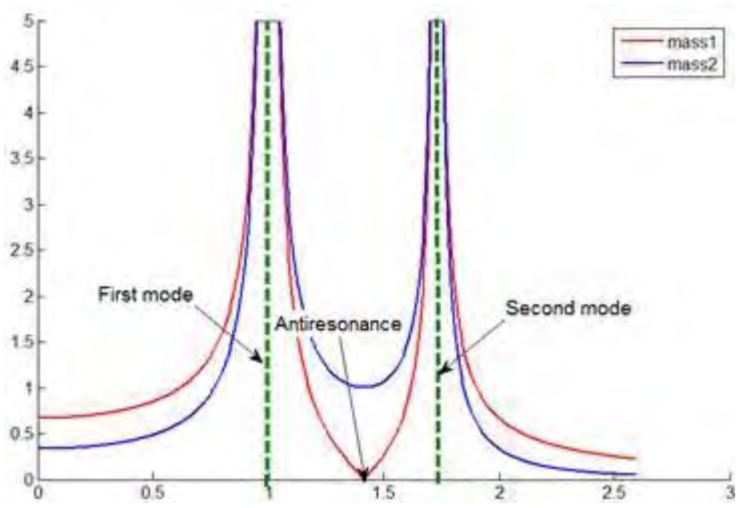


Figure 2.10: Graph of the effect of the absorber

According to **Figure 2.10**, it shows that the natural frequency of the system is shifted. The absorber act as isolator and also can be the anti-resonance as it reduce the amplitude of the previous natural frequency to almost zero. It could be a great help if the system is operating near the natural frequency and avoid catastrophic failure.

2.1.5 Natural frequency and mode shape

The vibration also associated with frequency which is without frequency, there is no vibration. Everything has its own frequency that called as natural frequency. While the system in a state of vibration as a force frequency usually applied to the system, the system itself has its own natural frequency. Natural frequency could be defined as the state where the object vibrates without external force which is the frequency where the undamped force response happens naturally (Graham, 2012). The frequency of vibration cannot be the same as natural frequency which will causing severe harm to the system. Thus, the frequency of vibration applied to the system has to be controlled.

Frequency is expressed in ω, while natural frequency is labeled as ω_n. The undamped natural frequency of an object could be expressed as;

$$\omega_n = \sqrt{\frac{k}{m}} \tag{2.3}$$

Where ω_n is natural frequency (rad/s), k is stiffness rate (N/m) and m is object mass (Kg).

There are also damped natural frequency at which the damper is present in the system. Thus, the damper factor will be included in the equation of damped natural frequency which is;

$$\omega_d = \omega_n \sqrt{1-\zeta^2} \tag{2.4}$$

Where ω_d is damped natural frequency (rad/s) and ζ is Damping factor, where it is equal to;

$$\zeta = \frac{c}{2m\omega_n} \tag{2.5}$$

Where c is damping rate (Ns/m).

2.2 Beam

Beam is one of the general structure element that can be implement in any structure or mechanism as the characteristics of the beam such as its stiffness is very important constraint element in bending (flexure) mechanism design (Jaini, 2014). Beam has an ability to maintain it forms and when the external force is applied to it, the external reaction is called as bending moment. It is likely used in building structure, machine frame and bridges.

Structural or mechanical system usually designed by the beam as its main frame. The different kind of the end-condition will causing the transferred vibration to react in different ways. The automobile main structure which contain the beam has its own different end-condition in order to receive the different excitation force from the outside force. If the vibration force excite the beam structure near to its natural frequency, large excitation force will causing the problem on the structure such as fatigue and stress over the yield stress of the structure. If it cannot be dealt swiftly, it can break the structural system apart.

The design of the structural system should be done thoroughly which the structure can work with the eccentric motor that causes the vibration. So, the proper design should be known of its detail characteristic such as its mode shape and excitation frequency in order for the system to work in the safe area away from its excitation force to avoid resonance and away from its limited area.

2.2.1 Characteristic

Beam has a very special trait which is it able to bend even though it is sturdy. This is due to stiffness in the beam. Every beam has its own stiffness which it could be determine using a formula which consists of Modulus of Elasticity (E) and its cross-section of length, width and height. Thus, the material of the beam is important and the dimension should be specific in order to get the more precise stiffness.

$$k = \frac{AE}{L} \tag{2.6}$$

Where; k is stiffness, A is cross section area (m^2), E is modulus of elasticity (Gpa) and L is length of a beam (m).

2.2.2 End-condition of beam

Beam has many different end-condition be use according to the design specification such as simply supported, overhanging, cantilever and fixed end beam. In this designing process, the type of beam that are likely to be used are simply supported beam, pinned end beam and cantilever beam.

Each beam has its effective length based on its end-condition which means the length of the beam cannot be calculate as a whole but only calculated on the effective length which the deflection may occurs. Thus, the theoretical value of effective length is used as the basic on calculation and analysis.

2.2.2.1 Simply supported beam

It is consists of a beam that are only supported by another item that are not permanently attached to it. The motor will be subjected at the centre of the beam and the spring will be attached at the end of both beam which will held the beam at high altitude.

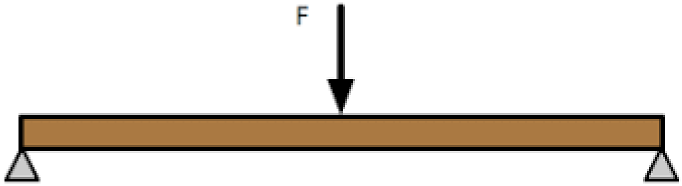


Figure 2.11: Simply supported beam diagram

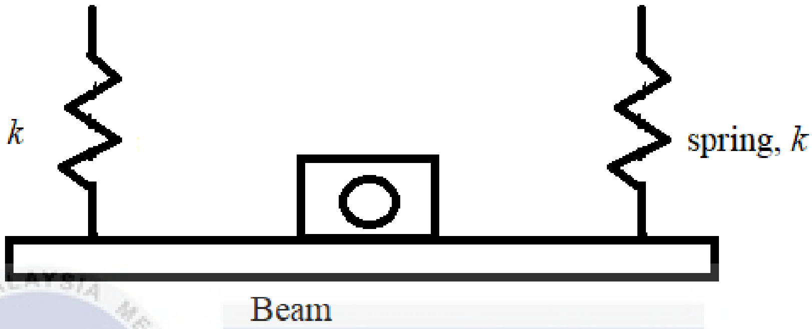


Figure 2.12: Sketch of motor on the beam supported by spring

From the figure 2.12, the condition of the beam likely to be used in the experiment of the attached spring at both end in order to determine the relationship between the eccentric mass rotation of the motor and the different stiffness of the attached spring at both end of the beam. The testing will neglect the stiffness of the beam and only its mass will be added into the parameter of the experiment.

2.2.2.2 Pinned end beam

Both end of the beam is pinned down to restrict the horizontal movement of the beam that will only allow the vertical movement. The imbalance motor attached at the centre of the beam will cause the beam to vibrate which the beam will show a repetitive vertical movement.

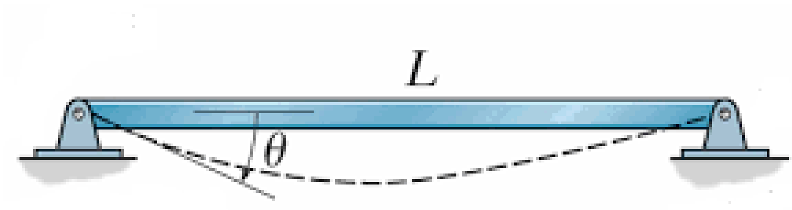


Figure 2.13: Pinned end beam

The design of the pinned end beam usually used in the structural design of beam which used bolt and nuts as its connector to the neighbour structure. So, the effect of the effective length of the vibration on the beam is different from the other type of end-condition.

2.2.2.3 Cantilever beam

Figure 2.14: Cantilever beam

Cantilever beam is the beam where one ends is fixed while the other one is hanged. The fixed-side will support all the forced applied on the beam such as moment force and shear stress. The purpose of the cantilever beam is to produce a bending effect until certain limit.

2.3 Past Research

The study of mechanical vibration in mechanical system is very important as many industry are in need to eliminate unnecessary vibration such as in machines, bridges and tall buildings. Many different concept has been developed for a suitable way to suppress the unwanted vibration. The uses of spring, mass and damping is the significant factor in controlling the vibration.

Some of the tall building is using a tuned-mass damper in order to isolate the vibration that likely happen during earthquake. The most famous tuned-mass damper that has been used is at Taipei 101 Building in Taiwan. The increasing in the high of high-rise structure possess extreme challenge that come with many problem and critical issues that are involved such as earthquake and the effect of the wind that challenge a person in the engineering field. TMD was first recommended by Frahm in 1909 that consists of spring mass and damper to reduce the unnecessary vibration in ships (Alex, 2014). By using a TMD, the unwanted vibration is able to be reduced during earthquake in May 2008. The TMD play a major part in that earthquake that moves to compensate for the movement of the building.

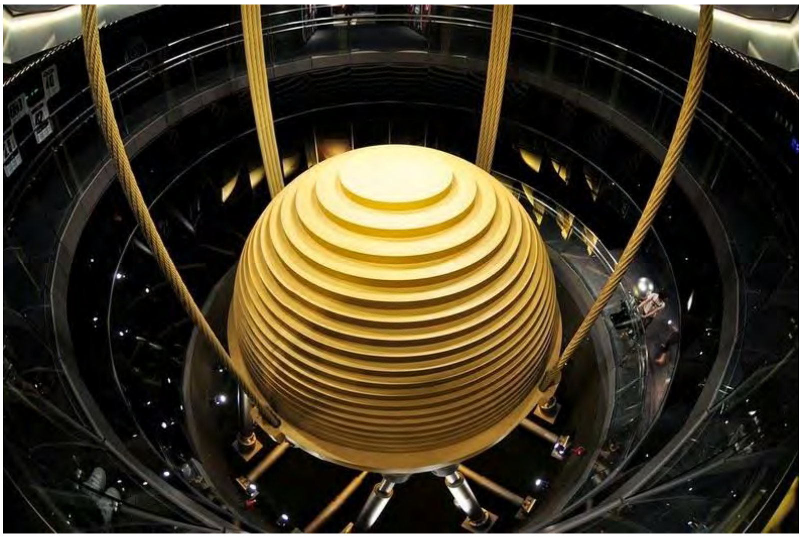


Figure 2.15: Shock absorber of Taipei 101 Building

In a research and the development of a test rig by Nikhil T. *et al.*, the resonance occurs in a machine as the operating frequency coincides with a natural frequency of the structure which causing a resonance where it would result in dangerously large oscillations. The vibrations of a system is transferred from its source to all the beam structure with different end conditions to the ground. As the beam characteristics is define by its length, cross-section, material and support conditions, the difference in the characteristics of the beam can have an effect on the vibrations produced. The test rig to determine the different vibration characteristics of a beam was developed which can support different type of beam support which are cantilever, simply supported and fixed-fixed end conditions. There were many variable to be test which are the effect of different location of the eccentric DC motor on the different type of beam and the different in the RPM speed of the DC motor. The result shows the different in the data about the variation in length, cross-section, material, boundary condition, centrifugal force and the location of the DC motor on the beam. Thus, Nikhil T. *et al.* successfully developed the test rig to study the effect of different characteristics and parameters on the response of the beam.

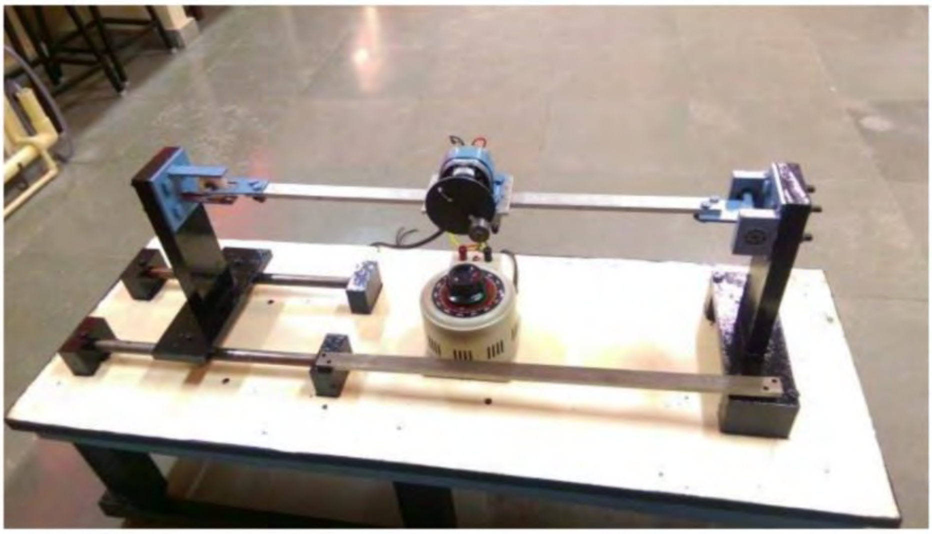


Figure 2.16: Test rig to determine the characteristics of beam

The research done by N. Jaini *et al.* was to prove the application of the dynamic vibration absorber (DVA) which is also known as TMD. The designing of the test rig was done by clamp the fixed end beam to a static structure and the shaker to generate the vibration is placed onto one side end of the beam. The shaker generate harmonic exciting frequency in transverse direction. The test is done by taking the measurement of the vibration without a DVA and after installing the DVA directly under the shaker by using accelerometer. The result shows positively as the present of DVA can reduce the vibration that are transferred directly to the beam.

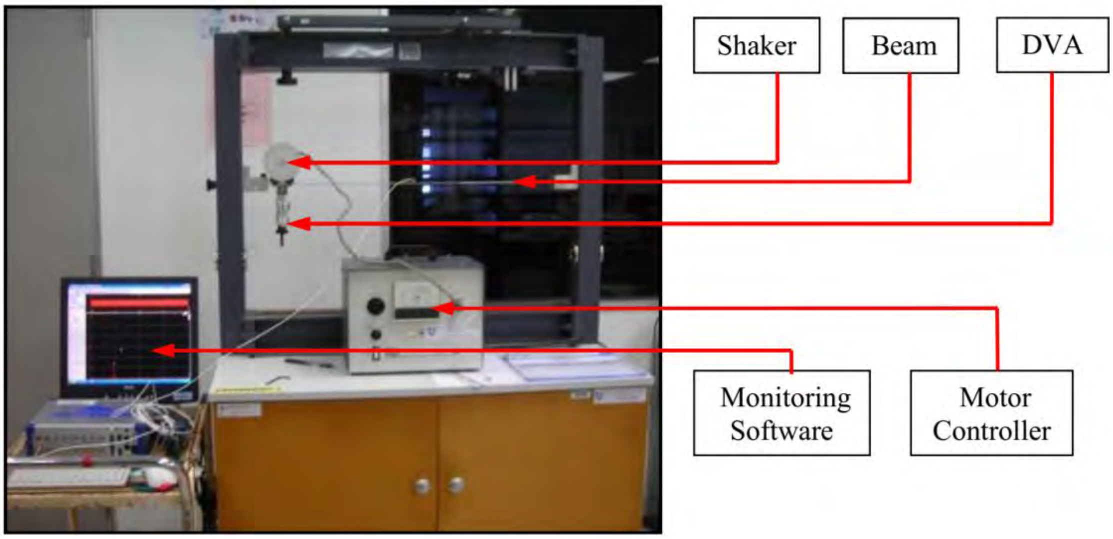


Figure 2.17: Vibration test rig setup

CHAPTER 3

METHODOLOGY

3.1 Introduction

In this chapter, the procedure and any method that has been used to develop the vibration test rig is provided. The process from the designing process to developing process and testing process also will be described in order to get the full information on the constructing process.

3.2 Product Design Specification

Product design specification (PDS) is a document structure that are created at early stage for the purpose, functions, characteristics and any related information about the product. The PDS will evaluate the product design which will fulfill the requirements and objectives of the product development. There are many characteristics of the PDS that need to be fulfill in order to define the product as a good product. The PDS for the frame of test rig vibration is shown below:

1. Performance

    i. The frame will be able to support the whole structure which consists of beam, spring, mass, motor and etc.

    ii. The height of the frame should be at the level of the eye.

2. Ease of installation

    i. The frame must be easy to be set up.

    ii. The fastener used to combine the structure should be simple.

3. Ergonomics

    i. The test rig should be easy to operate by the user.

    ii. The body posture for the user need to be taken into account to get the suitable height for the frame to prevent back pain problem.

4. Ease of maintenance

    i. The product can be fix easily.

    ii. The frame should be easy to assemble and disassemble.

5. Safety

    i. The material used for the frame should not have sharp edge and corner.

    ii. There are no excessive part that will restrict the movement of user and causing injury.

6. Cost

    i. The cost of material should be affordable and worthy.

7. Materials

    i. The material selected should be durable and anti-corrosion.

3.3 Parameter selection

The material used to build the product should be selected carefully. Thus, an orderly manners to fulfill the requirements is by specifying the criteria for the selected item to be used in the development of the product.

3.3.1 Beam selection

The selected beam is very important since the stiffness of the beam is one of the most important parameter in determining the result during testing. Thus, the dimension of the beam is the up most important and need to be well defined. The stiffness can be calculated using theory calculation and the characteristic of the beam should be well informed such as its modulus of elasticity and dimensions. However, there are several type of a beam and the theory calculation is based on the condition of the beam.

$$k = \frac{AE}{L} \tag{3.1}$$

Where, k is stiffness, A is cross section area (m^2), E is Modulus of Elasticity (Gpa) and L is length of a beam (m).

As mention before, the calculation will be different based on the type of a beam used such as cantilever beam, simply supported beam, overhanging beam and fixed-end beam.

3.3.2 Spring selection

Spring is important in developing the test rig vibration since the spring will be attached to both side of the beam to act as supported stiffness to see the vibration causes by the imbalance motor. Spring also is the main part in designing the absorber to isolate or reach the state of harmonic with the vibration of the beam.

There are many method to determine the stiffness of the spring such as the force applied divided by the final displacement of the extracted spring. The value of force is usually negative due to its reaction force applied on the spring. The formula of the spring's stiffness usually goes by;

$$k = \frac{-F}{x} \tag{3.2}$$

Where, F is force applied on the spring (N) and x is distance the spring is stretched from its equilibrium position (m)

3.3.3 Vibration absorber selection

Absorber to be used in the product and the testing is a simple spring connected with a mass. The absorber will be attached under the center of a beam to determine the effect of absorber on the vibration of a beam connected with imbalance motor.

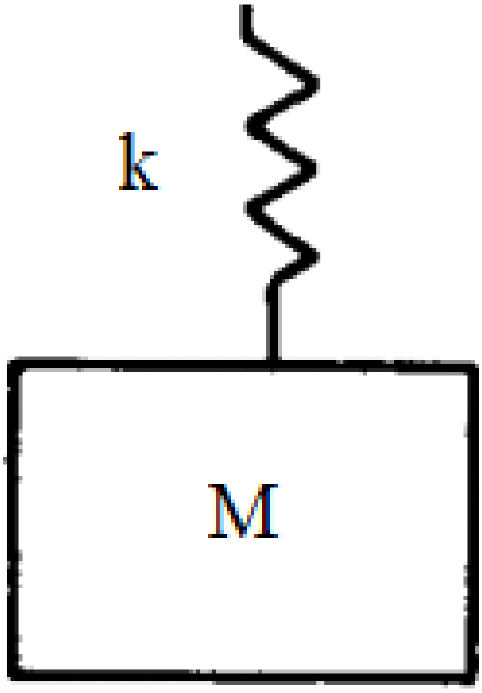


Figure 3.1: Absorber diagram

3.3.4 Frame material selection

The material of the frame should not be to heavy since the product should be portable and easy to transfer to anywhere. The suggested material to be used in development of vibration test rig is aluminium. The height of the frame should be at the level of the eye to ease the user to use the product. The dimension or profile of the frame used is T-slotted profile 40mm x 40mm.

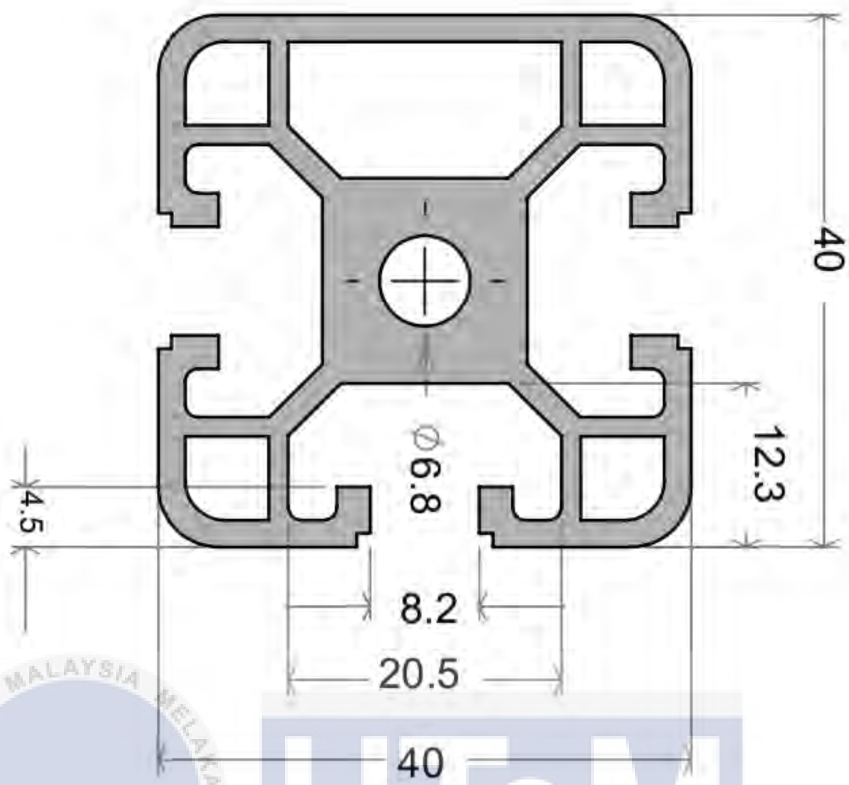


Figure 3.2: Frame profile

3.4 Conceptual Design

The process will be started with having a few of conceptual design to be evaluated based on the strong point and any requirement to be meet in order for the design to be selected. Each conceptual design will be described to provide better understanding on its advantages and disadvantages. Then, the conceptual design will be selected based on the requirement that need to be meets. CATIA V5 software is used to develop the conceptual design and the dimension of each concept will be displayed for the references during selection.

3.5 Design Selection

The design selection will be decided by using Pugh Matric Evaluation Chart which will provide a score for each of the conceptual design based on the criteria and requirement set. The score will be decide subjectively from one another. At the end, the conceptual design will be ranked in order to determine the best choice of the conceptual design and proceed to the development of the test rig.

3.6 Model Design Generation

The vibration test rig is a very important development product that will help in teaching and learning for the vibration analysis. Thus, a model for the test rig is constructed before finalize the prototype in order to prove the test rig is able to implement the teaching and learning in the class. The small scale vibration test rig is constructed by using a simple material of wooden frame, 12V DC motor with imbalance mass, spring attached at both end of a bar and a simple spring-mass absorber.

3.7 Prototype Development Process

The design will go through fabrication process to build the vibration test rig. The material will go through the process of cutting, grinding, fastening, and finishing to construct the test rig frame. Then, beam, eccentric motor and any other components that are to be used in the development of the test rig will be installed.

3.8 Simulation and Testing

The vibration test rig will be simulated by the help of the accelerometer that will be attached to the beam in order to get the data from the beam such as the vertical displacement in order to prove and compare with the theory calculation. The test that will take place are as follows:

1. Forced vibration test without absorber with different rpm speed.
2. Forced vibration test with absorber at different rpm speed to approve the function of absorber in the system.
3. Forced vibration test to see the effect of the absorber on the beam vibration with added mass to the absorber.
4. Forced vibration test to analyze the effect on the excitation based produced to approve the transmissibility graph.

As mention before, the data collected from the testing will be compare to the theoretical calculation in order for the test rig to be used as helping hand in teaching and learning.

3.9 Experimental set-up

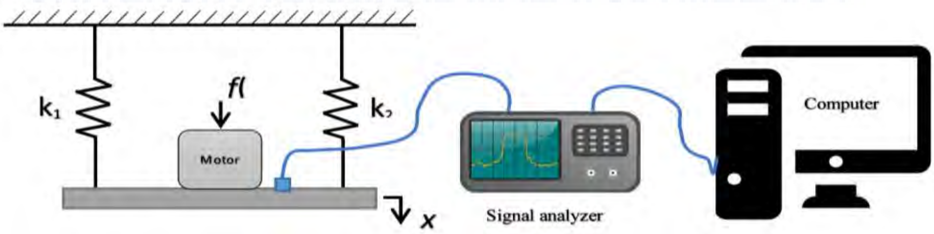


Figure 3.3: Experimental set-up for testing

The vibration test rig will be set-up in the way such as shown in **Figure 3.3**. The accelerometer will be placed on the beam in order to read the movement of the beam as shown in **Figure 3.4**. The accelerometer is connected to the signal analyzer which in this case is Data Physics as shown in **Figure 3.5**. Then, Data Physics will convert the reading taken from the accelerometer to make it readable by a computer. Software used in the testing is SignalCalc ACE which will shows the data in spectrum and graph. The computer will display the result of the test based on the condition on the beam.

Figure 3.4: Data Physics

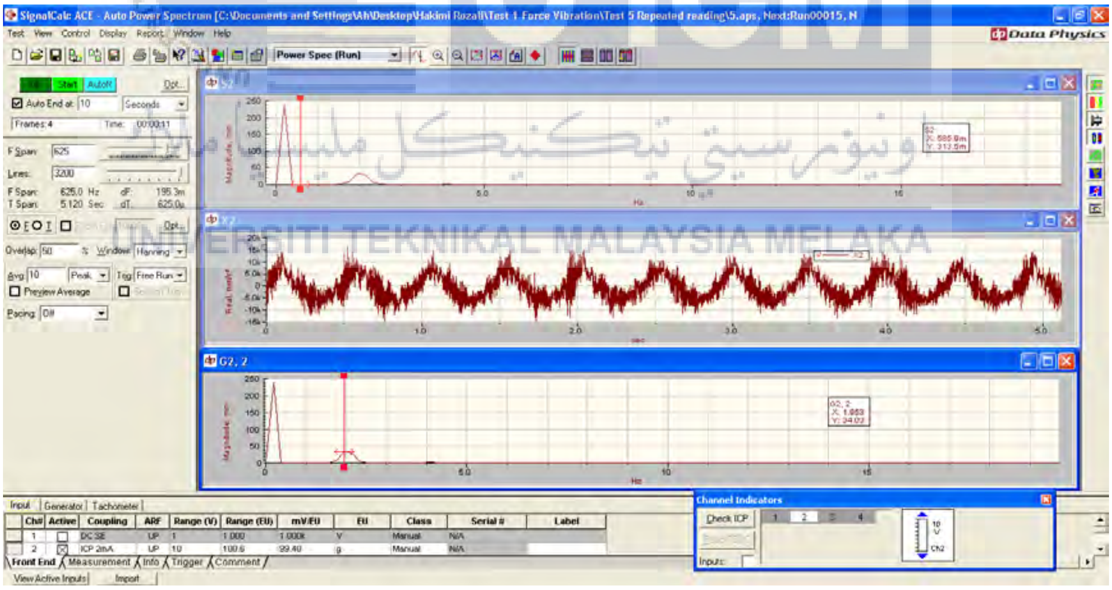


Figure 3.5: Example of testing data on SignalCalc ACE

Figure 3.6: Accelerometer on the beam

There are many accelerometer to be choose from which is in this testing there are two accelerometer to be used which have high sensitivity. The reason why high sensitivity accelerometer to be chosen is due to low frequency generated by the system. Low sensitivity accelerometer cannot pick up any signal below its designed limit. Thus, high accelerometer of 99.4 mV/g and 102.8 mV/g is selected.

Figure 3.7: Accelerometer

There will be several test which is the vibration test rig will be tested to achieve the objective of the development as mention before which are forced vibration without the absorber. Then the absorber will be placed underneath the motor. The test will be run and the effectiveness of the absorber can be seen by naked eye and the data will be display on the computer.

Next, the test will consists of testing the vibration test rig with different imbalance motor speed. The result will shown in the behaviour of the absorber that will change with the motor speed. The result will shows the different phase happens on the vibration test rig.

The result from the test and experiment will be compared with the theoretical calculation based on the theory learned by the student in the class. The similarity between the result of the experiment and the theoretical calculation will shows that the vibration test rig is ready to be used for teaching and learning in the class.

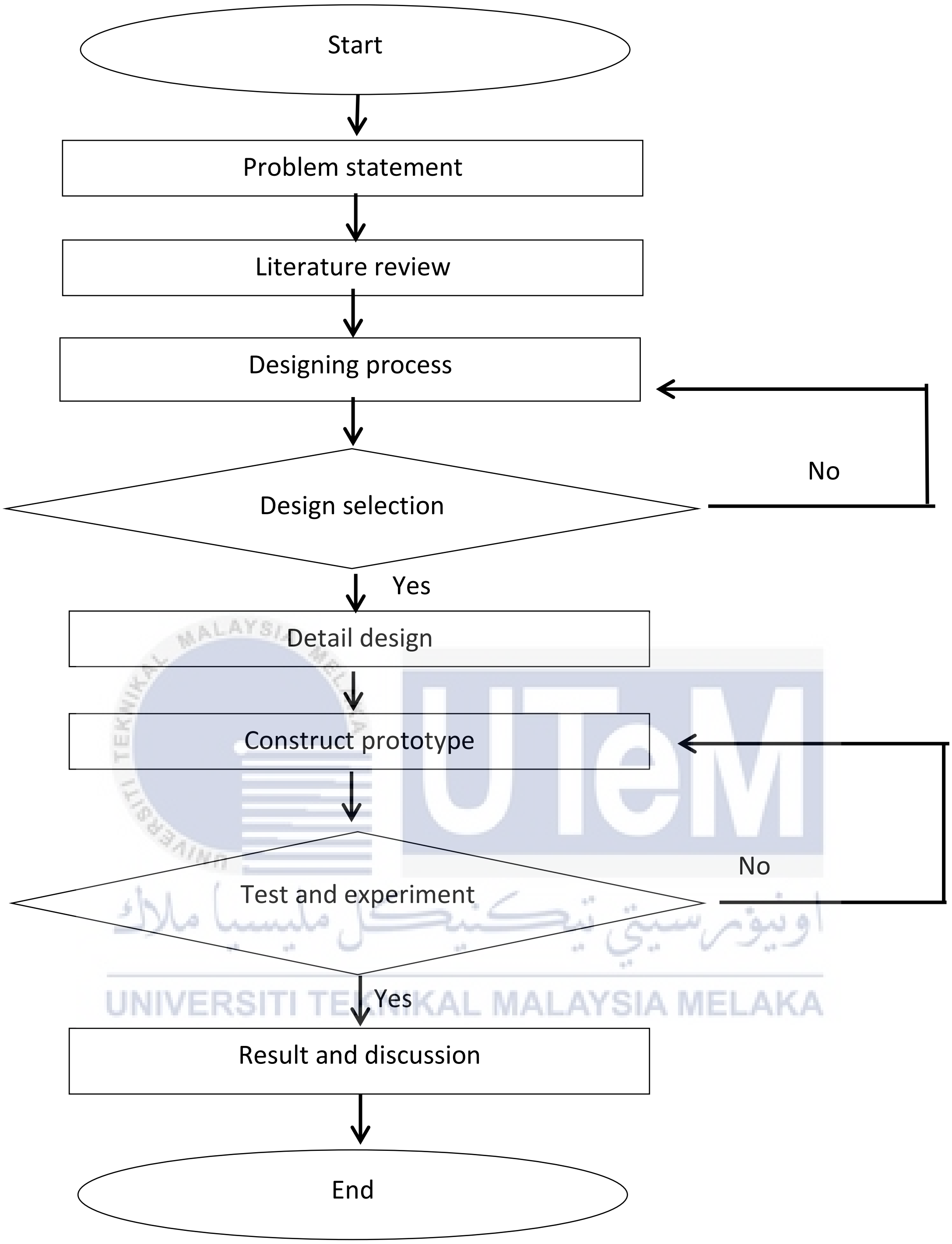


Figure 3.8: Flow chart

CHAPTER 4

DATA AND RESULT

4.0 Introduction

In this chapter, the conceptual design, Pugh matric evaluation, the model design generation and prototype development process are discuss in detailed. The detail result for the testing and simulation also will be presented and compared to the theoretical calculation in order to prove the effectiveness of the vibration test rig.

4.1 Conceptual Design

The conceptual design is generated by taking the spring-mass-damper design as it is based and supported by the frame. The design will be different from its size and shape. The design need to achieve the product design specifications in order for the design to be selected. Three designs have been developed to achieve the objective of the study and will be evaluated using Pugh metric evaluation chart.

4.1.1 Conceptual design 1

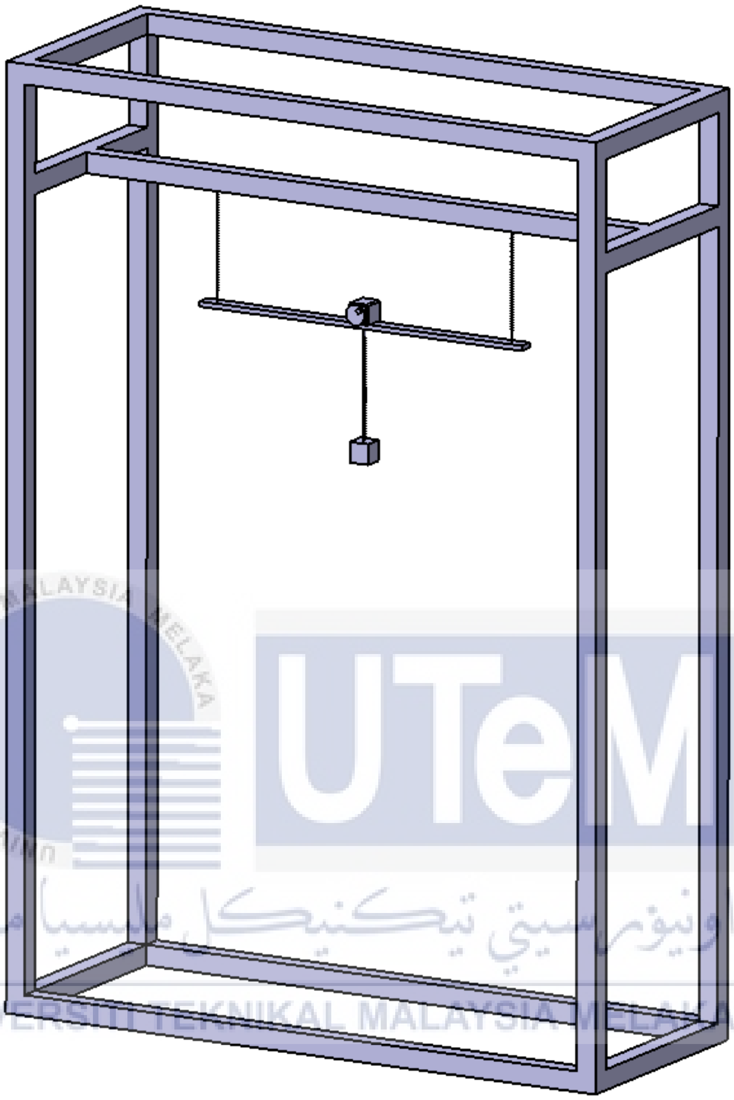


Figure 4.1: Conceptual design 1

Figure 4.1 shows conceptual design 1 which has the advantage of being more stable. The rectangular design will make the frame to stay put even there are force applied to the frame due to its width base area. The height of the frame is also good because the user can see the movement of the beam at the eye level which will prevent back injury. However, the cost to build the frame will be expensive due to many uses of material. Please refer ***Appendix B*** for further detailed design.

4.1.2 Conceptual design 2

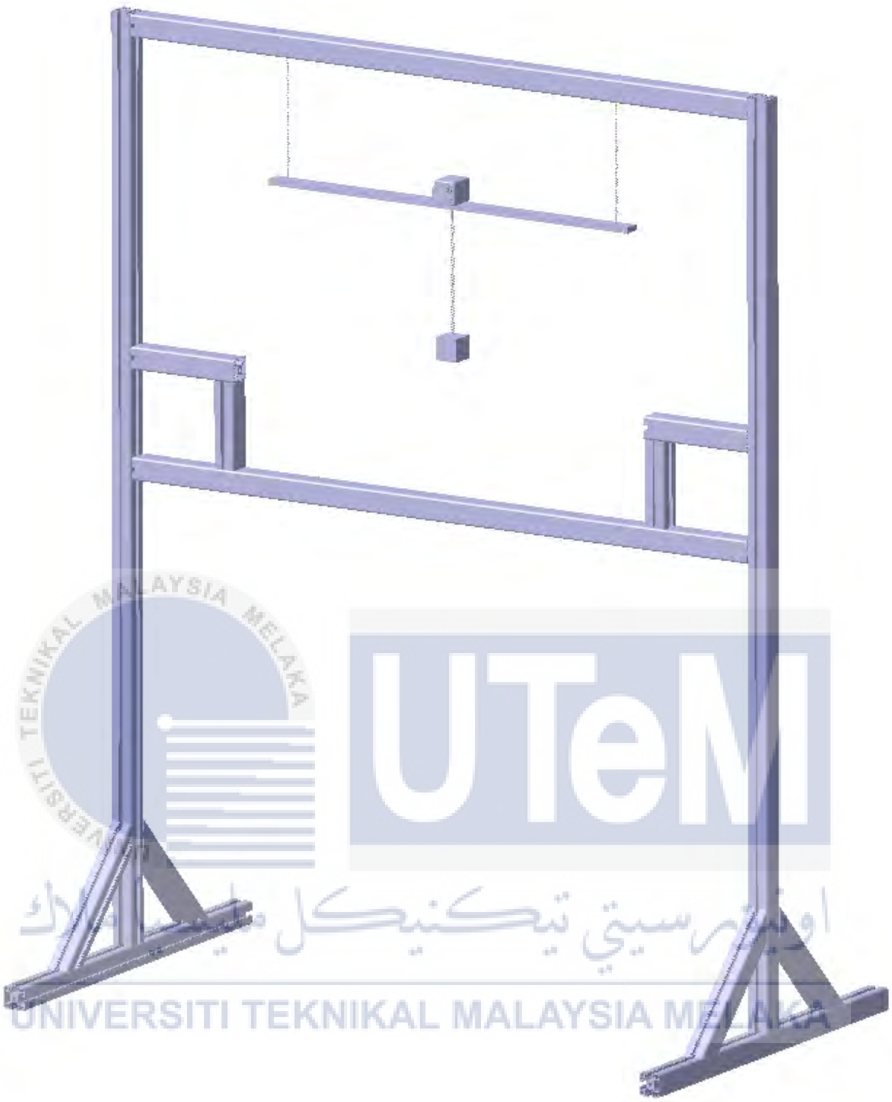


Figure 4.2: Conceptual design 2

Figure 4.2 shows the conceptual design 2 which is ergonomic for the users. The user will be able to operate the test rig without causing a back posture problem due to the beam is placed at eye level. The design is simple and the frame is supported at the bottom to prevent from the test rig to fall forward and backward. Please refer ***Appendix C*** for further detailed design.

4.1.3 Conceptual design 3

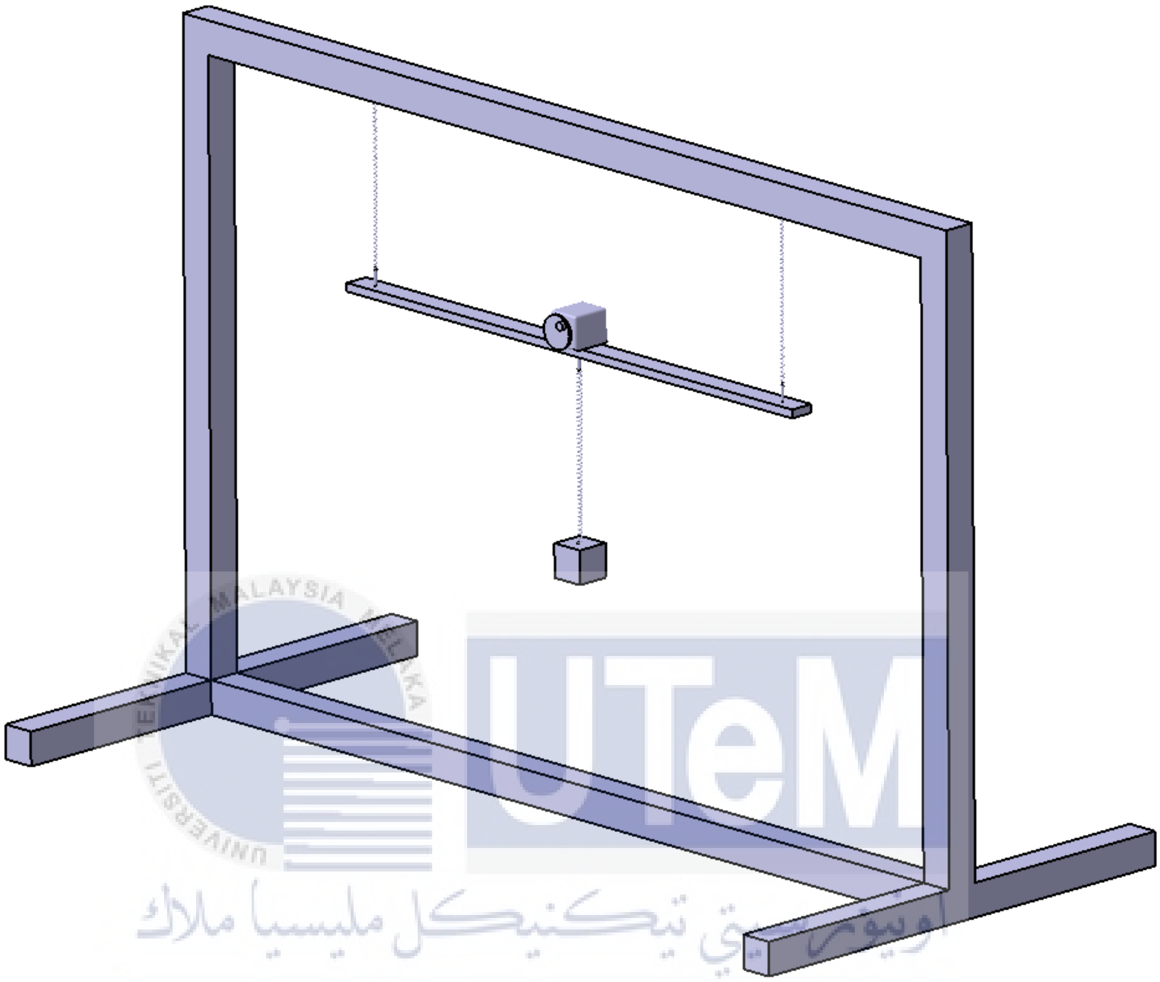


Figure 4.3: Conceptual design 3

Conceptual design 3 as shown in Figure 4.3 is a small scale frame without legs. The design will needed the user to place the test rig on the table or any high place to be used. The simplest design make it reduce the cost to develop the test rig. However, the user still need to find a place to put the test which will causing the problem if there are any thing that could provide a stable position for the test rig.Please refer ***Appendix D*** for further detailed design.

4.2 Pugh Metric Evaluation Method

The conceptual design is evaluated using Pugh metric evaluation method based on the project development specification. The highest of all characteristics of PDS will be selected as the final design.

Table 4.1: Pugh Matric Evaluation Table

Design / Criteria	Datum	1	2	3
Performance	0	+	+	+
Ease of Installation	0	-	1	+
Ease of Maintenance	0	1	1	+
Safety	0	+	1	1
Cost	0	-	+	+
Ergonomics	0	+	+	-
Materials	0	1	1	1
Sum of '+' s	0	3	3	4
Sum of '1' s	0	2	4	2
Sum of '-' s	0	2	0	1
Net Score	0	3	7	5
Rank	**0**	**3**	**1**	**2**

Based on the table 4.1, it is clearly shows that Conceptual design 2 from Figure 4.2 has the highest rank due to meet the criteria set compare to other design. Thus, the best selected design will go through developing and constructing process as it is already fulfill all the requirements to develop the vibration test rig.

4.3 Final design

The final design is selected according to the best result of the Pugh metric evaluation method. From the evaluation process, conceptual design 2 of **Figure 4.2** is selected as the final design to be developed.

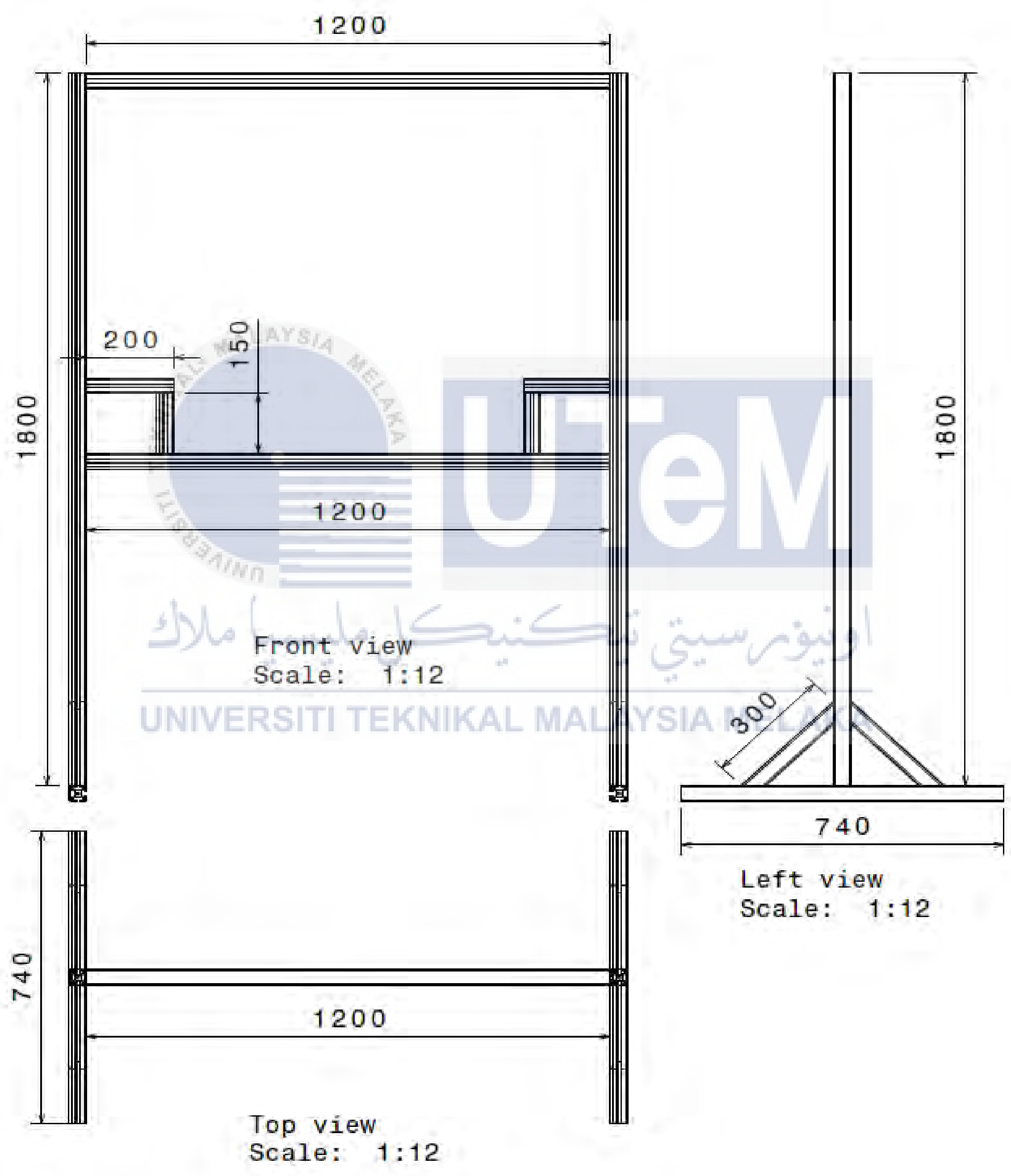


Figure 4.4: Detail design with dimension

4.4 Model design generation

The development of the model is to test the objective in the lower cost before proceed to develop the main vibration test rig. The development of the model which is smaller in scale and portable is good enough to be shown as a simple vibration test rig but it cannot go into more precise detail as the prototype to be developed later.

The development of the model is to justify the objectives, because if the model is failed, it means that the main product has a potential not to run accordingly. Both end of the beam are attached with the spring and the eccentric motor is placed at the center of the beam. The absorber consists of another spring and mass is attached below the center of the upper beam in order to balance the force that will be exerted by the absorber later on. The spring can be changed based on the stiffness to run the test and the mass can be added on the absorber to shows the different result of different masses and stiffness of the springs.

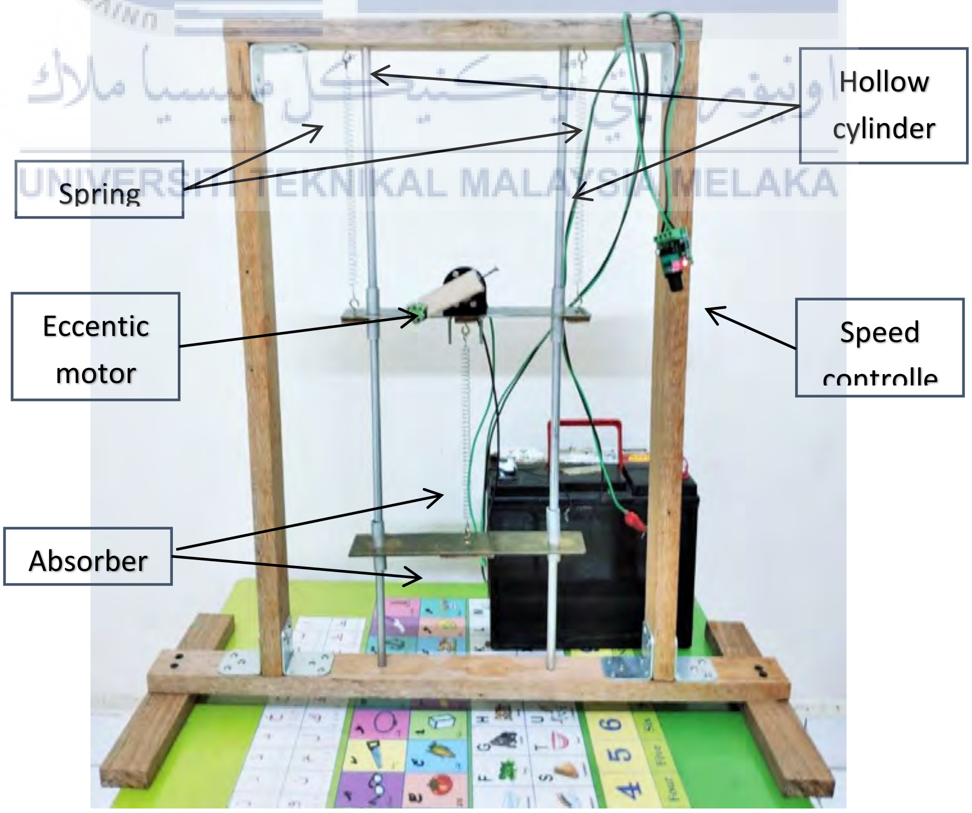


Figure 4.5: Model of vibration test rig

The hollow cylinder at both end of the beam is to support the movement of the beam as it is vibrating due to excitation force of the eccentric motor and to prevent the beam to run wild after the vibration force transfer to it. The cylinder will restrict the movement of the beam in horizontal direction which only allows the beam to move in vertical direction only. It is because the simple theoretical calculation of the mass-spring-damper is only in one axis. However, the cylinder will causing the friction on the beam and lowering the measurement of the displacement. Thus, the friction of the beam also need to be added into the calculation to justify the theoretical calculation and the experimental result to be not to vary of each other's.

4.5 Vibration test rig prototype

The development of the vibration test rig prototype involves of several processes from designing process to ordering process and constructing process which consists of cutting, grinding and fastening process. The process which involves in contributing the development process is very rough due to ensure the test rig follow the finalize design as planned to avoid any unnecessary problems likes unnecessary cost.

The design is follow thoroughly from its core which is the main frame to the parameter involves which is the beam and the spring in order to ensure the testing of the vibration test rig will be the same as the theoretical calculation.

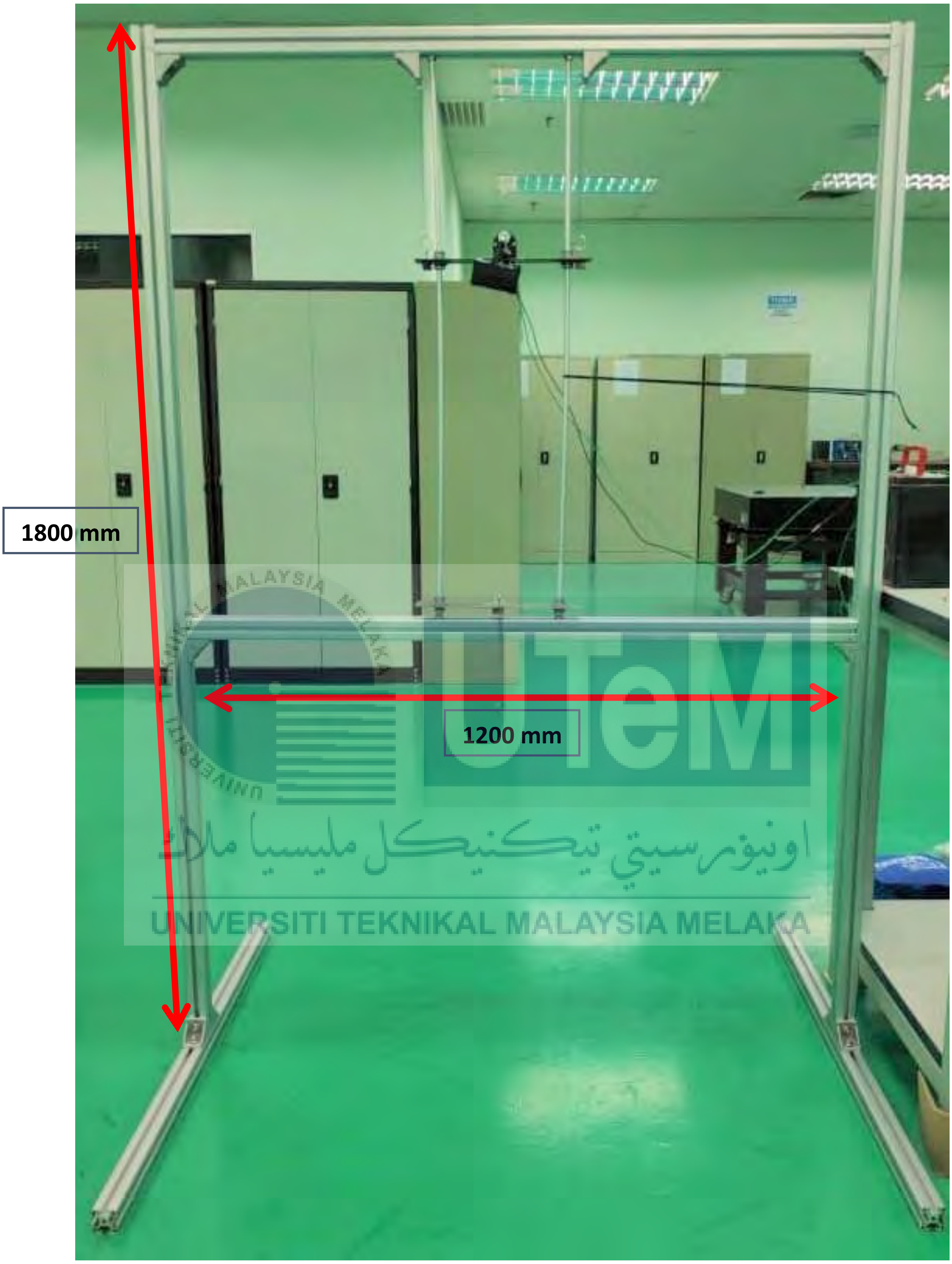


Figure 4.6: Vibration test rig's frame

The test rig is following the PDS which are ease of installation, ergonomics and build from the best material. The frame can be fastened by nuts and bolts which made it easy to be installed by the users. The suitable high of the test rig made it ergonomics in term of avoiding the back pain in order to use the test rig. The material is made from the aluminium which is anti-corrosion and high durability make it last longer.

Figure 4.7: Motor and the beam

The main component of the test rig is the motor and the beam itself. The motor used on the test rig is 12V DC motor which generate 531RPM at maximum 12V. The motor is attached at the centre of the beam which holding the imbalance mass of 0.12kg connected to the shaft using shaft coupling. The total mass of the beam and motor with imbalance mass is 0.7kg. Since the force generated is centrifugal force, some equipment is needed to restrict the movement of the beam in vertical axis only which is sliding bearing. The sliding bearing attached at both end will ensure the beam moving smoothly in vertical axis that supported by two hollow cylinder. The spring is attached besides the sliding bearing that will hold the beam and generate oscillation movement when the forced is applied by the motor. Please refer ***Appendix E*** for further detail on beam design.

Figure 4.8: Absorber and slotted mass

Figure 4.8 shows the absorber and the slotted mass used in the testing. The absorber mass is 0.325kg and the slotted mass is 0.150kg each. This absorber plays important role in the experiment as it is supposed to change the natural frequency of the system which will allow the system in widening its operating frequency range. The absorber also has two sliding bearings attached at both ends to restrict the movement in vertical axis only. Hook is attached at the centre of the absorber where the spring will be attached during testing and the other end of the spring will be attached directly under the motor.

4.6 Spring stiffness constant

The stiffness of the spring must be measure as it is one of the most important part and material in the testing and also calculation. The result is taken by using electronic scale which generate mass for each displacement of the extracted spring from its equilibrium position. The data is shown below.

Table 4.2: Measurement of stiffness of the spring

Displacement, x (m)	**Mass (Kg)**	**Force (N)**
0.01	0.045	0.4415
0.02	0.105	1.0301
0.03	0.165	1.6187
0.04	0.225	2.2073
0.05	0.280	2.7468
0.06	0.355	3.4826

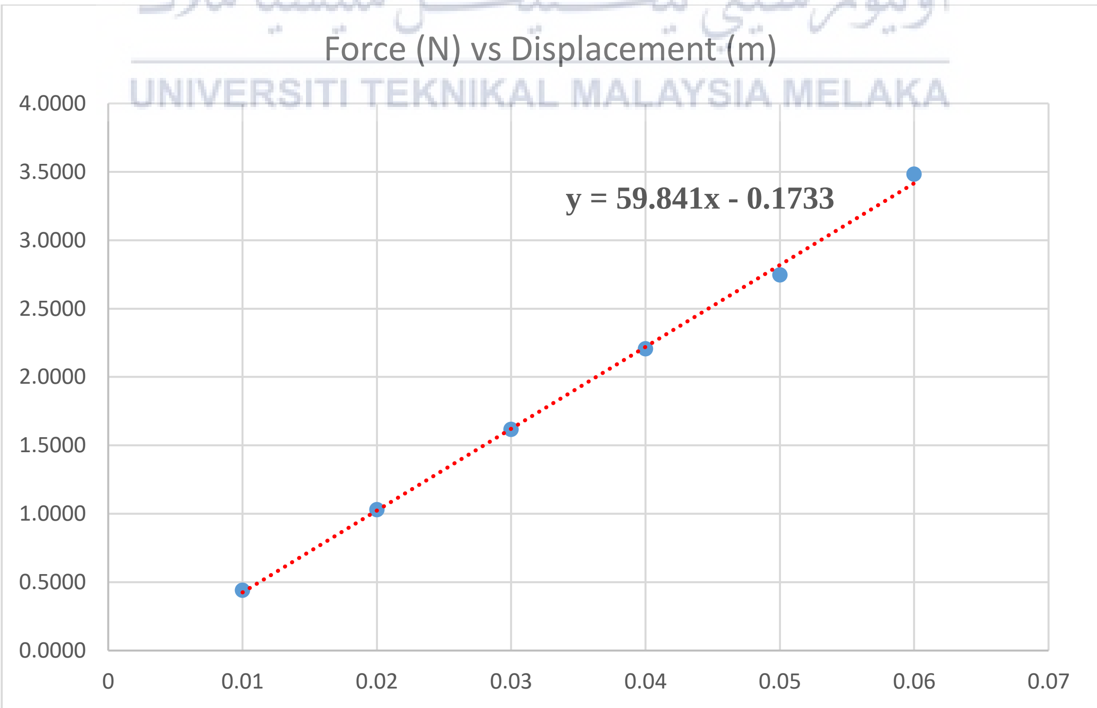


Figure 4.9: Stiffness of the spring

As shown in Figure 4.9, the point of the measurement displacement and force was plotted and the best fit line is generating the slope of the plotted point. Based on the equation on the graph, the value of the slope which is the stiffness of the spring, k is at 59.841 N/m. This means that force of 59.841 N is needed to extract the displacement of the spring for one meter.

4.7 Theoretical calculation

The theoretical calculation is calculated according to the condition of the set up test rig which is the vibration test without absorber and vibration test with the presents of absorber.

4.7.1 Calculation without absorber

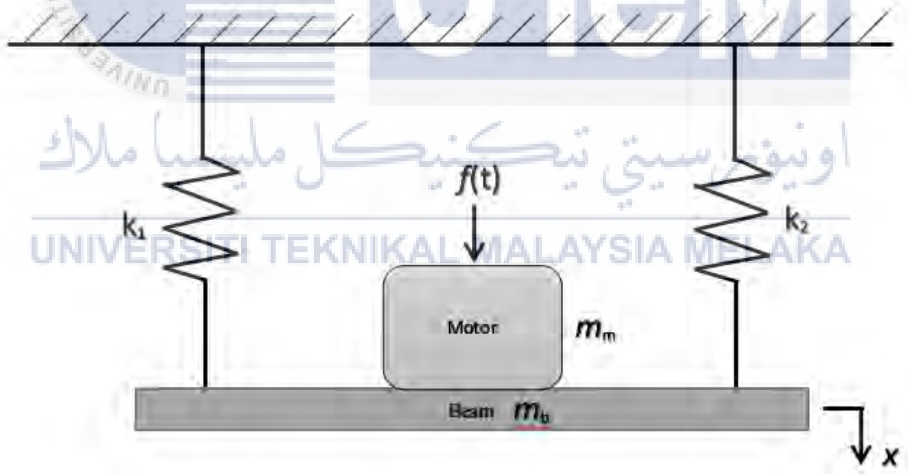


Figure 4.10: Vibration test rig diagram

$$m_b\ddot{x}+m_m\ddot{x}+k_1x+k_2x=f(t)$$
$$(m_b+m_m)\ddot{x}(t)+(k_1+k_2)x(t)=f(t)$$

$$f(t)=Fe^{j\omega t}$$

$$x(t)=Xe^{j\omega t}$$
$$\dot{x}(t)=j\omega Xe^{j\omega t}$$
$$\ddot{x}(t)=-\omega^2Xe^{j\omega t}$$

$$(m_b+m_m)(-\omega^2)X+(k_1+k_2)X=F$$
$$X=\frac{F}{(k_1+k_2)-\omega^2(m_b+m_m)} \tag{4.1}$$

4.7.2 Calculation with absorber

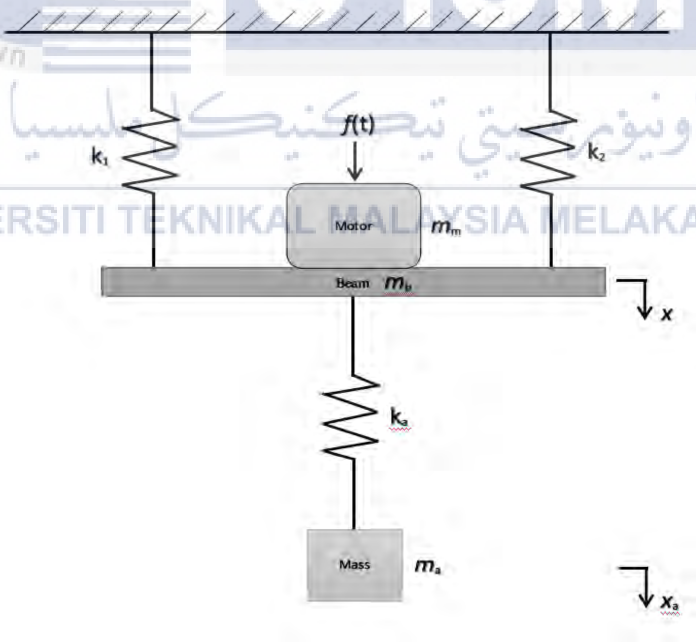


Figure 4.11: Vibration test rig with absorber diagram

$$m_b\ddot{x}+m_m\ddot{x}+k_1x+k_2x+k_a(x-x_a)=f(t)$$
$$(m_b+m_m)\ddot{x}(t)+(k_1+k_2+k_a)x-k_ax_a=f(t)$$

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

$$f(t)=Fe^{j\omega t}$$

$$x(t)=Xe^{j\omega t}$$
$$\dot{x}(t)=j\omega Xe^{j\omega t}$$
$$\ddot{x}(t)=-\omega^2Xe^{j\omega t}$$

$$\left[(k_1+k_2+k_a)-\omega^2(m_b+m_m)\right]X-k_aX_a=F$$

$$\left[k_a-\omega^2m_a\right]X_a-k_aX=0$$

$$\left(k_a-\omega^2m_a\right)\left[k_1+k_2+k_a-\omega^2\left(m_b+m_m\right)\right]X-k_a{}^2X=F\left(k_a-\omega^2m_a\right)$$

$$|X|=\frac{F\left(k_a-\omega^2m_a\right)}{\left[k_1+k_2+k_a-\omega^2\left(m_b+m_m\right)\right]\left(k_a-\omega^2m_a\right)-k_a{}^2} \tag{4.2}$$

4.8 Convert the vibration amplitude acceleration of g to the displacement of mm

The data of the testing was recorded in the amplitude acceleration of g. The data need to be convert to displacement of mm to ease the process of the comparison since the calculation of theoretical value is in displacement of mm. The vibration amplitude of g represent the acceleration from gravity which;

$$1g = 9.81 m/s^2$$

Thus, the data needed in order to convert from the acceleration is the speed of rotation of the motor. Below is one of the example of converting the vibration amplitude of g to displacement mm.

Table 4.3: Example calculation for conversion on amplitude unit

Rotating frequency, Hz	Amplitude g
2.130	1.1111

$$\begin{aligned} 2.130Hz &= 2.130 \times 2\pi \\ &= 13.3832 rad/s \end{aligned}$$

$$\begin{aligned} 1.111g &= 1.1111 \times 9.81 \\ &= 10.8999 m/s^2 \end{aligned}$$

$$\begin{aligned} displacement, x(m) &= \frac{10.8999}{13.3832^2} \\ &= 0.0609m \approx 61mm \end{aligned}$$

4.9 Testing result

The data from the test and experiment will be collected and displayed in the table and graph in order to ease the comparison value of the test and calculation.

4.9.1 Force vibration testing

Force vibration testing is done with the increasing of the speed of rotation of the motor or by increasing the value of operating frequency. The data and result of the testing is shown below.

Table 4.4: Result of force vibration testing

Rotating Frequency, Hz	RPM	Displacement X1, mm	Displacement X2, mm	Displacement X3, mm	Average Displacement X, mm
1.250	75.00	1.417	1.141	1.376	1.311
1.367	82.00	2.054	2.046	1.870	1.990
1.758	105.50	3.434	3.430	3.299	3.388
1.900	114.00	12.700	11.860	11.130	11.897
1.950	117.00	25.870	28.240	27.560	27.223
2.050	123.00	46.340	47.590	46.920	46.950
2.100	126.00	52.320	51.730	52.390	52.147
2.130	127.80	58.860	59.530	61.030	59.807
2.539	152.30	18.790	19.510	19.380	19.227
2.930	175.80	13.130	14.970	12.860	13.653
3.516	211.00	11.960	11.240	11.680	11.627
4.102	246.10	14.910	15.250	15.420	15.193
5.078	304.70	11.780	11.830	11.890	11.833
5.664	339.84	9.831	10.320	10.050	10.067

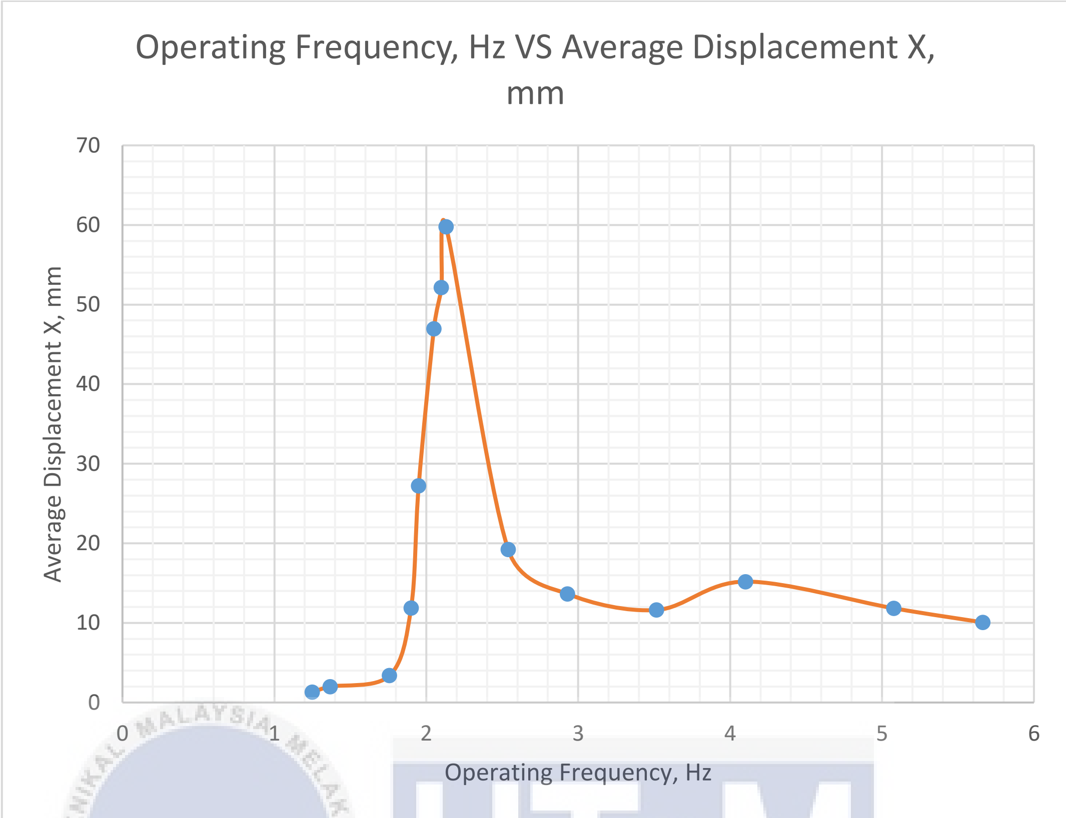


Figure 4.12: Graph of forced vibration testing

Based on the Table 4.4 and Figure 4.12, it shows the displacement of the beam at specific operating frequency. The average data from three readings is taken to ensure the data is reliable. The graph shows the line is changing from lower point then hit the highest point of 61mm at 2.13Hz which is the natural frequency of the system. After that, the line graph dropping and stay at constant displacement of about 11mm.

Example of the calculation of the forced vibration was shown below where the rotating frequency is 1.758 Hz, spring stiffness *k is* 59.841 N/m and total mass of beam and motor is 0.7 kg. The value of force will be calculated using the formula below where the mass of the imbalance is 0.12kg and radius of the imbalance to the center of the shaft is 0.065m

$$F = ma = m\left(\frac{v^2}{r}\right) = m\left(\frac{(\omega \times 2\pi \times r)^2}{r}\right)$$
$$= 0.12\left(\frac{(1.758 \times 2\pi \times 0.065)^2}{0.065}\right)$$
$$= 0.9517N$$

The information will be put into the eq. (4.1) to get the value of displacement of the beam.

$$|X| = \frac{F}{(k_1 + k_2) - \omega^2 (m_b + m_m)}$$
$$= \frac{0.9517}{(59.841 + 59.841) - 11.0458^2 (0.7)}$$
$$= 0.02777m \approx 27.77mm$$

Table 4.5: Comparison of testing and calculation forced vibration

Rotating Frequency, Hz	RPM	Testing, X [mm]	Calculation, X [mm]
1.250	75.00	1.311	6.292
1.367	82.02	1.990	8.461
1.758	105.48	3.388	27.792
1.900	114.00	11.897	55.892
1.950	117.00	27.223	80.368
2.050	123.00	46.950	368.125
2.100	126.00	52.147	612.004
2.130	127.80	59.807	243.994
2.539	152.34	19.227	33.934
2.930	175.80	13.653	22.481
3.516	210.96	11.627	17.149
4.102	246.12	15.193	15.004
5.078	304.68	11.833	13.391
5.664	339.84	10.067	12.881

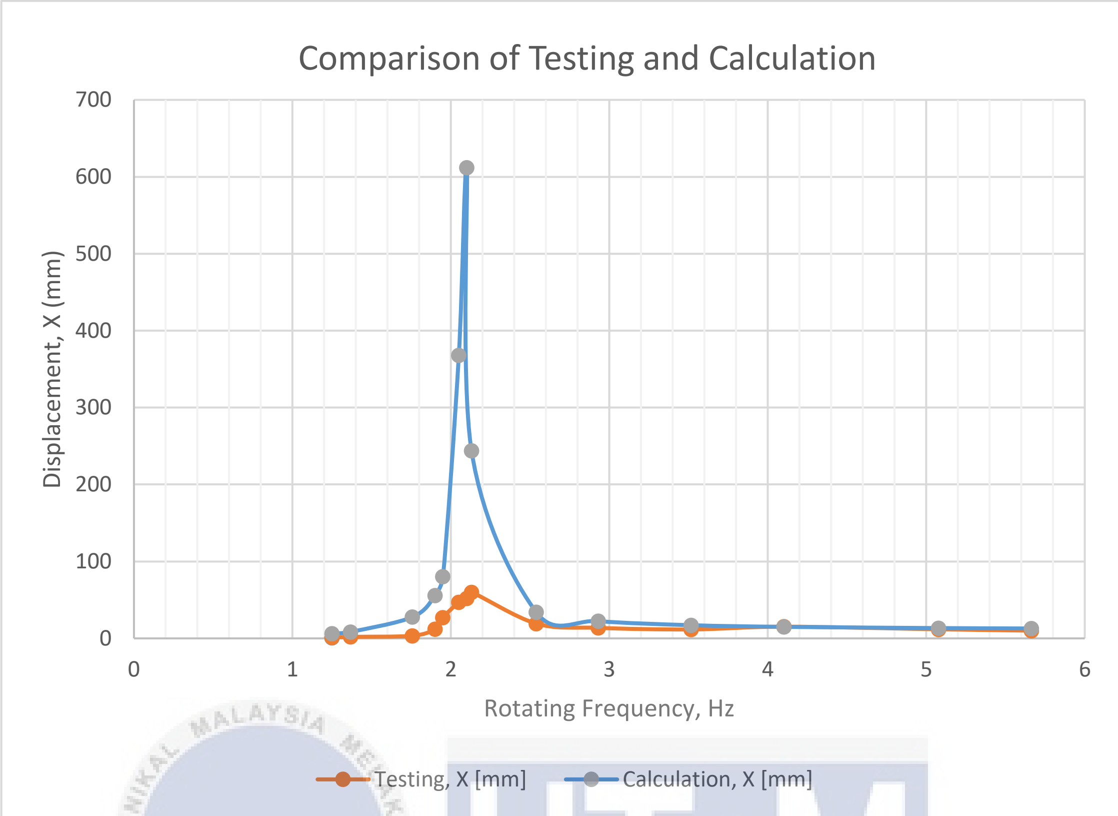


Figure 4.13: Comparison of testing and calculation forced vibration

By referring the Figure 4.13, the shape of both line graph is the same which has highest amplitude at about 2.13 Hz. The calculation line graph reached highest amplitude of 612mm or 0.61m, while the testing graph achieve the amplitude of 59mm or 0.059m. This is possible to be achieved by the testing graph because the length of the spring plays the significant factor. The length of the spring is 0.15m. Thus, it is possible for the spring to stretch until 0.61m due to its length limitation.

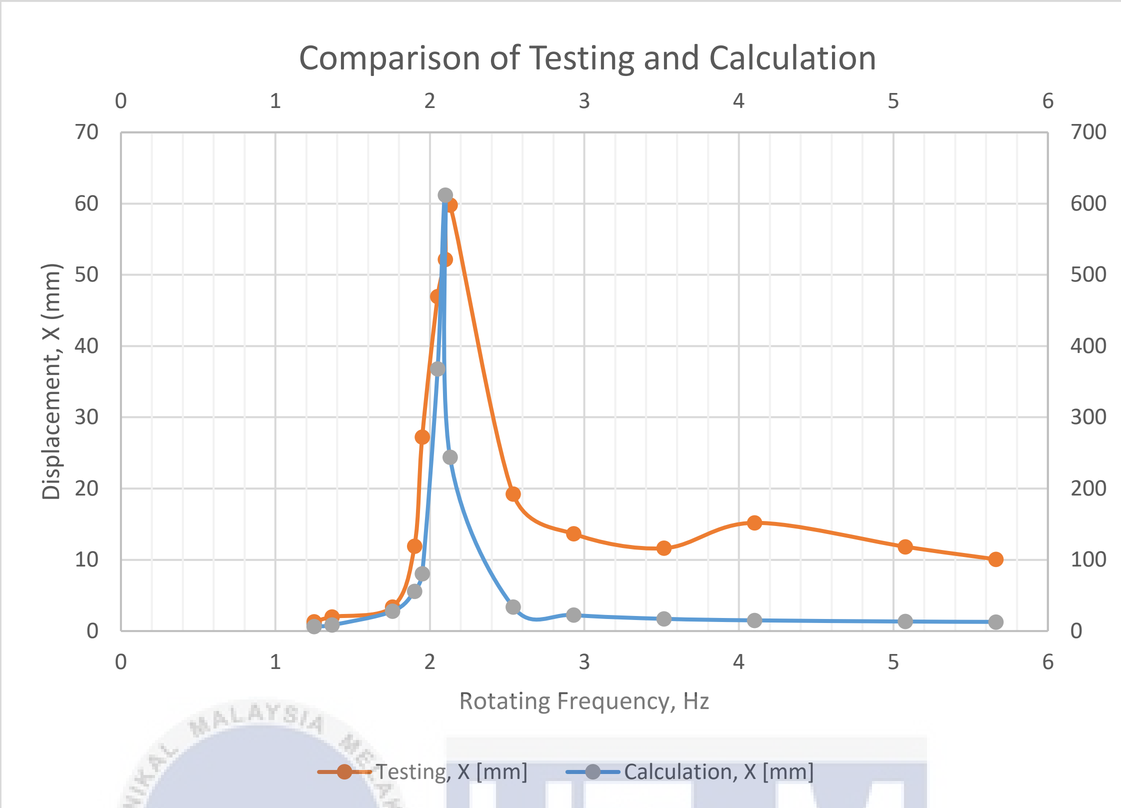


Figure 4.14: Comparison of testing and calculation forced vibration with secondary axis

Figure 4.14 shows the shape of the testing and calculation line graph after secondary axis is applied on the graph. Both line graph almost started and increased at the same rotating frequency, but the calculation line graph decrease first then follows by the testing line graph. From the result and graph, it could be said that the test rig is capable of generating the same graph for force vibration as in theory.

4.9.2 Force vibration with dynamic absorber testing

Force vibration with dynamic absorber testing is done with the absorber attached directly under the centre of motor. Then, the speed of rotation of motor is increased linearly and the data was recorded. The data and result of the testing is shown below.

Table 4.6: Result of force vibration with dynamic absorber testing

Rotating Frequency, Hz	RPM	Displacement X1, mm	Displacement X2, mm	Displacement X3, mm	Average Displacement X, mm
1.172	70.3	1.400	1.474	1.399	1.424
1.560	93.6	18.470	17.880	17.960	18.103
1.600	96.0	34.490	34.320	31.570	33.460
1.758	105.5	5.040	4.161	4.801	4.667
2.148	128.9	2.631	2.707	2.506	2.615
2.539	152.3	3.237	3.919	3.655	3.604
2.724	163.4	9.693	10.050	9.948	9.897
2.930	175.8	31.150	34.490	32.030	32.557
3.711	222.7	20.220	20.550	20.540	20.437
4.297	257.8	12.860	13.760	13.510	13.377
5.078	304.7	10.620	11.270	10.570	10.820

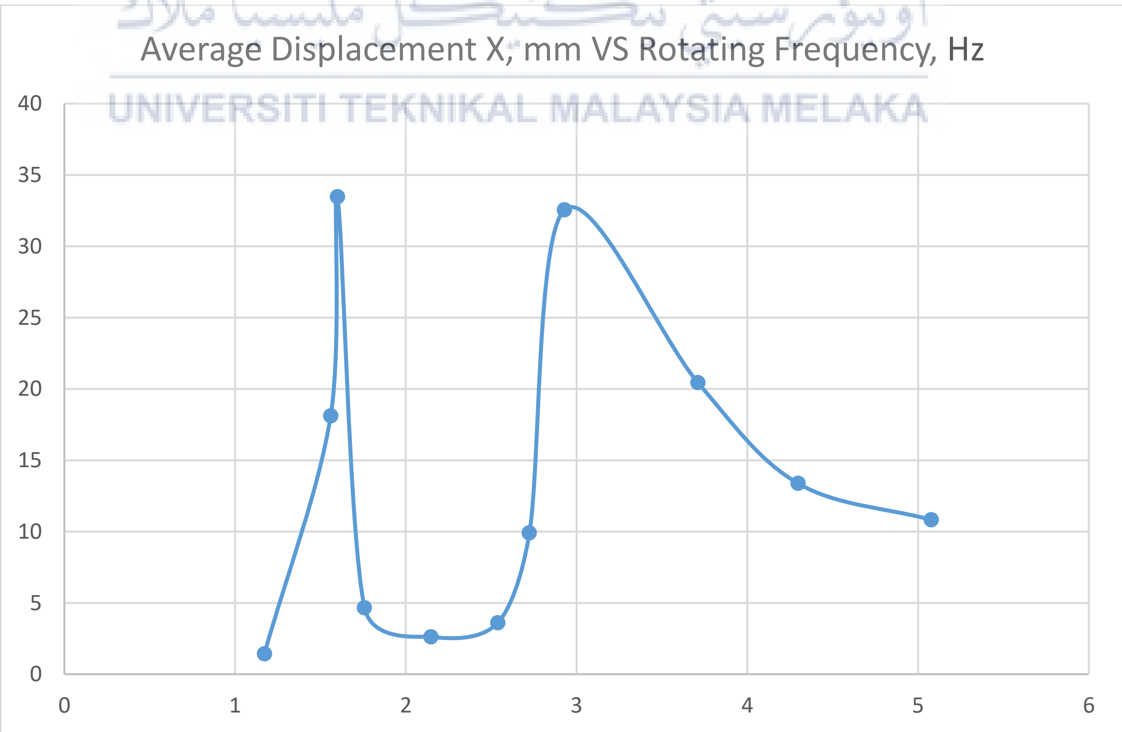


Figure 4.15: Graph of force vibration with dynamic absorber testing

The Table 4.6 and Figure 4.15 shows the displacement of the beam as the dynamic absorber is attached directly under the motor. The graph shows the decreasing of the value of the displacement at the previous natural frequency of the forced vibration which is at about 2.13Hz. As in theory, the absorber is causing the natural frequency to shift from its original places as it could manipulate the oscillation of the system by using different stiffness of the spring and mass of the absorber.

The most obvious trend in the graph is the significant reduce at the natural frequency. This will ensure the system could operate at such safe frequency and reduce the possibility of the resonance to occurs which could cause catastrophic failure to the system.

Example of the calculation of the forced vibration with absorber was shown below where the rotating frequency is 1.172 Hz, spring stiffness $k = k_a = 59.841$ N/m, total mass of beam and motor is 0.7kg and mass of the absorber is 0.325kg. The value of force will be calculated using the formula below where the mass of the imbalance is 0.12kg and radius of the imbalance to the center of the shaft is 0.065m

$$F = ma = m\left(\frac{v^2}{r}\right) = m\left(\frac{(\omega \times 2\pi \times r)^2}{r}\right)$$

$$= 0.12\left(\frac{(1.172 \times 2\pi \times 0.065)^2}{0.065}\right)$$

$$= 0.423N$$

The information will be put into the eq. 4.2 to get the value of displacement of the beam.

$$
\begin{aligned}
|X| &= \frac{F\left(k_a - \omega^2 m_a\right)}{\left[k_1 + k_2 + k_a - \omega^2\left(m_b + m_m\right)\right]\left(k_a - \omega^2 m_a\right) - k_a{}^2} \\
&= \frac{F\left(k_a - \omega^2 m_a\right)}{\left[k_1 + k_2 + k_a - \omega^2\left(m_b + m_m\right)\right]\left(k_a - \omega^2 m_a\right) - k_a{}^2} \\
&= \frac{0.423\left(59.841 - \left(7.3639^2\right)0.0.325\right)}{\left[3(59.841) - 7.3639^2(0.7)\right]\left(59.841 - \left(7.3639^2\right)0.325\right) - 59.841^2} \\
&= 0.00745m \approx 7.45mm
\end{aligned}
$$

Table 4.7: Comparison of testing and calculation forced vibration with dynamic absorber

Rotating Frequency, Hz	RPM	Testing, X [mm]	Calculation, X [mm]
0.750	45	-	1.806
0.900	54	-	2.945
1.172	70.3	1.424	7.459
1.520	81.0	-	240.758
1.560	93.6	18.103	58.031
1.600	96.0	33.460	32.944
1.758	105.5	4.667	11.417
2.148	128.9	2.615	0.251
2.539	152.3	3.604	12.587
2.724	163.4	9.897	30.230
2.930	175.8	32.557	197.930
3.711	222.7	20.437	24.846
4.297	257.8	13.377	18.307
5.078	304.7	10.820	15.272

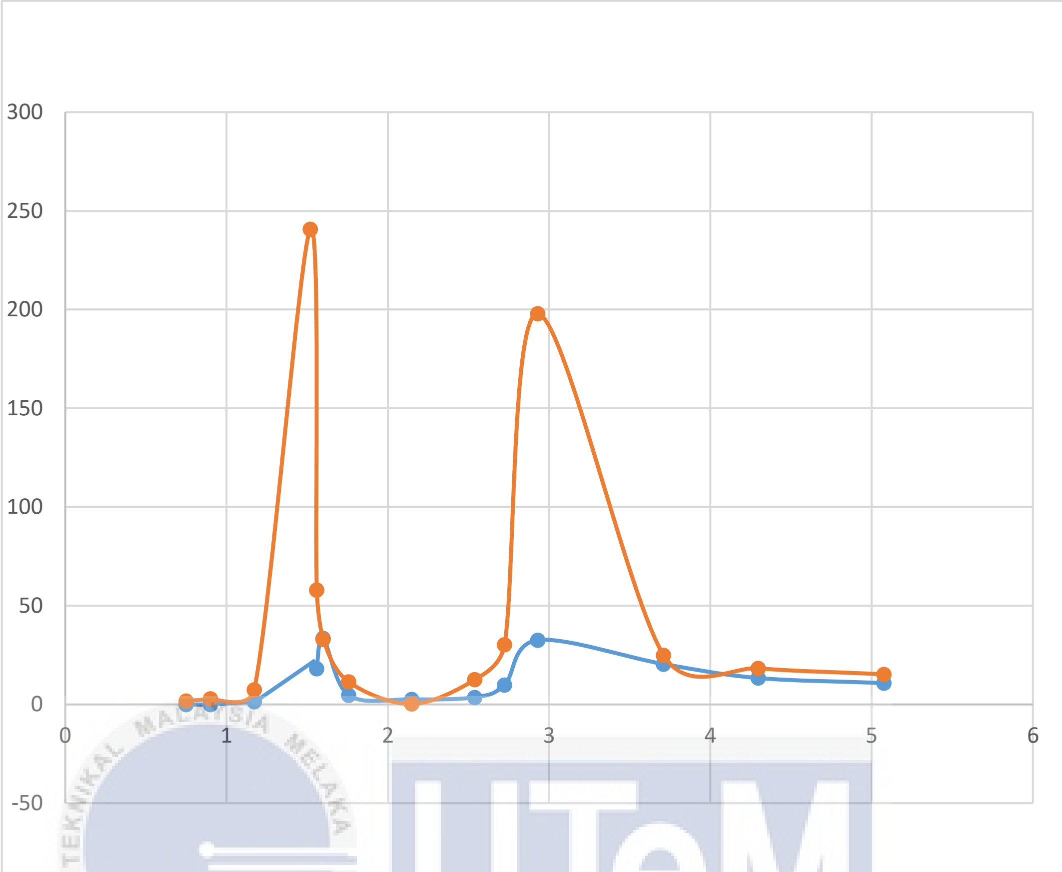


Figure 4.16: Comparison of testing and calculation forced vibration with dynamic absorber

Based on the Figure 4.16, It is clearly shows the difference between the theoretical calculation defining from the equation of mass-spring-damper diagram and the testing result. As mentioned before, the cause of such behaviour is due to the limitation in the length of the spring. The theoretical calculation could reach such high at about 1.4m without taking the length of the spring into the consideration. Some of the data is unavailable due to fault in the speed controller. Thus, some of the operating frequency was an assumption to get the better graph for the theoretical calculation.

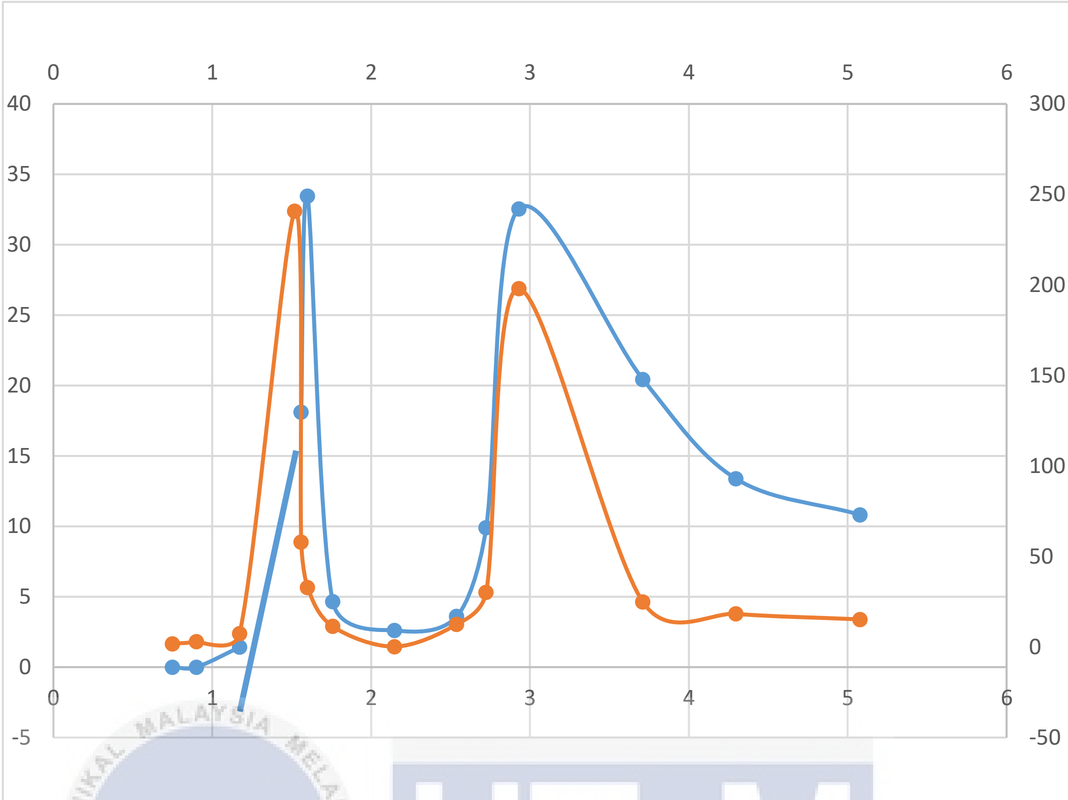


Figure 4.17: Comparison of testing and calculation forced vibration with dynamic absorber with secondary axis

The Figure 4.17 shows the graph after the secondary axis was applied to the graph. The shape of both testing result and theoretical calculation is almost the same. From the graph shows, the test rig is capable of producing the result of the forced vibration with dynamic absorber to such extend that it could has the same shape as in the theory of vibration. Some parameter could be taken into consideration as the theoretical calculation is assumed to be the ideal one. Some of parameter such as friction causing by the sliding bearing and centrifugal force that causing the system to move in horizontal axis slightly that reducing the force produce by the motor should be taken into calculation as it also can give different data based on different condition.

4.9.3 Transmissibility testing

The testing for transmissibility is done by using two sensors to detect the excitation force and the force transmitted. In this case, the force is taken in the form of the displacement of the beam and the base. The data was recorded in the acceleration amplitude of g that then convert into the displacement, mm.

Table 4.8: Result of transmissibility testing

Rotating Frequency, Hz	RPM	Displacement X1, mm	Displacement X2, mm	Displacement X3, mm	Average Displacement X, mm
1.563	93.78	1.324	2.135	1.938	1.7990
1.953	117.2	5.935	5.547	5.515	5.6657
2.140	128.4	20.135	21.890	19.843	20.6227
2.200	132.0	27.776	25.813	25.092	26.2270
2.340	140.4	30.778	29.087	28.447	29.4373
2.539	152.3	32.484	32.111	32.088	32.2277
2.734	164.0	30.815	30.373	29.533	30.2403
3.320	199.2	21.672	21.068	20.689	21.1430
4.102	246.1	14.604	15.428	14.842	14.9580
4.687	281.2	12.738	12.361	12.261	12.4533
5.078	304.7	11.531	12.059	12.138	11.9093

Rotating Frequency, Hz	RPM	Displacement Y1, mm	Displacement Y2, mm	Displacement Y3, mm	Average Displacement Y, mm
1.563	93.78	0.0043	0.0038	0.0035	0.0039
1.953	117.2	0.0083	0.0145	0.0082	0.0103
2.140	128.4	0.0392	0.0413	0.0349	0.0385
2.200	132.0	0.0452	0.0386	0.0348	0.0395
2.340	140.4	0.0567	0.0554	0.0521	0.0547
2.539	152.3	0.0635	0.0638	0.0633	0.0635
2.734	164.0	0.0765	0.0835	0.0792	0.0797
3.320	199.2	0.0870	0.1115	0.0940	0.0975
4.102	246.1	0.0860	0.1076	0.0789	0.0908
4.687	281.2	0.1430	0.1385	0.1413	0.1409
5.078	304.7	0.3287	0.1570	0.0889	0.1915

Rotating Frequency, Hz	RPM	X/Y
1.563	93.78	465.2586
1.953	117.2	548.2903
2.140	128.4	536.1179
2.200	132.0	663.4148
2.340	140.4	537.8319
2.539	152.3	507.2560
2.734	164.0	379.2684
3.320	199.2	216.8513
4.102	246.1	164.6752
4.687	281.2	88.3633
5.078	304.7	62.1789

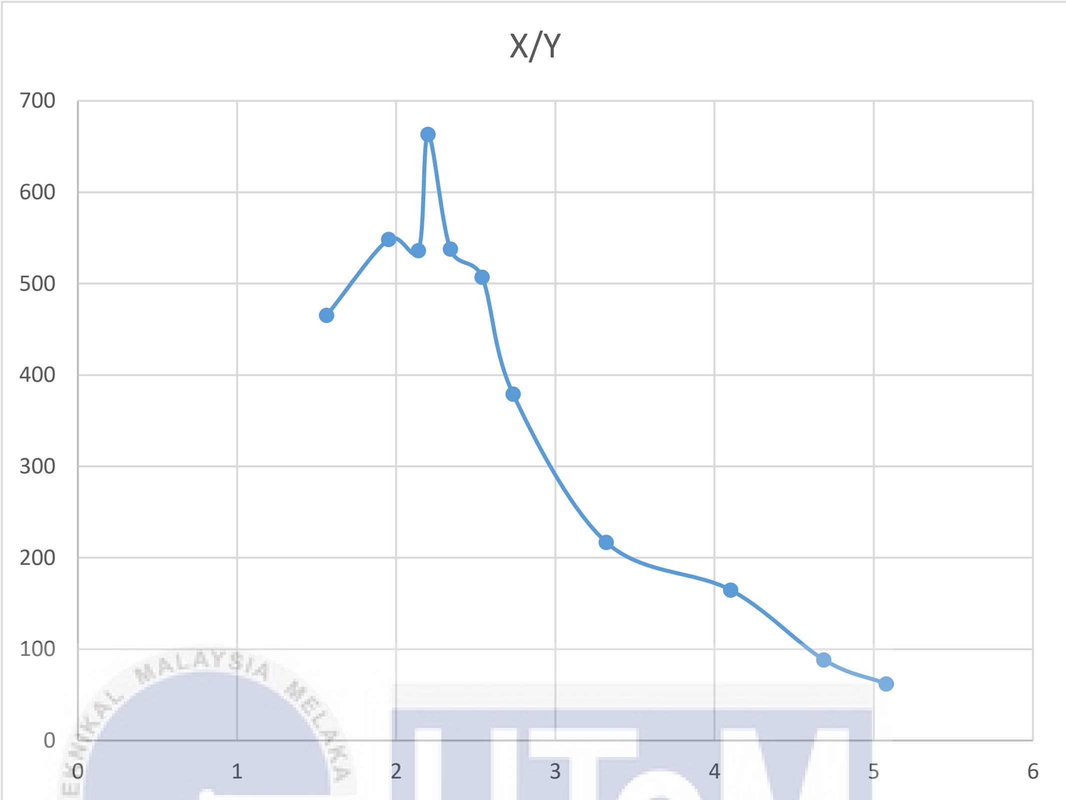


Figure 4.18: Graph of transmissibility testing

From Table 4.8 and Figure 4.18, it shows the testing result of the transmissibility testing. The highest amplitude was recorded at the frequency of 2.2 Hz which has the ratio of 663.4148. Then, the line graph start to drop after that which is almost the same as the theoretical graph of transmissibility. However, the result need to be compared with the theoretical calculation of the same condition as in the testing to ensure if the value is valid.

Table 4.9: Comparison of testing and theoretical calculation Transmissibility

Rotating Frequency, Hz	RPM	Testing, X/Y	Calculation, T
0	0	-	1.000
1.563	93.78	465.2586	2.167
1.953	117.2	548.2903	6.278
2.140	128.4	536.1179	106.250
2.200	132.0	663.4148	14.968
2.340	140.4	537.8319	4.833
2.539	152.3	507.2560	2.376
2.734	164.0	379.2684	1.544
3.320	199.2	216.8513	0.700
4.102	246.1	164.6752	0.369
4.687	281.2	88.3633	0.260
5.078	304.7	62.1789	0.214

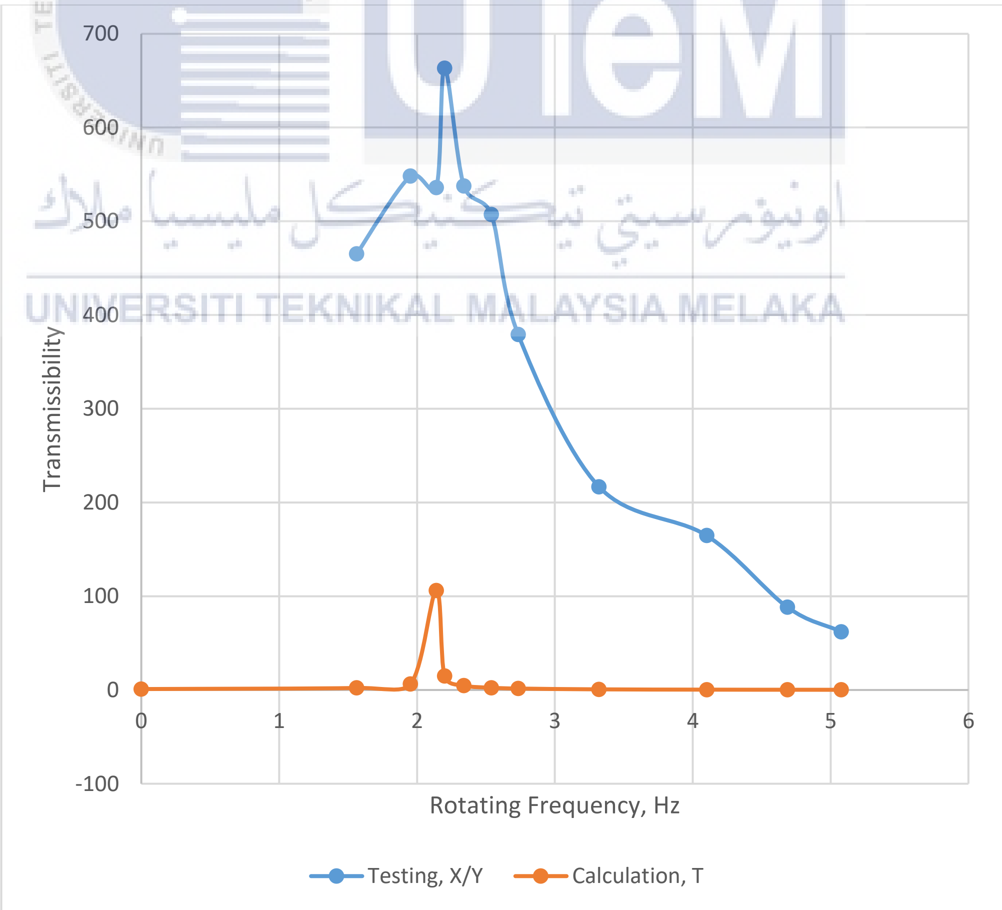


Figure 4.19: Comparison of testing result and calculation of transmissibility

Based on Figure 4.19, it shows a great different between testing result and theoretical calculation even-though the same condition was applied on the system. The transmissibility ratio of the theoretical calculation has lower value which only achieve the highest amplitude of 106.25 at 2.14 Hz while the testing result has the highest amplitude of ration at 663.4148 at 2.2 Hz.

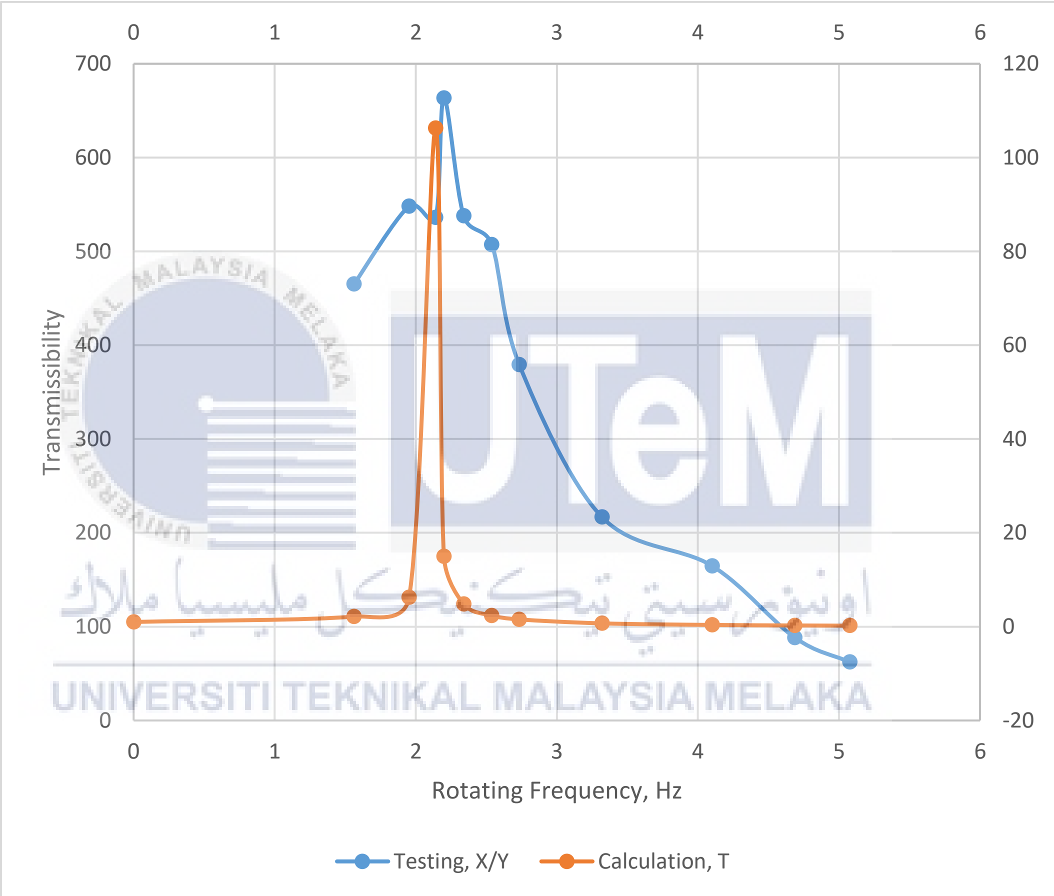


Figure 4.20: Comparison of testing result and calculation of transmissibility with secondary axis

Figure 4.20 shows the shape of the line graph after the secondary axis is added to the graph. The graph still shows excessively different in the transmissibility ratio. Even-though many factors could be the reason of the testing result is significantly different from the theoretical calculation, it could be concluded that the test rig is not capable to generate the best transmissibility ratio between the excitation force and the transmitted force. One of the reasons because of the stiffness of the spring as the one in the theory shows how the base has high stiffness to support the excitation force applied on the system.

4.10 Overall result discussion

Based on the result of the testing which is then to be compared to the theoretical calculation shows that the test rig is capable of generating the result of the same shape as in the theoretical calculation. One of the reasons that the graph of force vibration with and without absorber have low amplitude compared to the theoretical calculations is due to the limited length of the spring used. The theoretical calculation is the ideal value of the system could achieved where many parameters and factors are not included like the spring's length and the loss of force generated due to friction force on the system. The friction force could come from the centrifugal force created by the motor where it will push the sliding bearing onto the surface of hollow cylinder. Thus, friction force was created. The sliding bearing is used to restrict the movement of the motor to move only in vertical direction will initiate friction force which will reduce the force on the motor. The friction force should be taken into measurement to get more precise data to be compared to the theoretical calculation.

There are also problem on the force generated by the motor which the imbalance of the rotation will give an effect to the amplitude of the vibration. This factor should be taken into consideration because the peak of the amplitude spectrum will get very high on the rotating frequency of the motor due to imbalance of the motor. The design of the beam also play an important role where the beam should be balanced in all directions. The beam should be design in precise manner to ensure the testing result will be reliable and valid. The imbalance of beam will cause the vibration to unstable as the motor generating the force where both end of the beam will not move synchronously. This also will affect the data and result from the testing. Many parameter should be considered since the test rig is not an ideal as the theoretical calculation.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

Throughout the development process of the test rig, a lot of knowledge has been gained and every problem can be overcome from trial and error and by referring past research as medium to solve the problems. The conceptual design generated by CATIA V5 to achieve each product design specification took a lot of time as the main frame should support the whole system while be able to last longer and ergonomics to ease the student of using the test rig.

The development process of prototype and model of the test rig do took a lot of time due to lack of experiences in the material field. The correct equipment need to be selected in order to has the test rig to run smoothly and be able to achieve the objectives of this project. Consultation with supervisor and lecturers does help in overcome the barrier of the problem which solved the problem of the test rig not be able to move vertically. The testing of the test rig shows great result in having the same shape as one in the theory. This achievement could help student in their studies and help them learning how the theory and concept will really work in the real world.

More over, the difference in the theoretical calculation and testing result should be solved by having a few improvement on the test rig. Thus, the recommendation of this project should be to improvise the efficiency of the test rig and let the data be more accurate. Friction force and the irregularities of the beam movement should be considered in the calculations and enhance the data to be more accurate and precise. The motor can be changed to AC motor to let more variable in the speed of rotation which has higher torque than the DC motor. The torque of the motor is really important as it will support the imbalance mass of the motor. Low torque motor will produce unequal speed of rotation of motor thus effect the system. Besides, AC motor is more reliable and can be controlled more easily rather than DC motor which need to be controlled by voltage controller to control the speed of the rotation.

The design of the test rig also involves in designing the vibration of the beam where student can run the test of the vibration of the beam. The test rig should be improved in order for the beam vibration testing to be able to run on the test rig as the current design only support the place to hold the beam. After the improvement, the test rig should be able to run many type of different vibration testing thus help student to grasp the interest in learning the vibration itself.

REFERENCES

Alex Y. Tuan, G. Q. Shang, "*Vibration COntrol in a 101-Storey Building Using a Tuned Mass Damper*", 2014, Department of Civil Engineering, Tamkang University, Taiwan.

Chaudhari Arati G., *et al.,* 2017. A Review of Design of Shock Absorber Test Rig. *International Research Journal of Engineering and Technology* (03), pp. 2136 - 2140

Jaini N., 2014. *Vibration Analysis of a Beam Structure Attached with Two Dynamic Vibration Absorber*, Universiti Tun Hussein Onn, Malaysia.

Krunal C. Chaudhari, Shantanu J. Chaudhari, 2017. A Test Rig for Evaluation and Assessment of Vibration Isolators by Measuring Torque Transmissibility. *International Journal for Scientific Research and Development* (08), pp. 138 - 142

Nikhil S. Kothawade, *et al.*, 2014. Design of Shock Absorber Test Rig for Measurement and Analysis of Transmissibility. *International Research Journal of Engineering and Technology* (01), pp. 2827 - 2832

Nikhil T., Chandrahas T., Chaitanya C., Sagar I., Sabareesh G. R., 2015. Design and Development of a Test-Rig for Determining Vibration Characteristics of a Beam. *Science Direct Procedia Engineering* 144 (2016), pp. 312 – 320

Rutuja D., Sumant S. P., Amol K., "*Design and Tranmissibility Investigation of Shock Absorber Test Rig*", August 2017, Department of Mechanical Engineering, Hadapsar, Pune.

S. Graham Kelly, 2012. *Mechanical Vibration; theory and application*, The University of Akron, United States: CENGAGE Learning.

Shyh-Chin H., Kao-An L., A New Design of Vibration Absorber for Periodic Excitation. *Shock and Vibration* (2014), http://dx.doi.org/10.1155/2014/571421

Shorya Awtar, Alexander H. Slocum, Edip Sevincer, "*Characteristics of Beam-Based Flexure Modules*", December 2005, Massachusetts Institute of Technology, Cambridge, MA.

Singiresu S. R., "*Vibration of Continuous Systems*", March 2007, Department of Mechanical Engineering, University of Miami, Florida. John Wiley & Sons Inc.

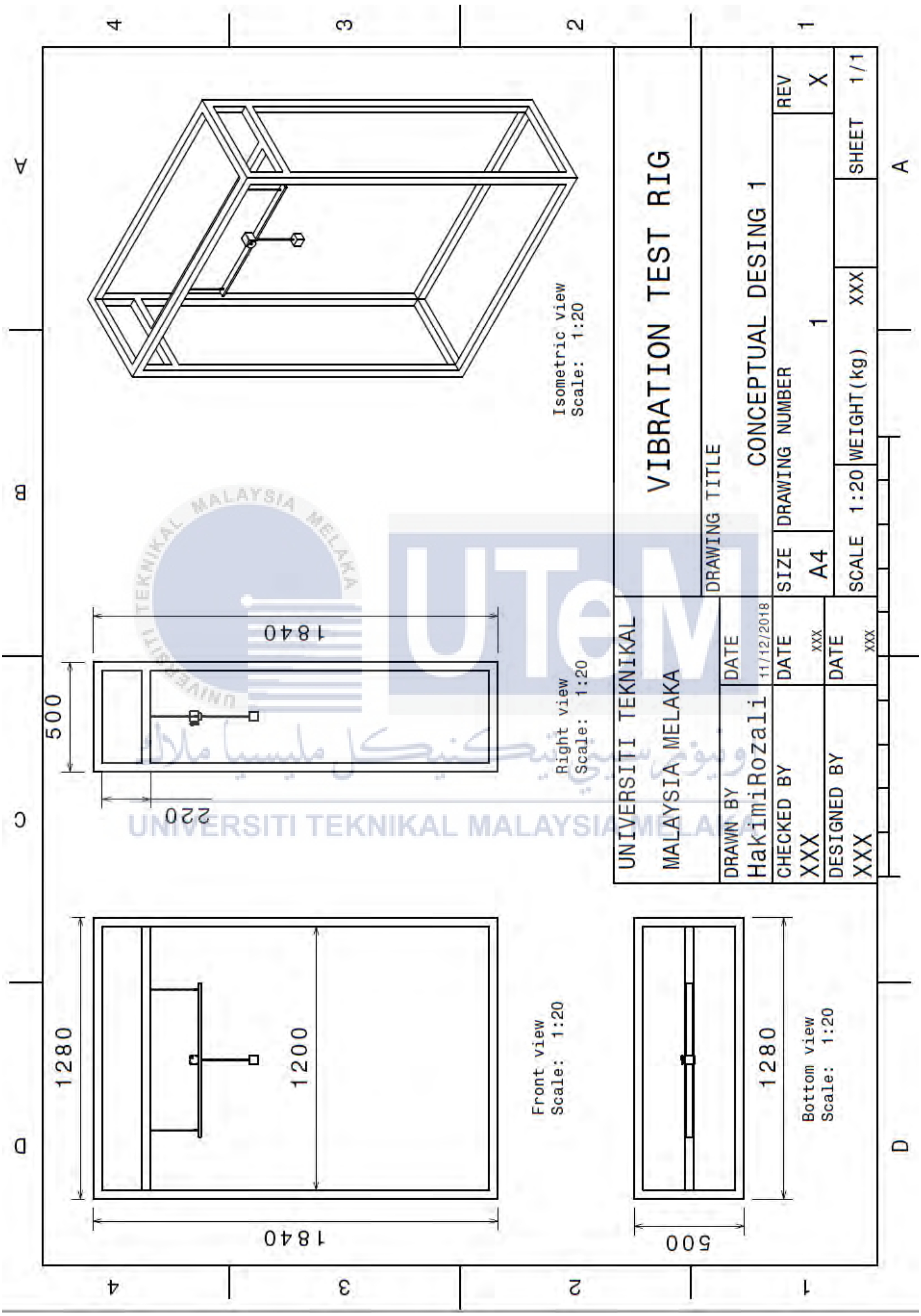
Y.E. Lage, *et al.*, Force transmissibility versus displacement transmissibility, *Journal of Sound and Vibration* (2014), http://dx.doi.org/10.1016/j.jsv.2014.05.038

Appendix A

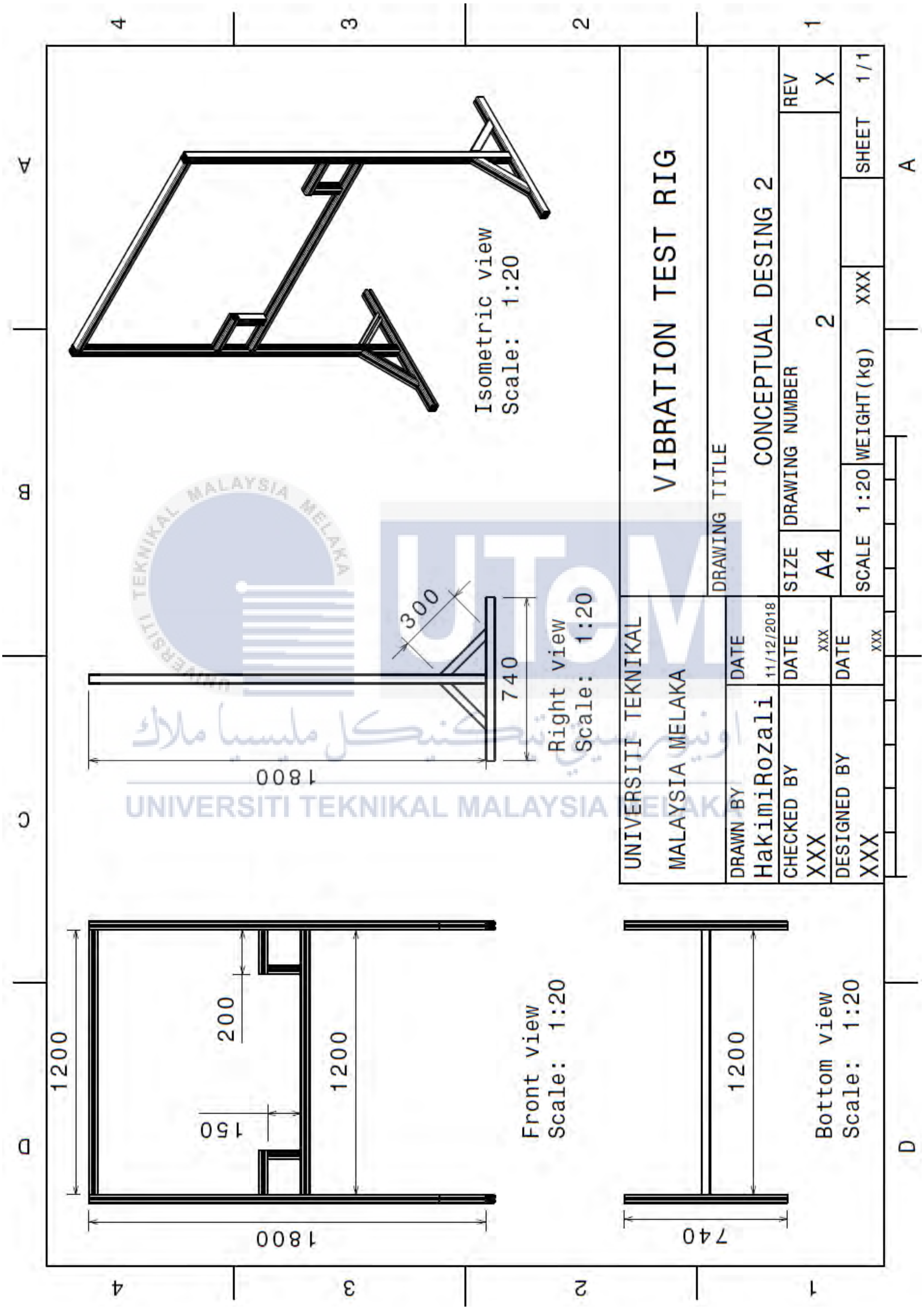
Gantt chart

	Activities	Week															
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
PSM 1	Title Discussion																
	Literature review																
	Designing process																
	Selection design																
	Detail design																
	Prototype construction																
	Report writing																
	Report submission																
	Presentation																
PSM 2	Test rig development																
	Laboratory testing																
	Theoretical calculation																
	Testing result analysis																
	Finalizing comparison of theory and testing analysis																
	Report writing																
	Report submission																
	Presentation																

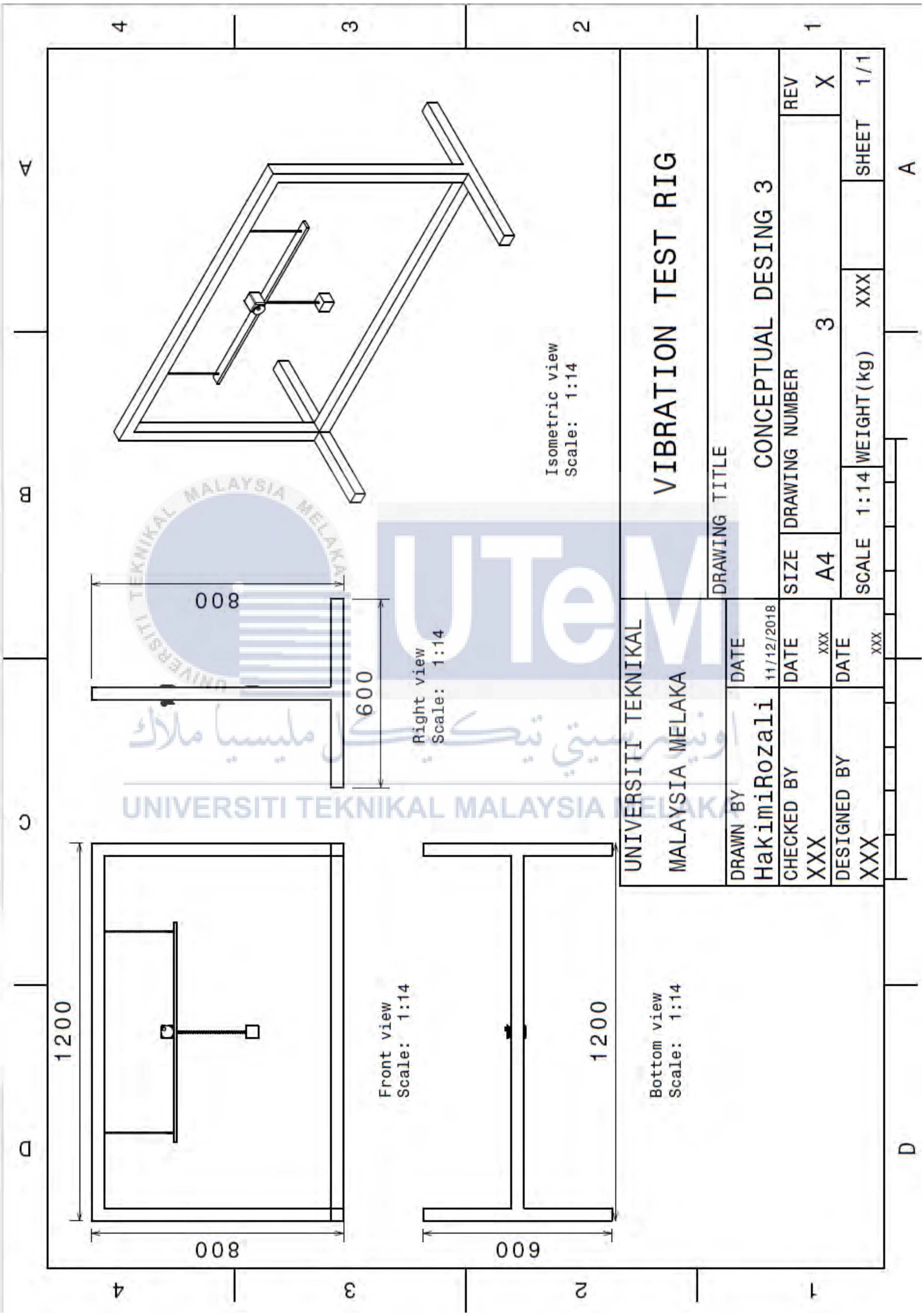
Appendix B


VIBRATION TEST RIG
DRAWING TITLE
CONCEPTUAL DESING 1
DRAWING NUMBER
1
SIZE
A4
SCALE 1:20
WEIGHT (kg) XXX
SHEET 1/1
REV
X
UNIVERSITI TEKNIKAL MALAYSIA MELAKA
DRAWN BY
HakimiRozali
DATE
11/12/2018
CHECKED BY
XXX
DATE
XXX
DESIGNED BY
XXX
DATE
XXX
Isometric view
Scale: 1:20
Right view
Scale: 1:20
Front view
Scale: 1:20
Bottom view
Scale: 1:20
1840
500
220
1280
1200
1840
1280
500

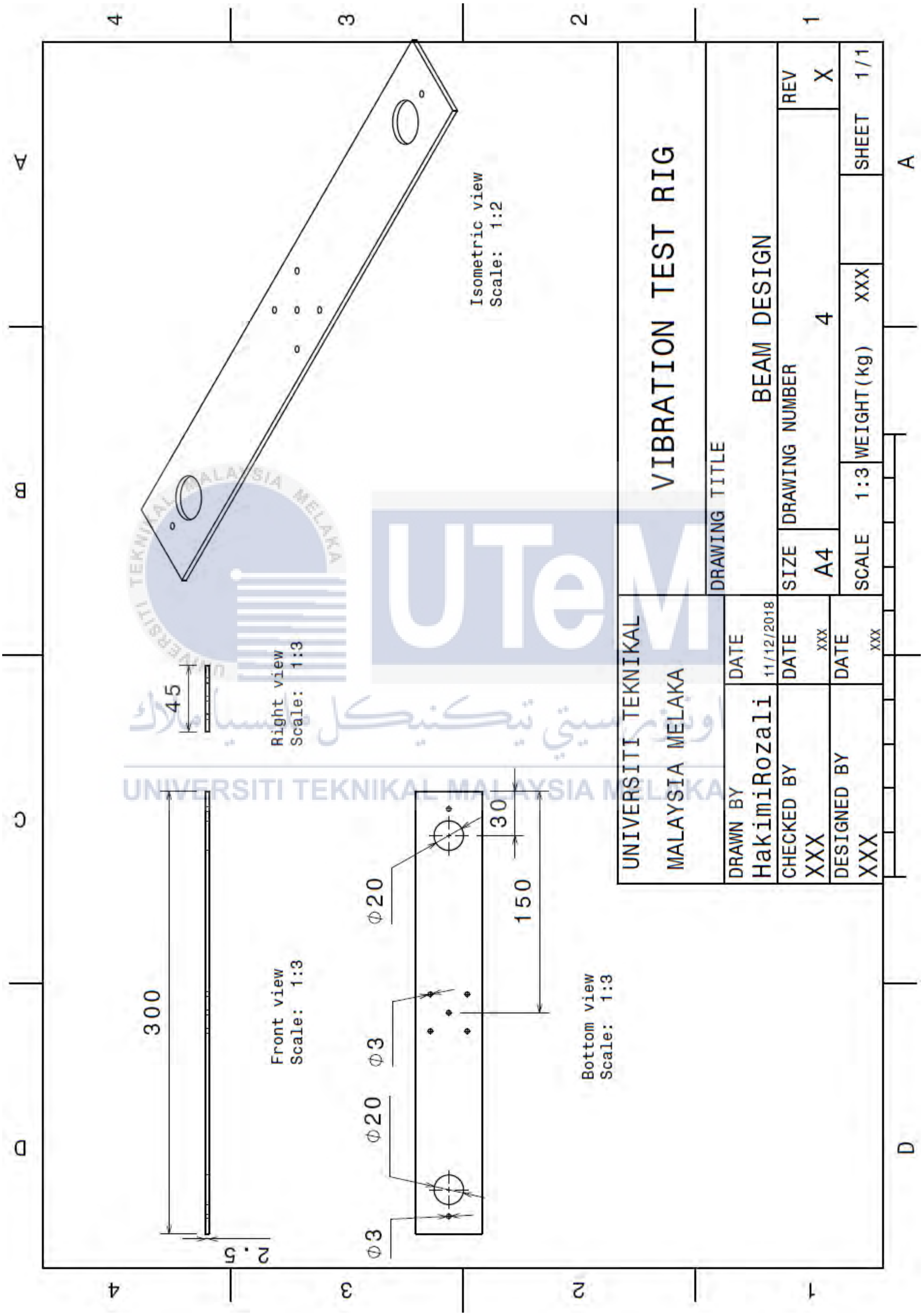
Appendix C


Isometric view
Scale: 1:20
VIBRATION TEST RIG
DRAWING TITLE
CONCEPTUAL DESING 2
REV
X
SHEET
1/1
DRAWING NUMBER
2
WEIGHT(kg)
XXX
SIZE
A4
SCALE
1:20
UNIVERSITI TEKNIKAL
MALAYSIA MELAKA
DRAWN BY
HakimiRozali
DATE
11/12/2018
CHECKED BY
XXX
DATE
XXX
DESIGNED BY
XXX
DATE
XXX
300
740
1800
Right view
Scale: 1:20
1200
200
150
1200
1800
Front view
Scale: 1:20
1200
740
Bottom view
Scale: 1:20

Appendix D


VIBRATION TEST RIG
UNIVERSITI TEKNIKAL
MALAYSIA MELAKA
DRAWING TITLE
CONCEPTUAL DESING 3
DRAWN BY
HakimiRozali
DATE
11/12/2018
CHECKED BY
XXX
DATE
XXX
DESIGNED BY
XXX
DATE
XXX
SIZE
A4
DRAWING NUMBER
3
REV
X
SCALE 1:14
WEIGHT(kg)
XXX
SHEET 1/1
Isometric view
Scale: 1:14
Right view
Scale: 1:14
Front view
Scale: 1:14
Bottom view
Scale: 1:14
800
600
1200
800
1200
600

Appendix E


VIBRATION TEST RIG
UNIVERSITI TEKNIKAL MALAYSIA MELAKA
DRAWING TITLE
BEAM DESIGN
DRAWN BY
HakimiRozali
DATE
11/12/2018
CHECKED BY
XXX
DATE
XXX
DESIGNED BY
XXX
DATE
XXX
SIZE
A4
DRAWING NUMBER
4
REV
X
SCALE
1:3
WEIGHT(kg)
XXX
SHEET
1/1
Isometric view
Scale: 1:2
Right view
Scale: 1:3
45
Front view
Scale: 1:3
300
2.5
Bottom view
Scale: 1:3
Φ20
Φ3
Φ20
Φ3
30
150