BALANCING EQUIPMENT FOR HIGH RPM PULLEY

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This report is submitted
in fulfilment of the requirement for the degree of
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Faculty of Mechanical Engineering

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DECLARATION

I declare that this report entitled *"Balancing Equipment for High RPM Pulley "* is the result of my own research except summaries and quotations which have been acknowledged. The report has not been accepted for any other degree and is not concurrently submitted in candidature of any other degree.

Signature : ....

Name : Abdul Hakam Bin Syed Syamsudin

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 (Plant and Maintenance).

Signature	:	....
Name of Supervisor	:	Dr. Reduan Mat Dan
Date	:	……………………………....

DEDICATION

I would like to dedicate to

My father,

SYED SYAMSUDIN BIN SYED AHMED

My mother,

ZAHARAH BT AHMAD

My supervisor,

DR. REDUAN MAT DAN

and

All my friend,

for their assistances & supportive efforts.

ABSTRACT

Unbalanced mechanical systems are always be the problem to engineers because of its impact can distrupt the smoothness and the reliability of a system. From the past, unbalance is known as the most causes of machine vibration and cause more vibration and generates excessive force in bearing area and will reduces the life of machine used. In oder to expert and understand the unbalance characteristics of these pulley, a pulley balancer equipment that can stand untill to 3000 RPM speed of motor must be designed in this project. In balancing the pulley, method of mass addition and vibration measurement technique are used. The design must consist the suitable material and strong to it stand untill the maximum speed. The result outcome the increasing of speed untill to 300 RPM without vibrate the pulley and unbalanced the pulley. As conclusion, the pulley must be designed by using the right method, requirement and properly assemble. Therefore helps in monitoring the health of rotating component where could be useful to prevent the breakdown.

ABSTRAK

Ketidakseimbangan sistem mekanikal sentiasa menjadi masalah kepada para jurutera kerana kesannya boleh menganggu kelancaran kebolehpercayaan sistem itu. Dari masa lalu, ketidakseimbangan dikenali sebagai punca utama getaran mesin dan menyebabkan lebih banyak getaran dan menghasilkan daya yang berlebihan di kawasan galas dan akan mengurangkan kehidupan mesin yang digunakan. Untuk pakar dan memahami ciri-ciri ketidakseimbangan takal ini, peralatan pengimbang takal yang bole bertahan sehingga 3000 RPM kelajuan motor mestilah direka dalam projek ini. Dalam mengimbangi takal, kaedah penambahan berat dan teknik pengukuran getaran digunakan. Reka bentuk mestilah terdiri daripada bahan yang sesuai dan kuat untuk bertahan sehingga kelajuan maksimum. Hasilnya adalah peningkatan kelajuan sehingga 3000 RPM tanpa gegaran takal dan ketidakseimbangan takal. Sebagai kesimpulan, takal mesti direka dengan menggunakan kaedah, keperluan dan pemasangan yang betul. Selain daripada itu, membantu dalam pemantauan kesihatan komponen berputar di mana berguna untuk mencegah pecahan.

ACKNOWLEDGEMENT

In the name of Allah, most Gracious, most Compassionate. I would like to express my full gratitude to Allah for granting me the best condition of health and strength in completing my final year project. Special thanks to my supervisor, Dr. Reduan Mat Dan for his moral supports, commitment to the students and continuous guidance in helping completing my final year project in these two semesters. Besides that, special appreciation to the technical staff Puan Hidayah for his commitment in helping by using the laboratory equipment. I also would like to appreciate to all my friends for giving me a full assistance and additional knowledge in helping me to complete my research. Not to forget my beloved mother and father who have encouraged me during their life to learn until this high levels and also to my beloved family for their understanding the time I to sacrifice in this semester and also previous semester for my research. Thank you for their priceless moral support that they have given to me. Last but not least, appreciation to Universiti Teknikal Malaysia Melaka for giving the chance to me to complete my undergraduate program and gaining tonnes of priceless knowledge.

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LIST OF ABBEREVATIONS

RPM	Rotation Per Minute
MFS	Machinery Fault Simulator
VSD	Variacle Speed Driver
CLF	Cyclic Load Factor
DFT	Discrete Fourier Transform
HZ	Hertz

LIST OF SYMBOL

T= belt tesion

p = average normal pressure between belt and pulley

p_2= average normal pressure at slack side end of belt

R= radius of pulley

θ= angle of contact

α= active angle

τ= average shear stress

A_a= real contact area

A= apparent contact area

μ= coefficient of friction

c, a, B= constant

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

Unbalance in rotating machinering has been found to be one of the most common causes of machinery vibration. In any industry, rotating machinery is a basic part. Unbalance may develop in the system due to the operating conditions such as manufacturing, assembly, installed machines and other causes. Fabrication problems for example distorted castings, offbeat machining and poor gathering can also create unbalance. As example distortion problems is rotational stresses, aerodynamic and temperature change. Many of these occur during manufacture and others during operational existence of machine.

In addition, improper assembly is the reason why unbalance occurs when a rotor is being fabricated. As a theory, when a unbalanced shaft and unbalanced rotor united, a radial diplacement occurs from the necessary assembly which will produce an unbalance condition. These such as include consumption of the machine, wear and less right connection. The large unbalances will cause require large weight corrections and this can have negative impact on the integrity of the rotor.

The driven pulley and belt also must be studied. Forces is produced from normal and tangential and friction characteristics between an abrasive belt and pulley. Then, suitable size and material that can be used to withstand until 3000 RPM speed. As a theory, classical Euler equation state that for a flat belt power transmission assumes a constant coefficient of friction between pulley and belt. It also state that the coefficient of friction between rubber structure and hard surface depends to normal pressure, material consistent and shear stress. (Kim,H., 1987)

There are many equipment that are used to verify the imbalance system.The equipment will be used based on to check either the system is unbalance or not. One of the equipment is using Machinery Fault Simulator (MFS) and Vabbit pro vibration machinery diagnostic system by analyzing generated frequency at spectrum. The equipment can detect faults in gears, bearings and other mechanical components. A parameter-free method to analyze sensor signals that incorporates two or more frequency demodulation, phase demodulation of the raw signal data and amplitude demodulation. Any of equipment will use a sensors to detect the unbalance condition.

One example of sensors is an accelerometer. This device can measures a proper acceleration and have multiple applications in science or industry. It can detect and monitor vibration in rotating machinery and has a single or multi-axis to detect direction and magnitude of proper acceleration. Accelerometer works in many ways, two from it are capacitance sensor and piezoelectric effect. The capacitance accelerometer senses changes in between microstructures located next to the device and if a force moves, the capacitance will change to voltage for interpretation. For the piezoelectric effect, it is the most common form accelerometer and uses microscopic crystal structures that become accelerative forces. (Natalia, 2013)

1.2 PROBLEM STATEMENT

As known, the pulley is one of the important component in many machinery and industry users. However, the manufacturing is not perfect and there will be some defects occurred. Unbalance pulley will result in vibration that can affect bearing and many others component. In addition, the vibration sensor might not detect at very low rpm, but if the speed increase to 3000 RPM, the unbalance might become significant. This project would fabricate and test unbalance pulley with the speed of 3000 RPM.

1.3 OBJECTIVE

The objectives of this project are as follows :

1. To investigate unbalance pulley at various speed.
2. To do balancing on pulley using vector method.

1.4 SCOPE OF PROJECT

The scopes of this project are:

1. The cheapest way to design pulley balancing equipment with 3000 RPM speed of rotation motor.
2. Result of balancing testing to it stand until 3000 RPM speed of motor.

CHAPTER 2

LITERATURE REVIEW

2.1 THEORY OF BALANCING

Unbalance in the general definition is the combination between the "dynamic" unbalance and "static" unbalance (Krysinski, T., & Malburet, F, 2007). It also one of the conventional vibratory sources in rotating systems. The mass that circulate in rotating parts around the axis of rotation may generate inertial effects in specific cases. These will create vibrations in cyclic loads in the links and bearer structure involved. These loads generally noticeable and it is important for the structure to rotate at high speeds or when the structure is rotate there has a large mass with an inappropriate mass contribution around the axis of rotation and circulation. In mechanical systems rotating machinery is commonly used, including industrial turbomachinery, machining tools and etc. Vibration caused by mass unbalance is a normal problem in rotating machinery. Unbalance occurs if the geometric axis is not coincident with the principal axis of inertia of the rotor.

The center of inertia is on the axis of rotation and the axis of rotation is a principal axis of inertia must be to sure so that the equilibrium is obtain and to avoid unbalance. The systems are in equilibrium when the masses of rotating elements are distributed equitably around the axis of rotation and the resultant inertial effects are zero. The vibration and noise will not happened when a machine is in equilibrium. For a proper dynamic operation of machines these two parameters are important to be considered. A slight of asymmetry in rotating parts is enough to create an unbalance that causes dynamic responses at the bearings for high rotation speeds. Higher rotation speeds also can cause much greater centrifugal unbalance forces and current pattern of rotating equipment toward higher operational speeds to higher power density openly leads.

2.2 TYPES OF UNBALANCE

There are three types of unbalance :

i. Static unbalance
ii. Couple unbalance
iii. Dynamic unbalance

2.2.1 Static unbalance

Defined as the eccentricity of the center of gravity (MacCamhaoil, 1989) of a rotor that caused when the center of rotation has point mass at a certain radius. It also defines when the rotor will roll and it stops when its heavy spot is at the lowest position and has *in-phase* motion between both end of the rotor. It required to restore from the center of gravity to the center of rotation when an equal mass is placed at angle of 180° to the unbalanced mass and at the same radius. The static balancing including settling primary forces into one plane and adding a correction mass into that plane only. Many rotating parts that have most their concentrated mass in or very near one plane, such as car wheels, flywheels or etc can be determined as static balancing problems.

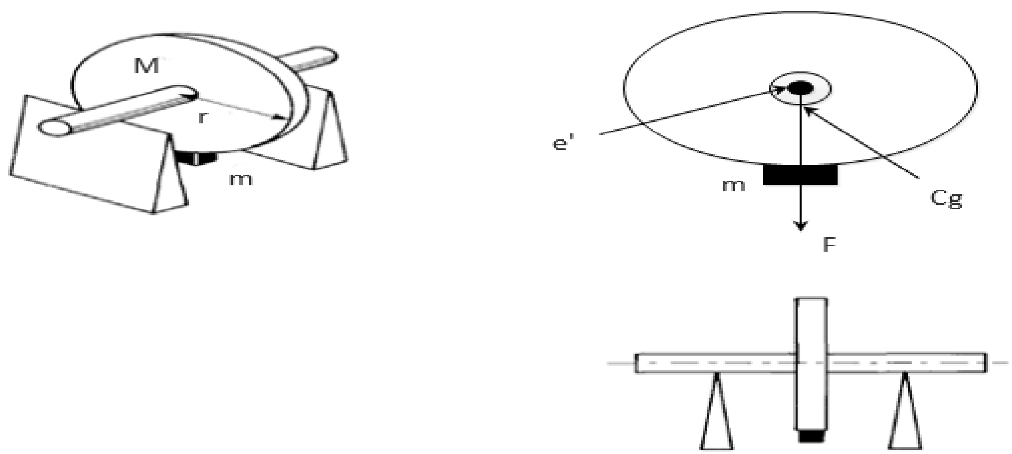


Figure 2.1: Static unbalance (MacCamhaoil, 1989)

Centrifugal force

$F = \overline{m\vec{r}}\omega^2$ [N]

Unbalance

$\vec{U} = \overline{m\vec{r}}$ [g mm]

Specific unbalance

$\vec{e} = \frac{m\vec{r}}{M}$ [gmm/kg]=[μm]

Equation [1]

2.2.2 Couple unbalance

Condition where are two same unbalance masses on the inverse end of the rotor, but 180° opposite each other. Between both ends of the rotor there are *out-of-phase* movement. It may happen when the diameter of the rotor is less 7 to 10 times its width. But it is imposibble to have an equal masses in case of cylinder that placed symmetrically about the center of gravity and positioned 180° to each other. As shown in figure 2.2, there is no eccentrity to the center of gravity but when the rotor turns, it will cause two masses shift in inertia axis and it no longer aligned with rotation axis that caused bearing to vibrate violently. The unbalance can only redressed by taking vibration measurements by adding the correction masses and rotor turning in two planes.

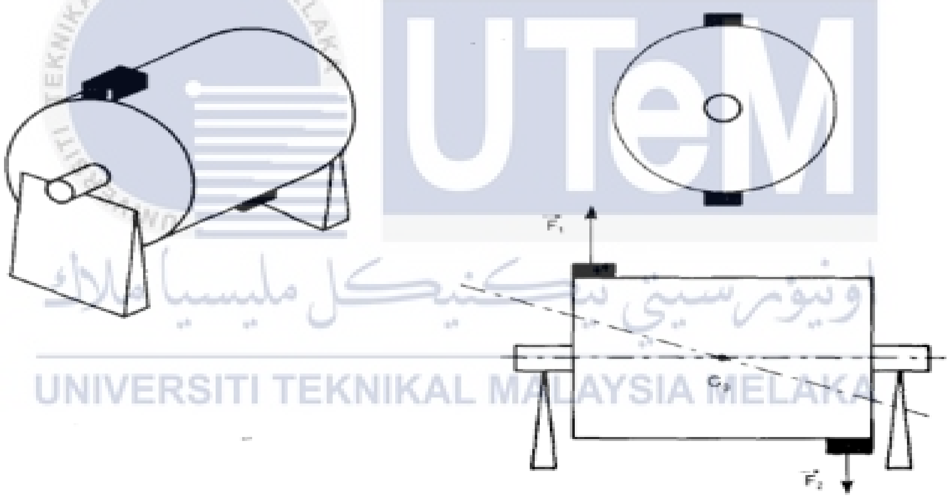


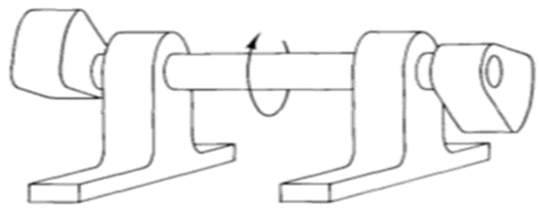
Figure 2.2: Couple unbalance (MacCamhaoil, 1989)

$|F_1| = |F_2|$

$\angle F_1 + 180^{\theta} = \angle F_2$

Equation [2]

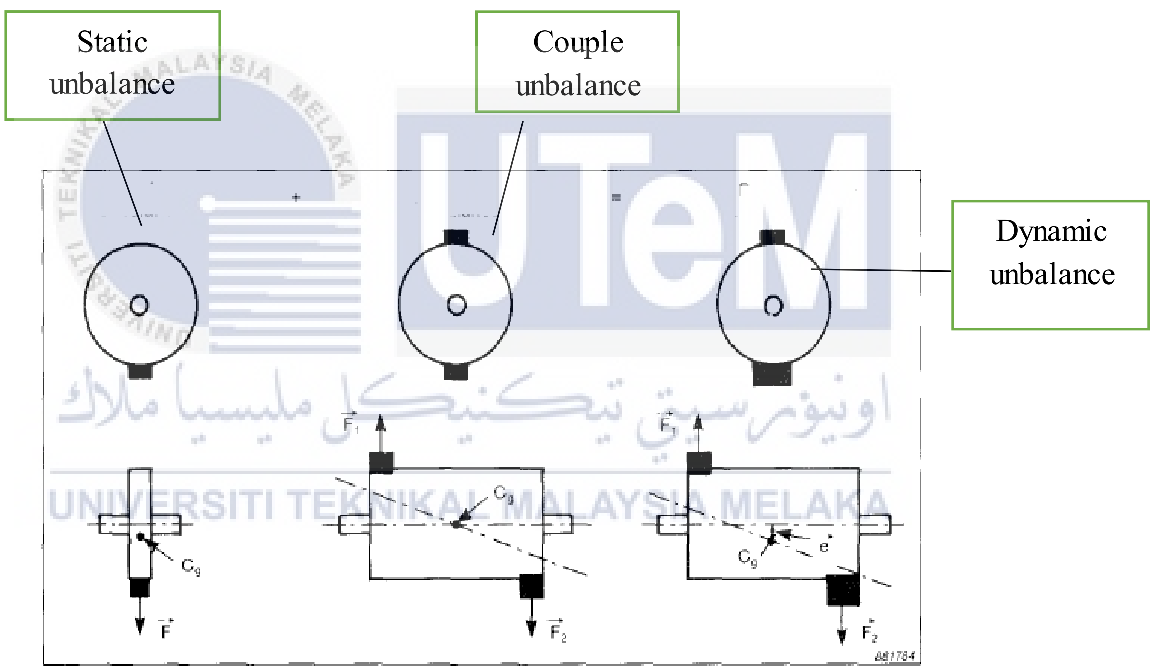
It can be seen that the rotor is stationary when the end masses is balanced each other but however a strong unbalanced occur when it starts to rotates and the difference between static balance and couple balance is indicated in figure 2.3.



Figue 2.3 : Static unbalance and couple unbalance (MacCamhaoil, 1989)

2.2.3 Dynamic unbalance

Eventhough static unbalance and couple unbalance happen, in practice, condition of these two are rarely found. Usually combination of these motion which gives dynamic unbalance where the most common type of unbalance usually found in rotors. The axis of the rotor will rotate as well as moves in radial direction. It is important to measure the vibration while the machine is running and add balancing masses in two planes to fixed the dynamic unbalance.



Figue 2.4: Dynamic unbalance (MacCamhaoil, 1989)

2.3 FORMATION OF EQUATIONS AND ANALYSIS

The unbalance occurs in the rotating machine when the mass centerline unbalance and the geometric center don't concur on each other. It will generates vibration and can make the component damages. To stand untill to 3000 RPM, vibration due to unbalance must be decreased to some acceptable and limit level. These levels or limits must be defined.

Unbalance amount $U,$ is expressed as (Algule & Hujare, 2015):

$$U = m * r \ldots \quad \text{Equation [3]}$$

Where, m= unbalance mass (kg)

r= distance from unbalance mass to shaft or rotor center line (in m)

The unbalance force generating the vibration is expressed as:

$$F = mr\omega^2 \ldots \quad \text{Equation [4]}$$

Where, F= force (N)

m= mass (kg) r= radius (m)

ω= speed (radian/sec)

Unbalance vibration = Unbalance force / Dynamic stiffness (Algule & Hujare, 2015)

Set of rotating elements is related with a non-bending solid marked S. Rotation is defined by the angle $\alpha(t)$ with the end that $\alpha(t) = \Omega t.$ Ω represents the rotation speed of the solid and considered fixed.

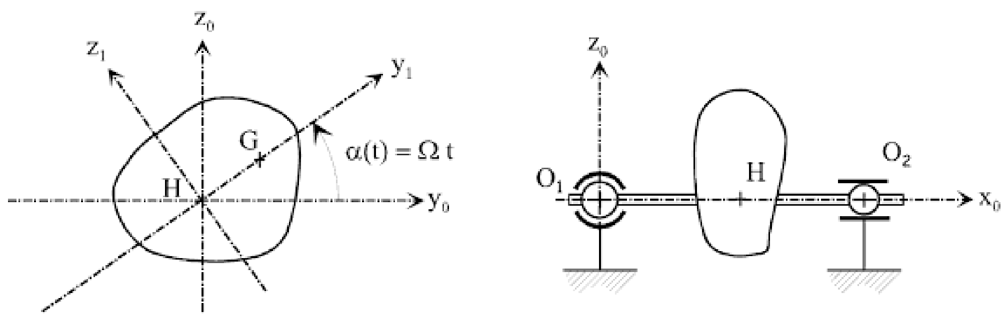


Figure 2.5: Parametric transformation associated with the solid in rotation (Krysinski, T., & Malburet, F, 2007).

O_1 and O_2 is the link centers of the two bearings. While G is the center of inertia of rotating system. The geometric dimensions are characterised by (Krysinski, T., & Malburet, F, 2007):

$$\overrightarrow{O_1O_2} = L\overrightarrow{x_0} \quad \overrightarrow{O_1H} = 1\overrightarrow{x_0} \quad \overrightarrow{HG} = e\overrightarrow{y_1} \ldots \qquad \text{Equation [5]}$$

There are several hypotheses must follows such as the links are without friction, rotating solid has certain shape, all resistant mechanical actions and mechanical action from the engine are modeled by pure pairs represented by $\overrightarrow{x_0}$ and the geometric center of the link are modeled by a pure resultant from the mechanical actions transfer through links. The rotor inertial matrix is write as follows (Krysinski, T., & Malburet, F, 2007):

$$\overline{\overline{I_G}}(S) = \begin{bmatrix} I_{XX} & -I_{XY} & -I_{XZ} \\ -I_{XY} & I_{YY} & -I_{YZ} \\ -I_{XZ} & -I_{YZ} & I_{ZZ} \end{bmatrix}_{(\vec{x}_1, \vec{y}_1, \vec{z}_1)} \ldots \qquad \text{Equation [6]}$$

Through the fundamental principle of dynamics that applied to the solid S, the loads in the links are obtained by (Krysinski, T., & Malburet, F, 2007):

$$m(S)\vec{A}_{G,S/R_g} = \vec{F}(bearing1 \ \rightarrow S) + \vec{F}(bearing2 \rightarrow S) + \ldots + \vec{F}(engine \ \rightarrow S)$$
$$+ \ \vec{F}(receiver \rightarrow S)$$

$$\vec{\delta}_1(S/R_G) = \vec{M}_{O_1}(bearing1 \ \rightarrow S) + \vec{M}_{O_1}(bearing2 \rightarrow S) + \ldots + \vec{M}_{O_1}(engine \ \rightarrow S) +$$
$$\vec{M}_{O_1}(receiver \rightarrow S) \ldots$$

Equation [7]

Only momentum equation in projection on $\vec{y}_0$ and $\vec{z}_0$ and resultant equation in projection on $\vec{y}_0$ and $\vec{z}_0$ provide necessary equation to study (Krysinski, T., & Malburet, F, 2007).

At the level of link centers result of load expressions is as follows:

$$\begin{cases} \left(F_{O_2}\right)_z = \frac{I_{XZ}}{L}\Omega^2\cos(\Omega t) - (\frac{I_{XZ}}{L} + \frac{mel}{L})\Omega^2\sin(\Omega t) \\ \left(F_{O_2}\right)_y = \frac{I_{XZ}}{L}\Omega^2\sin(\Omega t) + (\frac{I_{XZ}}{L} + \frac{mel}{L})\Omega^2\cos(\Omega t) \\ \left(F_{O_1}\right)_y = -\left(\frac{I_{XZ}}{L} + \frac{me(l-L)}{L}\right)\Omega^2\cos(\Omega t) - \left(\frac{I_{XZ}}{L}\right)\Omega^2\sin(\Omega t) \\ \left(F_{O_1}\right)_z = \left(\frac{I_{XY}}{L} + \frac{me(l-L)}{L}\right)\Omega^2\sin(\Omega t) - \left(\frac{I_{XZ}}{L}\right)\Omega^2\cos(\Omega t) \end{cases} \ldots$$

Equation [8]

From these equation several observations can be determined: There are harmonic function that angular frequency is correspond to the speed of rotation and the loads are proportional to Ω^2. The "$m\ e$" product is one of the important feature of unbalance. Effect that caused by this term is called “static unbalance” (Krysinski, T., & Malburet, F, 2007). This mentions the fact that focal point of inertia may not be on axis of rotation where ($e \neq 0$).

Other than that, products of inertia from the matrix of inertia I_{xy} and I_{xz} are the unbalance characteristics that different from other. Effect that come from these terms is called “dynamic unbalance” (Krysinski, T., & Malburet, F, 2007). This appropriate with the fact that suggest axis of rotation is not adjusted with principal axis of inertia.

To obtain balance in rotating system, it is solved by adding or removing mass in oder to adjust the mass distribution in space. Most preferred the smaller volume of added or removed mass than the volume of the rotor. The added or removed masses are considered dependable. Two masses m_1 and m_2 are added to individual points M_1 and M_2. Position of each mass m_i is expressed by (Krysinski, T., & Malburet, F, 2007):

$$\overrightarrow{O_1M_1} = a_i\vec{x}_0 + b_i + c_i\vec{z}_1 \quad \text{Equation [9]}$$

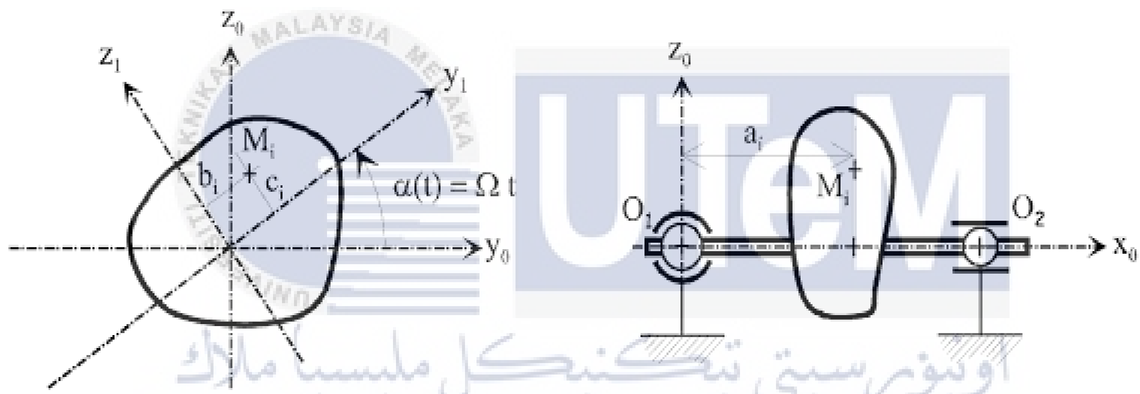
Balance can be obtained if the masses m_1 and m_2 are canceled with the inertial effects involved by the rotor or if following equations are confirmed (Krysinski, T., & Malburet, F, 2007):

$$\begin{cases} m(S)\vec{A}_{G,\frac{S}{Rg}} + m_1\vec{A}_{\frac{M_1}{Rg}} + m_2\vec{A}_{\frac{M_2}{Rg}} = \vec{0} \\ \vec{\delta}_1\left(\frac{S}{R_G}\right) + O_1M_1 \wedge \left(m_1\vec{A}_{\frac{M_1}{Rg}}\right) + O_2M_2 \wedge \left(m_2\vec{A}_{\frac{M_2}{Rg}}\right) = \vec{0} \end{cases} \quad \text{Equation [10]}$$

After projection, the equations are obtained (Krysinski, T., & Malburet, F, 2007):

$$\begin{cases} me + m_1b_1 + m_2b_2 = 0 \\ m_1c_1 + m_2c_2 = 0 \\ I_{XZ} + mel + m_1a_1b_1 + m_2a_2b_2 = 0 \\ I_{XZ} + m_1a_1c_1 + m_2a_2c_2 = 0 \end{cases} \quad \text{Equation [11]}$$

In order to pick the set of values, other criteria can be introduced (mass, inertia, etc.) when notice there are four equations to solve, with eight parameters $(m_1, m_2, a_1, a_2, b_1, b_2, c_1, c_2)$. It is impossible to ensure balance with $a_1 = a_2$ if the term I_{XZ} is non-zero. Therefore, to acquire both "static and dynamic" balance, it is important to act to axis of rotation perpendicularly to two distinct planes. Other than that, it is essential to know the position of the mass dispersion and the center of inertia on the system. Value of parameters I_{xy}, I_{xz}, e, m must be observe. One of the methods is to measure their effects, the loads in the



bearings in order to observe the characteristics of the existing system.

Figure 2.6: Geometric definition of the position of balancing masses (Krysinski, T., & Malburet, F, 2007).

2.4 PULLEY AND BELT SYSTEM

Pulley system are used when there is a need to transfer a motion into rotary system. When the motor is turned on it will rotates the driver pulley wheel and the belt will make the driven pulley wheel rotates as well. Pulley that smaller in size are called as driver pulley because it connected to the power that give all the power supply or drive to entire pulley system. While the larger pulley is driven round by the driver pulley wheel that rotates because the power from the driver pulley wheel that have provided from the motor operated.

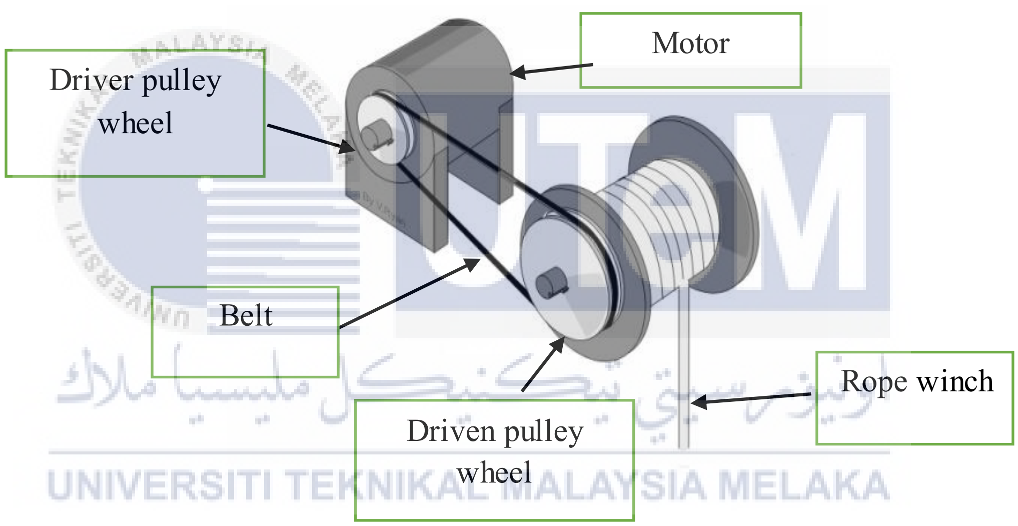


Figure 2.7: Simple pulley system

Pulley wheels are grooved to prevent the belt to slip off. Between the two pulley wheels there are tension that pulled tight to the belt that caused friction to make the driven follows the rotating driver. Without proper tension, the belt would slip and tension and friction is forced to it.

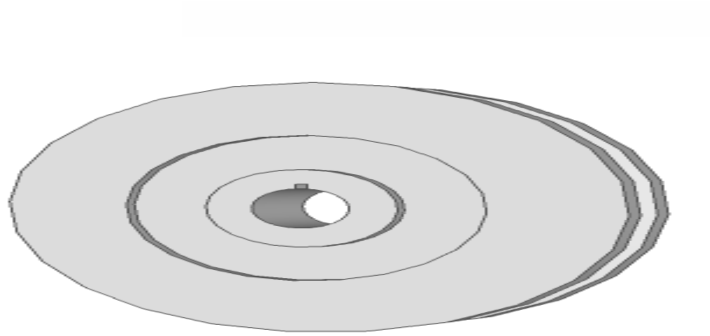


Figure 2.8: Pulley wheel

In general, the RPM(revolutions per minute) for the larger driven pulley will be less than smaller driver pulley wheel.

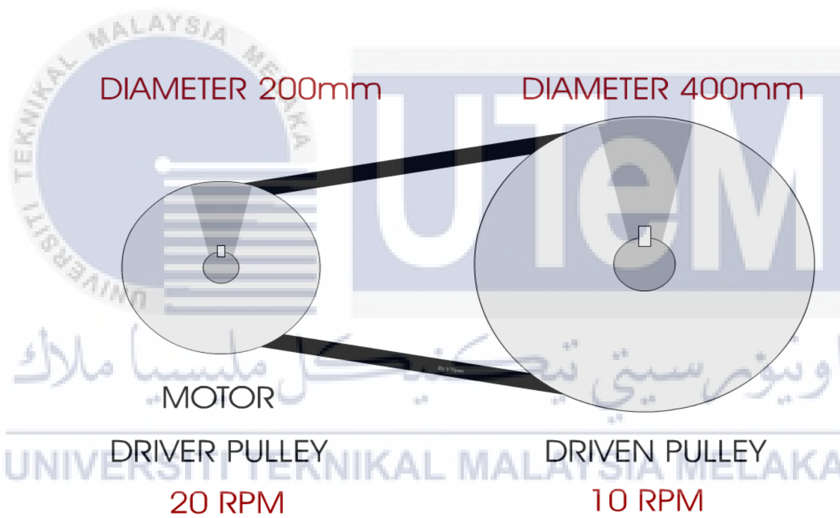


Figure 2.9: Velocity ratio (Kim, H., 1989)

As an example, the system shown above is when the driver pulley is attached to a motor and was switched on to revolves the driver pulley at 20 RPM. The diameter for driver pulley is 200mm and the diameter for driven pulley is bigger than driver pulley where the diameter is 400mm. This will make the smaller driver pulley wheel rotates twice as for one revolution of the larger driven pulley wheel. This happen from the velocity ratio that can be worked out in different ways mathematically.

Alternative method to calculate the distance moved is by:

$$\frac{\textit{Diameter of driven pulley}}{\textit{Diameter of driver pulley}} = \frac{N_{Dr}}{N_{Dn}} \quad \text{Equation [12]}$$

2.4.1 Forces between belt and pulley

From the classical Eular equation, it assumes a constant coefficient friction between belt and pulley is needed for a flat belt power transmission such as the coefficient of friction is independent of normal belt pressure (Kim & Marshek, 1987). Tangential and normal static equilibrium for a belt element with length $Rd\theta$ is called as active arc of contact as shown in figure 10.

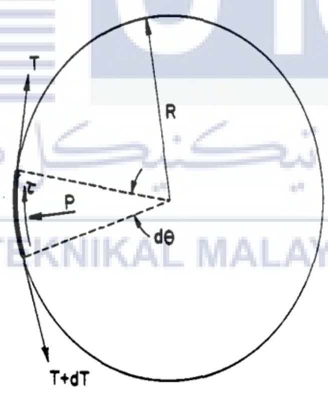


Figure 2.10: Diagram of abrasive belt on driven pulley (Kim & Marshek, 1987)

The elements requires that:

$$T - pR = 0 \qquad \text{Equation [13]}$$

$$dT - \tau R d\theta = 0 \quad \text{Equation [14]}$$

Since the friction per unit area τ is proportional to the actual contact area (Kim & Marshek, 1987).

$$\tau = c(\frac{A_a}{A}) \qquad \text{Equation [15]}$$

Where the c is a friction constant or shear stress limit and $(\frac{A_a}{A})$ is ratio of actual contact area to the clear contact area and the ratio alco can be write as where a and p are constant :

$$A_{A/}A = a \qquad \text{Equation [16]}$$

The substituting equations from (13), (15) and (16) into equation (14) gives :

$$dp - Bp^n d\theta = 0 \qquad \text{Equation [17]}$$

B is a constant $(B = ca)$ which depends on the belt pulley surface characteristic and belt material and the boundary condition is applied where $p = p_2$ at yields $\theta = 0$:

$$p = [p^{\frac{1}{2}-n} + (1-n)B\theta]^{\frac{1}{1}-n} \text{ for } n \neq 1 \quad \text{Equation [18]}$$

$$p = p_2 \exp(B\theta) \text{ for } n = 1 \qquad \text{Equation [19]}$$

A coefficient of friction μ and friction force τ can be found by combining the equation from [15] and equation [16] by using $\tau = \mu p$ as :

$$\tau = Bp^n \qquad \text{Equation [20]}$$

$$\mu = Bp^{n-1} \qquad \text{Equation [21]}$$

The constant B for three values n can be determined using equation [20].

Value for equation (Kim & Marshek, 1987):

T= *belt tesion at* $\theta, \theta = \alpha, \theta = 0$

p = *average normal pressure between belt and pulley*

p_2 = *average normal pressure at slack side end of belt*

R= *radius of pulley*

θ= *angle of contact*

α= *active angle*

τ= *average shear stress*

A_a= *real contact area*

A= apparent contact area

μ= *coefficient of friction*

c, a, B= *constants*

2.5 PREVIOUS STUDY

2.5.1 Balancing a system using variable speed drivers

In previous study, Oliviera L. Natalia and etc used two ways to handling the unbalance problems in mechanical systems that equipped with variable speed drivers. These two methods are capable to verify the unbalance of the system based on the cyclic load factor (CLF) and discrete fourier transform (DFT), torque signal that obtained from the drive is used by both methods. Real time balancing control can be achieved from the development of algorithms that can get necessary parameters from the drive that change into ladder programs able to be embedded to the drivers. Possibility to assist balancing in mechanical system based on created algorithms is the experimental result.

2.5.2 Load torque problem

Load torque or known as mechanical torque is a quantity related to the turning force exerted for purpose of turning or rotating objects which is also called the moment of force (Natalia, Cícero, & Herman, 2013). According to Halliday and Resnick, when the particle acts in relation to the origin, it is associated with distance and strength (1985). To determine percentage torque ouput the relationship between rated motor torque current used and torque current required by the motor is shown below :

$$\tau = \frac{T_M \times 100}{I_M} \times Y \qquad \text{Equation [22]}$$

Where the T_M is torque motor current (A), I_M is the rated motor current (A) and Y is determined by :

$$Y = 1 \text{ when } N \leq \frac{V_O \times N_{SINC}}{V_R} \text{ or} \qquad \text{Equation [23]}$$

$$Y = \frac{N_{SINC}}{N} \times \frac{V_O}{V_R} \text{ when } N > \frac{V_O \times N_{SINC}}{V_R} \qquad \text{Equation [24]}$$

V_O is the maximum output voltage (V), N_{SINC} is the motor synchronous speed (RPM), V_R is the instant output voltage of motor (V) and N is the speed of motor instant (RPM). Therefore the function of torque current of VSD, output voltage of VSD and speed of motor instant is defined as torque.

2.5.3 Cyclic Load Factor (CLF) approach

According to Takács (2003) CLF is the increased force required by motor also defined as required torque of the system. Equation [24] is used to decide the CLF :

$$CLF = \frac{\sqrt{\frac{\int_{\theta-0}^{2\pi}[T(\theta)]^2 d\theta}{2\pi}}}{\frac{\int_{\theta-0}^{2\pi}[T(\theta)] d\theta}{2\pi}} \qquad \text{Equation [25]}$$

Where T is mechanical torque and θ is angle of the crank as mechanical torque is proportional to the electrical torque motor. But due to limitations of the drive the calculations of CLF is simplify as shown :

$$CLF = \frac{\sqrt{\frac{\sum x^2}{n}}}{\frac{\sum x}{n}} \qquad \text{Equation [26]}$$

x is the value torque read by the drive and n is number of samples. One of the application CLF is to reduce power consumption of motor. According to Takács (2003) minimizing CLF will lower the force exerted by the motor and reduced the power which lower in by lower costs and reduce wear at equipment.

2.5.4 Discrete Fourier Transform (DFT) approach

Equation [26] is used to calculate the DFT which shown below :

$$x[m] = \sum_{n-0}^{N-1} x[n] \left[\cos\left(\frac{2\pi mn}{N}\right) - jsen\left(\frac{2\pi mn}{N}\right)\right] \qquad \text{Equation [27]}$$

The signal obtained is periodic and N is number of sample, m is index related to frequency and n index related to time while $x[n]$ is sample collected. To identify the period sample where the signal is periodic signal the calculations is given by :

$$F_s = \frac{1}{T} \qquad \text{Equation [28]}$$

F_s is the sampling frequency and T is sample period.

$$F_s = \frac{f_o \times N}{m} \quad \text{Equation [29]}$$

f_o is frequency option determined by :

$$f_o = \frac{CPM}{f} \quad \text{Equation [30]}$$

CPM is the cycles per minute of machine and f input frequency of VSD, a constant value and CPM can be fined by :

$$CPM = \frac{RPM}{f} \quad \text{Equation [31]}$$

Where RPM is unit speed of motor. The system signals contain information about functioning and it is possible to implement the DFT to analyze spectrum that can identify each frenquency and harmonic that cause defects in system.

2.5.5 Result using CLF method

Expected to find CLF = 1 that perfectly balanced and greater means that the machine is less balanced. Three different test were conduct. The first one are without any weights and the second and the third with 8kg and 16kg attached 72cm from rotation center.

Table 2.1 : Values of CLF calculated unit with balanced load and unbalanced load with 8kg and 16kg (Natalia et al., 2013)

CLF for unit with Balance Load	CLF for Unit with 8 kg at 72 cm from rotation axis	CLF for Unit with 16 kg at 72 cm from rotation axis
0.9999971	1.0040721	1.0097249
0.9999865	1.0040888	1.0099884

As expected, value CLF for unit with balanced load is close to one and for unbalance unit with 8kg and 16kg the algorithm calculated values is greater than 1. (Natalia et al., 2013)

2.5.4 Result using DFT method

The purpose is to analyze frequency range of electrical torque where particularly the amplitude of first harmonic. By analyzing amplitude of first harmonic, the unbalance load can be identified. This is because unbalance load has sinusoidal torque with mechanical frequency that make the first harmonic increases the unbalance. Similar to the three test before, firstly without any weights and next is 8kg and 16kg attached at 72cm from axis of rotation.

Table 2.2 : Values of the first harmonic with DFT calculated for the unit with balanced load

First Harmonic for Unit with Balance Load	First Harmonic for Unit with 8 kg at 72 cm from rotation axis	First Harmonic for Unit with 16 kg at 72 cm from rotation axis
252.71	429.25	825.80
252.79	429.36	825.71

and imbalancedload using 8kg and 16kg (Natalia et al., 2013)

Figure 2.11: Variable speed driver

As conclusion the result from both methods is satisfied and the embeded algorithms is able to identify whether the load of machine driven by variable speed driver is unbalance or in balanced system as calculate level of unbalance to dynamically control direct from VSD. This two method are the best way to analyze balancing in the system which can be reffered to making another balancing equipment using another mechanical system.

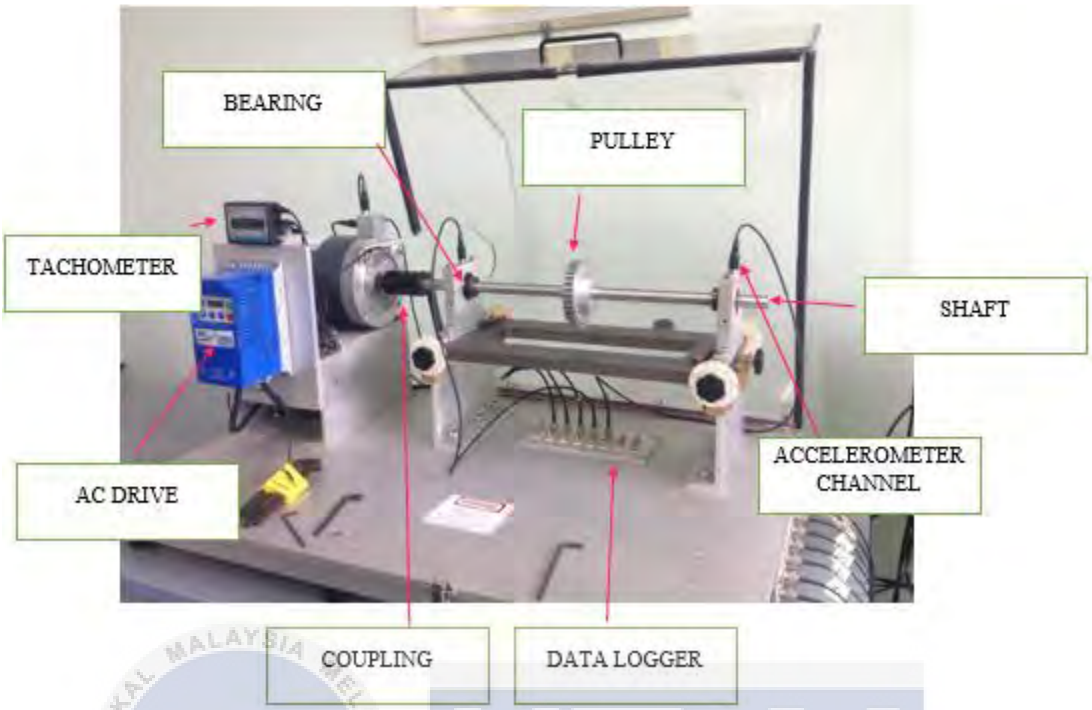
CHAPTER 3

METHODOLOGY

2.1 INTRODUCTION

This chapter would describe the methodology that is used in this project to balance pulley 3000 RPM without vibrate and to make sure the system is in balance and find the way to solve the unbalance that happen in the pulley system. There are two method that can be used for balancing where as the method is vector method and four-run method but only one methood that are used in this project. The sketching drawing of pulley are done to make the suitable pulley that can stand to 3000 RPM speed of motor where the pulley are attached with the weight that are calculated from the vector method. This project starts from the flow chart sketching so that the overall of project flow can be analyzed and visualized first so that project progress can be done step by step.

3.2.EXPERIMENTAL SETUP



Figu

re 3.1: Machinery Fault Simulator

3.2.1 Schematic diagram

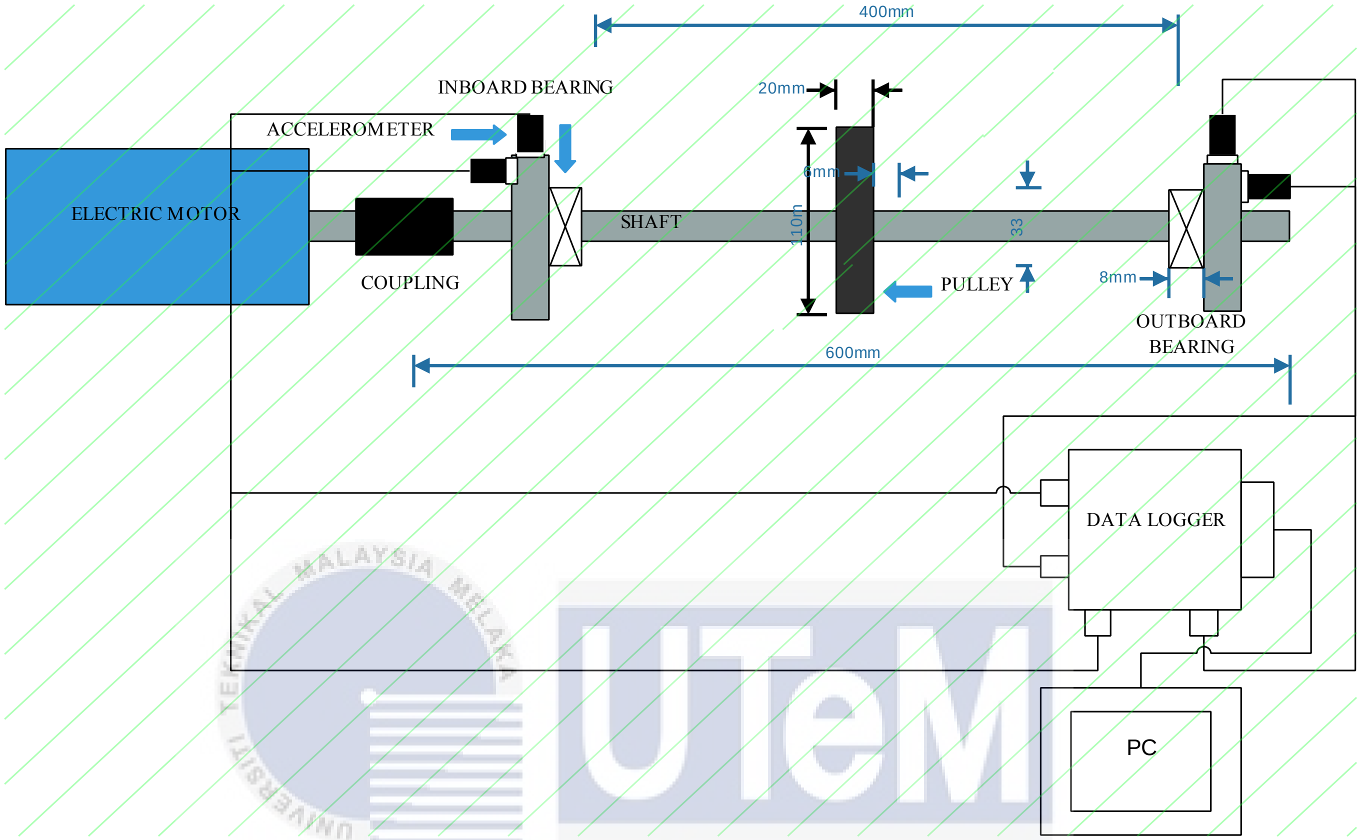


Figure 3.2: Machine Fault Simulator operation (schematic diagram)

Machine Fault Simulator (MFS) is used to run this experiment. This machine is basic simulator which is used to study the vibration spectra of common faults and learn fault signatures such as crack shafts, fan and mechanical rub. To study about the balancing, this simulation can shows the result which it can detect fault of unbalanced when the machine is running. By using this machine, it can be known whether the pulley is balanced or unbalanced condition. For this experiment, the pulley will be run untill to 50 Hz to detect whether it can stand up to 50 Hz as equal to 3000 RPM speed of motor. From this machine, there are 8 channels which connected to data logger which will be exposed to the PC. The first channel was from tachometer. The second, third, and fourth channel are from motor in horizontal, vertical and axial axis accordingly. The 5th and 6th channel are from inboard bearing in vertical and horizontal axis the 7th and 8th channel are from outboard bearing in vertical and horizontal axis. These channels are connected by accelerometer channel.

Table 3.1: Data Measurement Setup

No of channel	Couple	Name of channel	Volts/unit	Units	ICP	Sensor Type
1	AC	Tachometer	1.0000	v	OFF	Tachometer
2	DC	Motor-horizontal	0.1000	g	ON	Accelerometer
3	DC	Motor-vertical	0.1000	g	ON	Accelerometer
4	DC	Motor-axial	0.1000	g	OFF	Accelerometer
5	DC	Inboard bearing - vertical	0.1000	g	ON	Accelerometer
6	DC	Inboard bearing - horizontal	0.1000	g	ON	Accelerometer
7	DC	Outboard bearing - vertical	0.1000	g	ON	Accelerometer
8	DC	Outboard bearing - horizontal	0.1000	g	ON	Accelerometer

The table above shows the data measurement that needs to do the setup. This is important because it can give the data of vibration analysis based on the table given. Since the sensitivity accelerometer is 100 mv/g so the reading of volts is 0.1 v.

The data acquisition is setup as follow:

1. Frequency limit = 1000 Hz
2. No of spectra lines = 400 lines
3. Sampling rate = 100 × 2.56 = 2560 Hz
4. No of samples = 400 × 2.56 = 1024

3.2.2 Function of part

Here a few function of main part on this equipment.

1. Ball bearing

Support axial and radial loads also reduce the rotational friction.

2. Shaft

Transmit torque and rotation and usually used to connect each other component of a drive train because of distance and allow movement between the components.

3. Motor

Electrical machine that converts electrical energy into mechanical energy supply power into the equipment.

4. Pulley

Wheel on shaft that to support movement and change of direction of belt along its circumference.

3.3 FABRICATION OF PULLEY

To achieve the objective of this project some actions are carried out where it start with some literature review from the journal, internet, book, article or any material that related with this project title to design pulley balancing equipment for high RPM. The design of pulley were delivered from the literature review and previous study

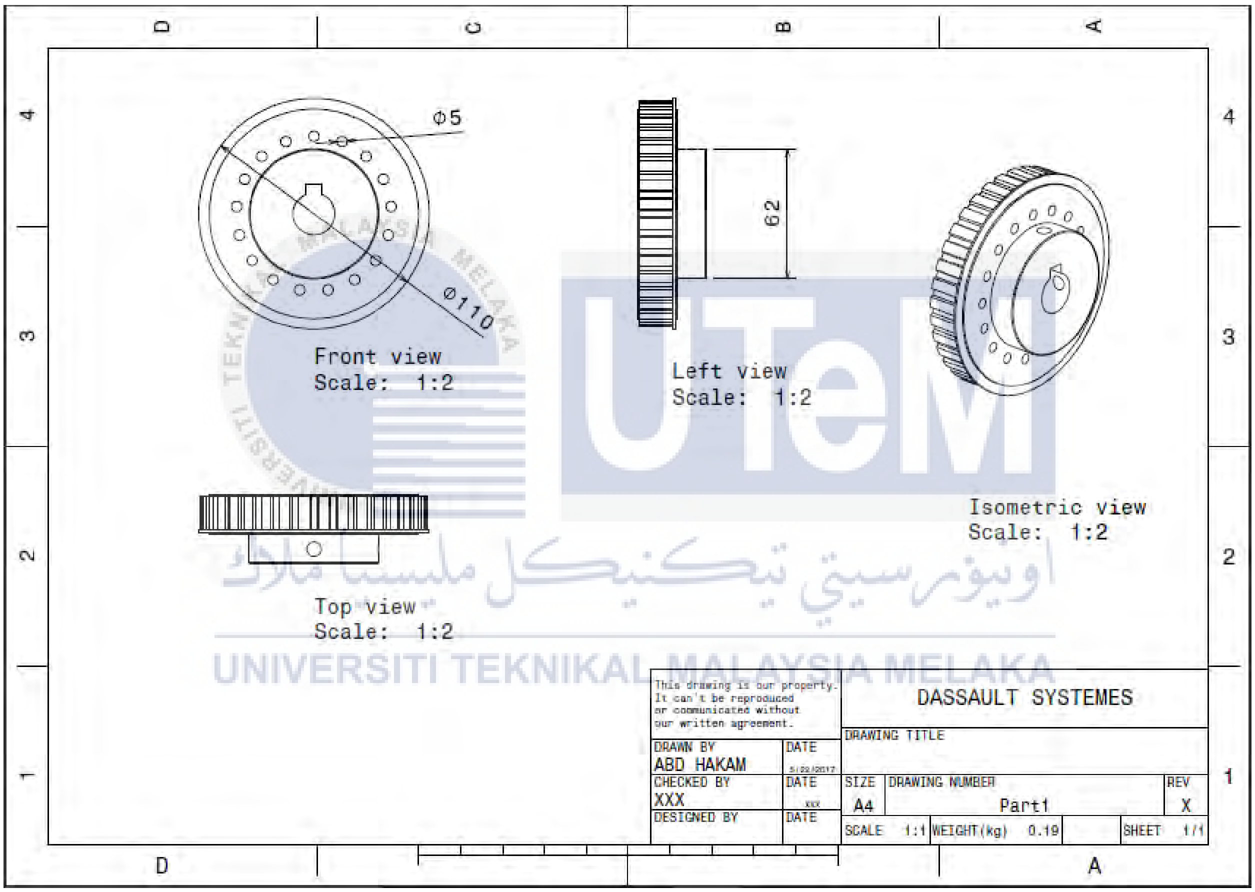


Figure 3.3: Drawing of pulley to use at MFS machine

This pulley is designed and fabricated to be used at MFS machine to test its unbalance A balancing weight is to used to reduce the amplitude vibrations. There is a hole on the pulley that allows weight to attach to it. This that can enhance the stability of the pulley in positioning using the vector method

3.3.1 Spectrum frequency analysis

A few example of unbalance spectrum frequenc diagram analyze.

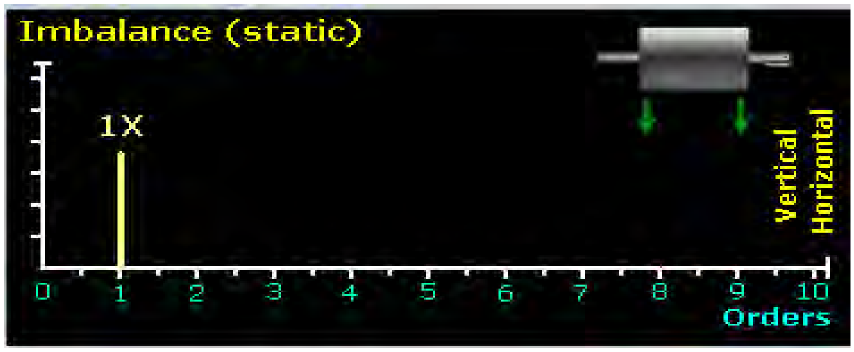


Figure 3.4: Unbalance (static) spectrum frequency

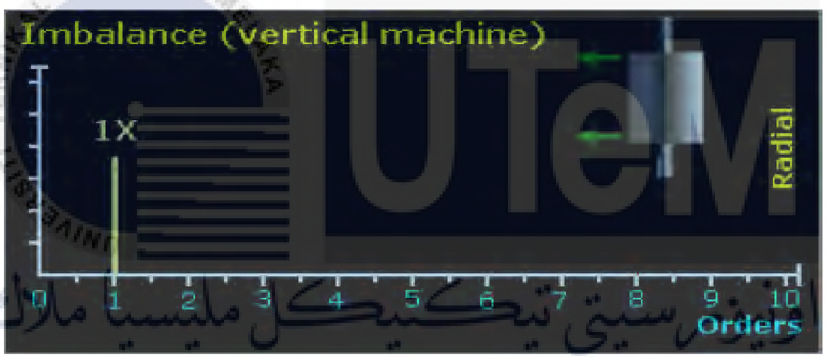


Figure 3.5: Unbalance (vertical) spectrum frequency

Figure 3.6: Unbalance (overhug) spectrum frequency

High 1x vibration level will shown in diagram to analyze there are unbalance problem in the system.

3.4 BALANCING METHOD

The type of balancing method which will be conducted in balancing experiment are vector method.

3.4.1 Vector Method

In this method, a balance will be done by making two runs. The first run is the measure of the original amount of vibration and the reading of the phase angle. The second run occur after the trial weight is added and read the vibration and phase angle again. From this information, a plot is made to determine the amount and location of balancing weight. This will occur when the trial weight is removed and the balance weight is installed. There is some running to check the balance to know whether the value is higly decreased.

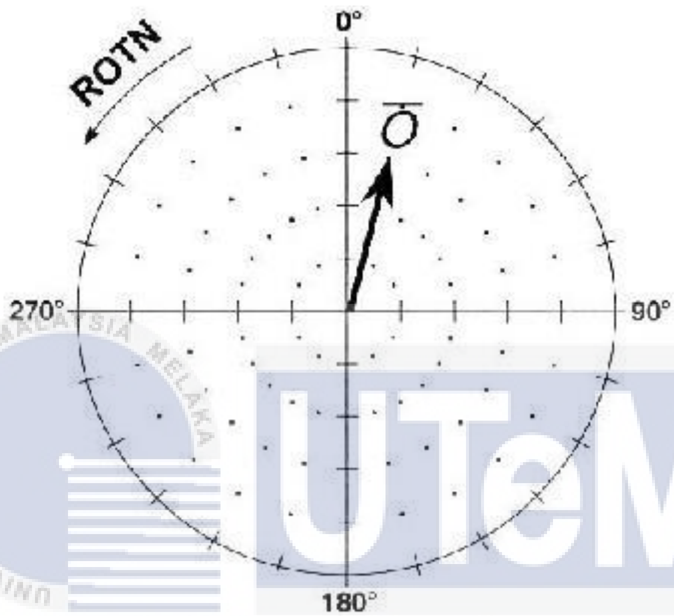
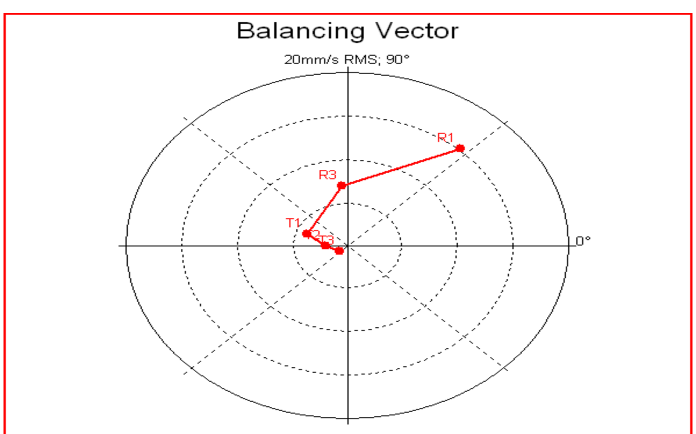


Figure 3.7 : Vector Plot Diagram

Figure above shows the vector plot diagram. There are many procedures in reaching the results. The machine will be run to record the amount of unbalanced which is referred to as "O". Next, some trial weight is attached to a relevant and suitable positions. The mass of trial weight should be large and heavy that could impose effects on the dynamics of the rotating system without wrecking



the machine. Subsequently, the frequencies of the vibration is measured as well as the phase angle which may be referred to as “O+T”. Then, a vector is plotted to represent the original “O” on the graph. Then, another vector is plotted to represent the “O+T” on the same graph. Based on vector calculation, the trial vector “T” can be obtained. The length if “T” and the angle between “T” and “O” is measured. This is for the phase shift angle. The direction of the phase shift from “O” to “O+T” is also checked. Further, some calculation which require the balance weight by multiply the trial weight with “O”/”T” is calculated. Finally, the trial weight is removed and is placed with the balance weight from the trial weight. The balance weight must be placed at the same radius as the trial weight. The condition the rotor should also be balanced. Run again the rotor if the balancing was successful.

Figure 3.8: Balancing vector plot for determination balanced mass and position

Figure above shows the determination of “O” and “O+T” in vector plot. From this information, the value of “T” will be noticed.

3.4.2 Flow chart

This method of this project is summarized in the flow chart as shown in figure below:

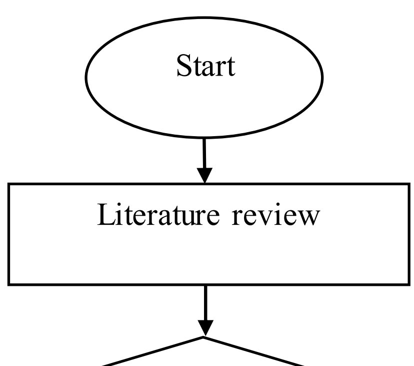


Figure 3.9: Flowchart of Methodology

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

This chapter will discuss on the results of the experiment whereby three different types of parameters are conducted. Initially, the pulley are run up to 50 Hz or as equal as 3000 RPM to find where the occurence of the resonance happen. Generally, resonance happen when the object oscillation amplitude are increased as a result of another forced object vibrations matches with the object's natural frequency. In this situation, when the pulley are vibrated at the same natural frequency of a motor forces, it matches into vibrational motion that increases the amplitude. Analysing the result from the graph plotted, it is seeming that resonance occur at the frequency 40 Hz to 50 Hz. Subsequently, the graph is minimized from 40 Hz until 50 Hz to locate where the range of the resonance happen. After identifying the resonance range, this experiment are run with 30 Hz to avoid the resonance that produce at 40 Hz to 50 Hz from disrupting the amplitude results and with three different parameter using the pulley at center of shaft, pulley at inboard of bearing and pulley at outboard of bearing.

Figure 4.1: Front view of pulley that has been fabricated

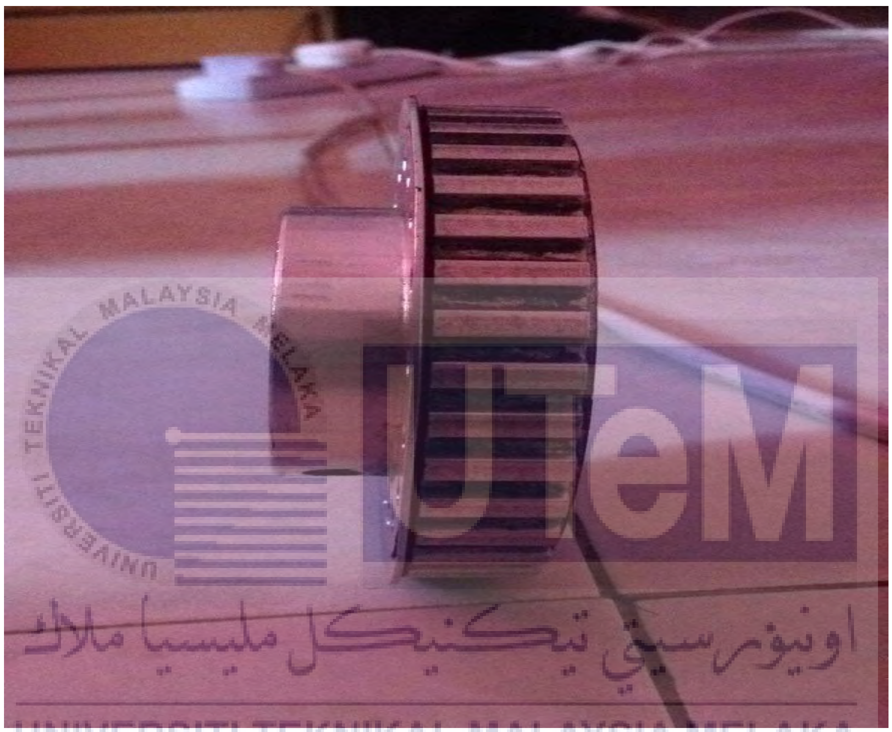


Figure 4.2: Side view of pulley that been fabricated

4.2.RESONANCE

4.2.1 10hz-50hz

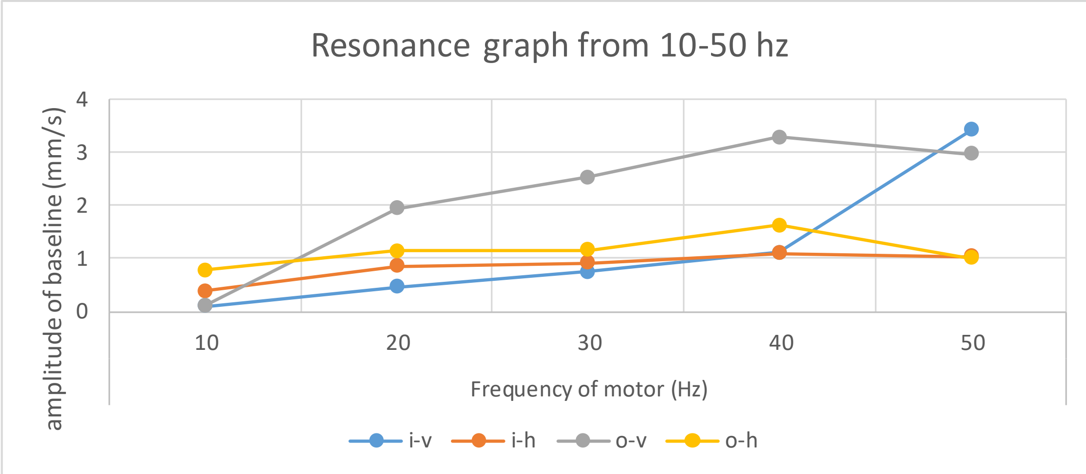


Figure 4.3: Graph of amplitude against frequency of motor from 10 Hz to 50 Hz

	Frequency of motor (Hz)				
Type of channel	**10**	**20**	**30**	**40**	**50**
Inboard vertical	0.106238	0.465455	0.755971	1.116592	3.421859
Inboard horizontal	0.384923	0.848607	0.910055	1.099188	1.032486
Outboard vertical	0.117404	1.946421	2.540119	3.292161	2.973633
Outboard horizontal	0.787703	1.147951	1.156355	1.623502	1.005531

Table 4.1: Results of amplitude of pulley from 10 Hz to 50 Hz

Based on the figures and tables above, the results show the increasing trend of the amplitude at 40 Hz and a slight decrease at 50 Hz except for the amplitude at the inboard vertical. This means that there is a resonance at frequency 40 Hz that affects the result of the amplitude for the pulley and further makes the object more vulnerable towards vibration. This could be due because the pulley are vibrated with the same natural frequency of motor force

4.2.2 40hz-50hz

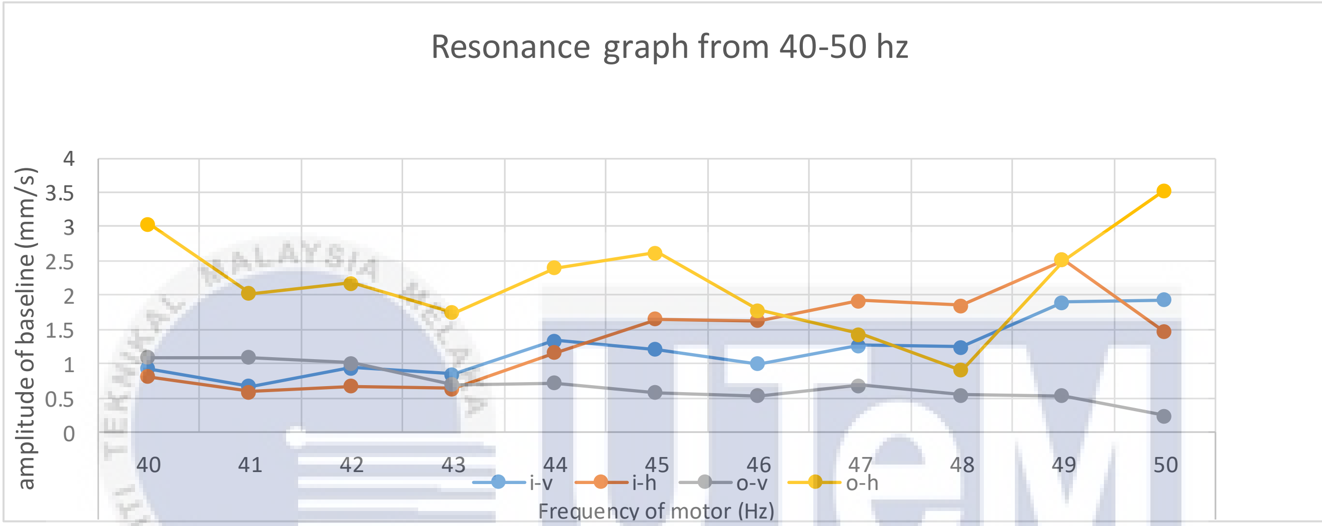


Figure 4.4: Graph of amplitude againts frequency of motor from 40 Hz to 50 Hz

	Frequency of motor (Hz)										
Type of channel	**40**	**41**	**42**	**43**	**44**	**45**	**46**	**47**	**48**	**49**	**50**
i-v	0.931977	0.674323	0.943729	0.850748	1.340430	1.209250	1.000079	1.275848	1.245920	1.900953	1.930243
i-h	0.817806	0.603642	0.684913	0.644487	1.171078	1.654357	1.638067	1.927096	1.856936	2.518739	1.474947
o-v	1.10156	1.094972	1.014573	0.704540	0.723530	0.584590	0.539307	0.691799	0.557023	0.542277	0.251292
o-h	3.037417	2.026483	2.174361	1.755057	2.405028	2.621319	1.782233	1.432765	0.906554	2.51486	3.528757

Table 4.2: Results of amplitude of pulley from 40 Hz to 50 Hz

Based on the figures and tables above, the results show the amplitude from 45 Hz to 48 Hz to be fluctuated and increased at 49 Hz. Further, it decreases at 50 Hz. The resonance that occur at 40 Hz to 50 Hz are minimized to get a more accurate value of frequency that could be disrupted by the resonance. Referring to this graph, an increase is observed at 49 Hz which indicates that resonance occurred at that value. Other than that, the amplitude reading from outboard bearing at horizontal axis are not significant maybe due to error happen at accelerometer sensor or the outboard bearing.

.4.3 BASELINE

4.2.1 MOTOR ONLY

This experiment setup is for baseline data which take data for motor only, the shaft without pulley, pulley at center of shaft, pulley at inboard bearing and pulley at outboard bearing. The data is taken in vertical axis and horizontal axis. For motor only, the data is taken from the amplitude when running the motor only

Figure 4.5: Figure of motor only

Table 4.3: Table of amplitude of vibration for motor only

Type of motor	Amplitude of vibration (mm/s)
Horizontal	0.1626
Vertical	0.182

Table above show about 0.1626 mm/s for horizontal and the vertical is 0.1820 mm/s. This is to make sure there is no fault on the motor and from the result it show less vibration because the motor is in balanced.

4.2.2 WITHOUT ANY PULLEY

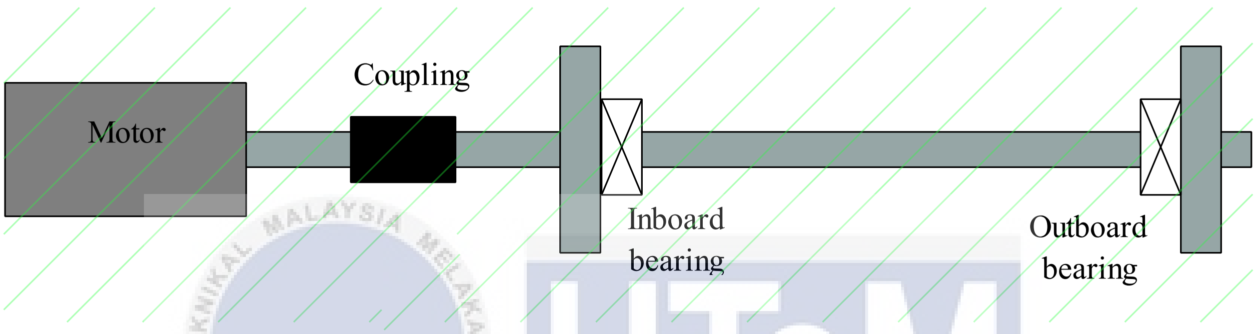


Figure 4.6: Schematic diagram of experiment runs without any pulley

Type of bearing	Amplitude Vibration at Inboard (mm/s)	Amplitude Vibration at Outboard (mm/s)
Horizontal	0.03149	0.01719
Vertical	0.01959	0.03295

Table 4.4 Results of baseline in inboard and outboard bearing

Based on the results of baseline for both bearing at the shaft without any pulley, the amplitude of vibration is still balanced. This is because there is no any puley or load that attached on the shaft that can make it vibrate.

4.2.3 SINGLE PULLEY (center shaft)

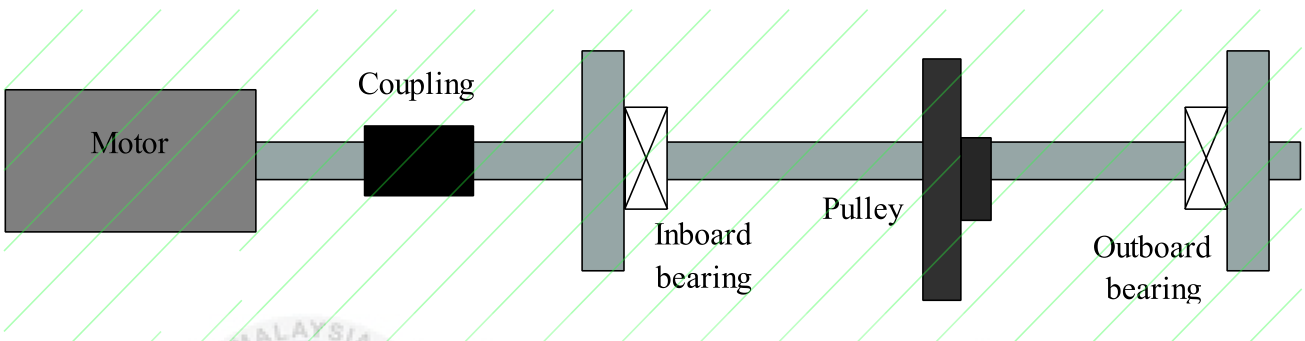


Figure 4.7: Schematic diagram of experiment runs with pulley at center of shaft

Type of bearing	Amplitude Vibration at Inboard (mm/s)	Amplitude Vibration at Outboard (mm/s)
Horizontal	0.786062	3.000385
Vertical	0.754262	0.564335

Table 4.5: Results of baseline both bearing with pulley at center of shaft

Based on the table 4.5 above, it shows the higher amplitude are produced compared to the baseline without any pulley that attached to the shaft. This is due to some load that may effect the result of amplitude to become higher. The amplitude at the horizontal outboard bearing shows the higher reading compared to the other because of fault that come from the bearing when running or the accelerometer itself.

4.2.4 SINGLE PULLEY (at inboard bearing)

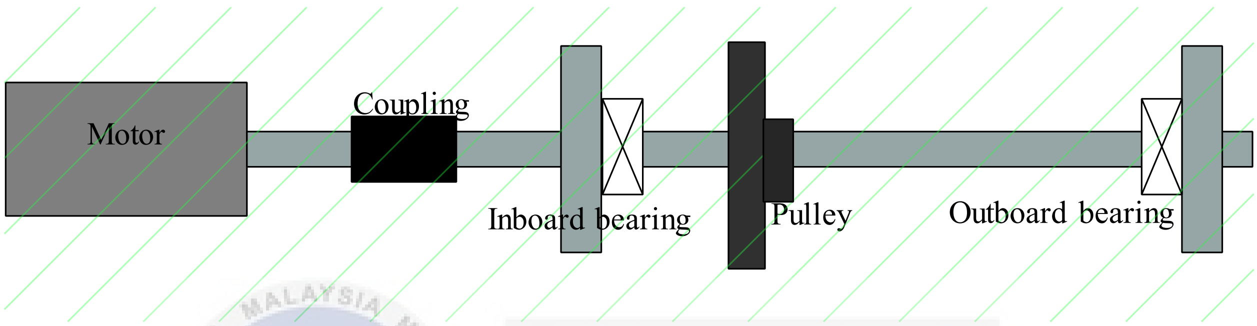


Figure 4.8: Schematic diagram of experiment runs with pulley at inboard bearing

Type of bearing	Amplitude Vibration at Inboard (mm/s)	Amplitude Vibration at Outboard (mm/s)
Horizontal	0.420314	2.613173
Vertical	0.749362	0.878196

Table 4.6: Results of baseline both bearing with pulley at inboard bearing

Based on the table 4.6 above, the average from the amplitude vibration pulley at inboard bearing are lower than amplitude vibration inboard bearing of pulley at center of shaft. This is because the inboard bearing is balanced with load from the pulley that attached to inboard bearing. The amplitude vibration of horizontal outboard bearing still high compared to the other maybe because of the fault of bearing.

4.2.5 SINGLE PULLEY (at outboard bearing)

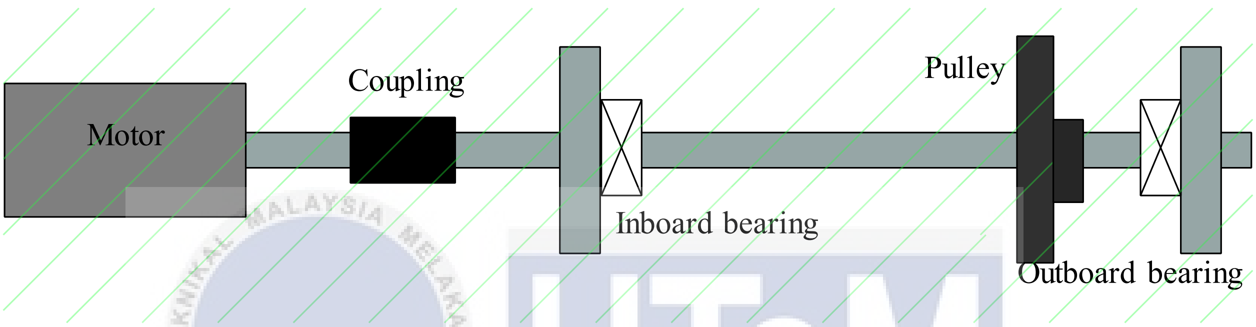


Figure 4.9: Schematic diagram of experiment runs with pulley at outboard bearing

Type of bearing	Amplitude Vibration at Inboard (mm/s)	Amplitude Vibration at Outboard (mm/s)
Horizontal	0.516199	2.897311
Vertical	0.751814	1.309939

Table 4.7: Results of baseline both bearing with pulley at outboard bearing

Based on the table 4.7 above, the amplitude vibration at outboard bearing at both of axis is higher than amplitude of vibration for both of axis at inboard bearing. This is because the outboard bearing especially at horizontal axis is affect with the fault or from the sensor.

4.4 UNBALANCED

4.4.1 SINGLE PULLEY (center shaft)

4.4.1.1 Effect of adding mass (7 g)

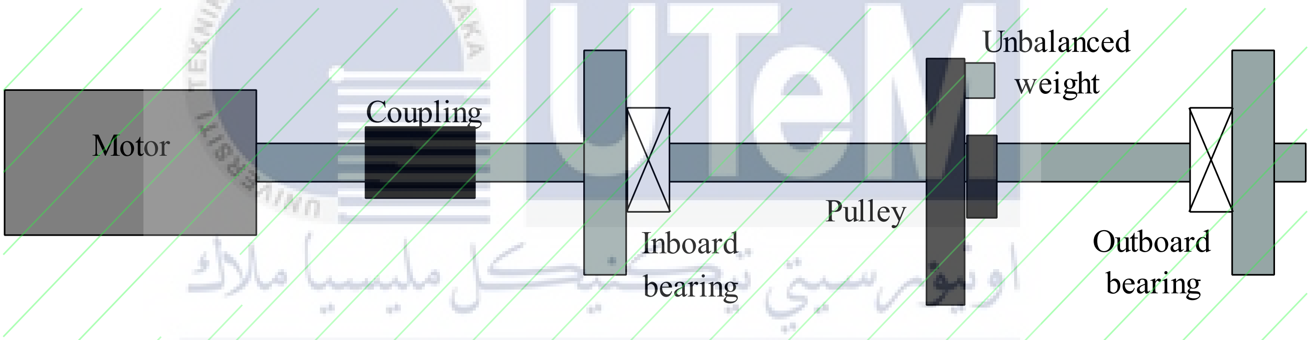


Figure 4.10: Schematic diagram of experiment runs with unbalance pulley at center shaft

Type of bearing	Amplitude Vibration at Inboard (mm/s)	Amplitude Vibration at Outboard (mm/s)
Horizontal	0.946097	3.451151
Vertical	0.697575	0.954208

Table 4.8: Results of amplitude both bearing with unbalance pulley at center shaft

Based on table 4.8 above, the range value amplitude vibration for inboard and outboard bearing are higher than the range value of amplitude vibration at inboard and outboard bearing before the unbalance weight is attached to the pulley at center shaft. This is actually the right value for the amplitude vibration because from the unbalance weight will increase the force that cause the increasing of amplitude vibration.

4.4.2 SINGLE PULLEY (at inboard bearing)

4.4.2.1 Effect of adding mass (7 g)

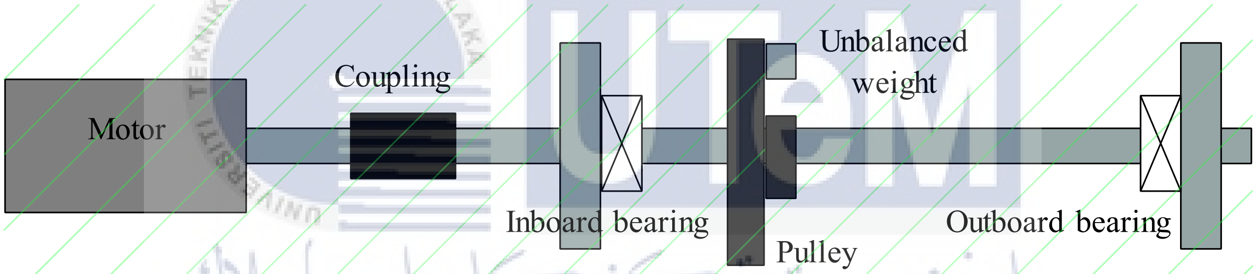


Figure 4.11: Schematic diagram of experiment runs with unbalance pulley at inboard bearing

Type of bearing	Amplitude Vibration at Inboard (mm/s)	Amplitude Vibration at Outboard (mm/s)
Horizontal	0.762285	3.551396
Vertical	0.531827	1.218200

Table 4.9: Results of amplitude both bearing with unbalance pulley at inboard bearing

Based on table 4.9 above, the range value amplitude vibration for inboard and outboard bearing are higher than the range value of amplitude vibration at inboard and outboard bearing before the unbalance weight is attached to the pulley at inboard bearing. This is actually the right value for the amplitude vibration because from the unbalance weight will increase the force that cause the increasing of amplitude vibration.

4.4.3 SINGLE PULLEY (at outboard bearing)

4.4.3.1 Effect of adding mass (7 g)

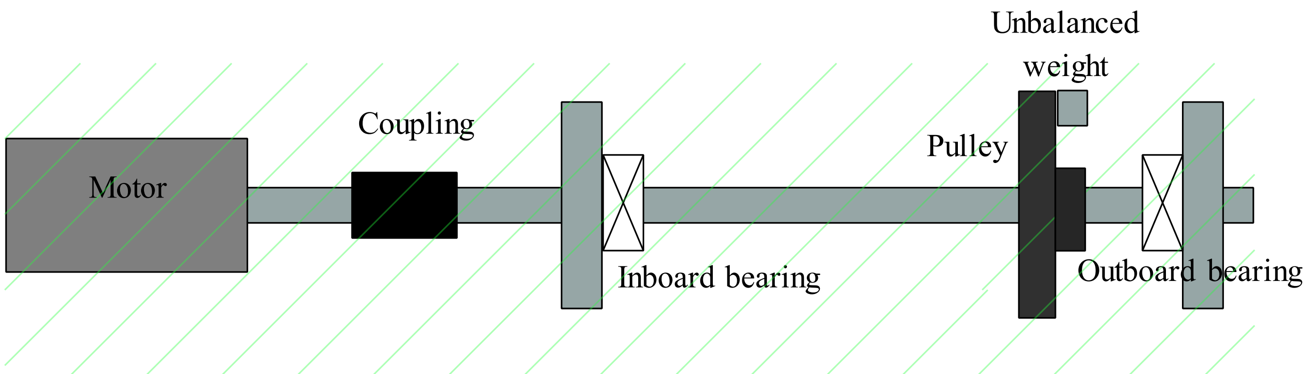


Figure 4.12: Schematic diagram of experiment runs with unbalance pulley at outboard bearing

Type of bearing	Amplitude Vibration at Inboard (mm/s)	Amplitude Vibration at Outboard (mm/s)
Horizontal	1.014347	3.949582
Vertical	0.527836	1.386937

Table 4.10: Results of amplitude both bearing with unbalance pulley at outboard bearing

Based on table 4.10 above, the range value amplitude vibration for inboard and outboard bearing are higher than the range value of amplitude vibration at inboard and outboard bearing before the unbalance weight is attached to the pulley at inboard bearing. This is actually the right value for the amplitude vibration because from the unbalance weight will increase the force that cause the increasing of amplitude vibration.

4.5 BALANCING

4.5.1 SINGLE PULLEY (center shaft)

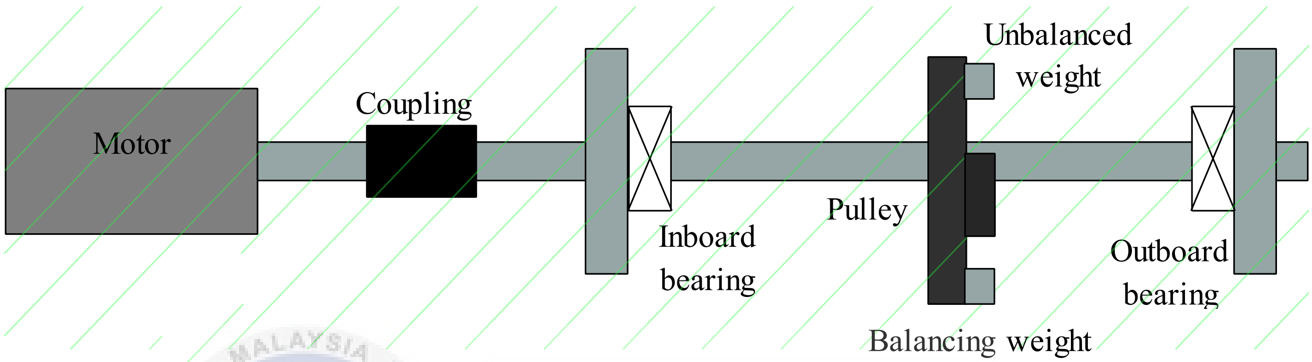


Figure 4.13: Schematic diagram of experiment runs with balance pulley at center shaft

4.5.1.1 Vector Method

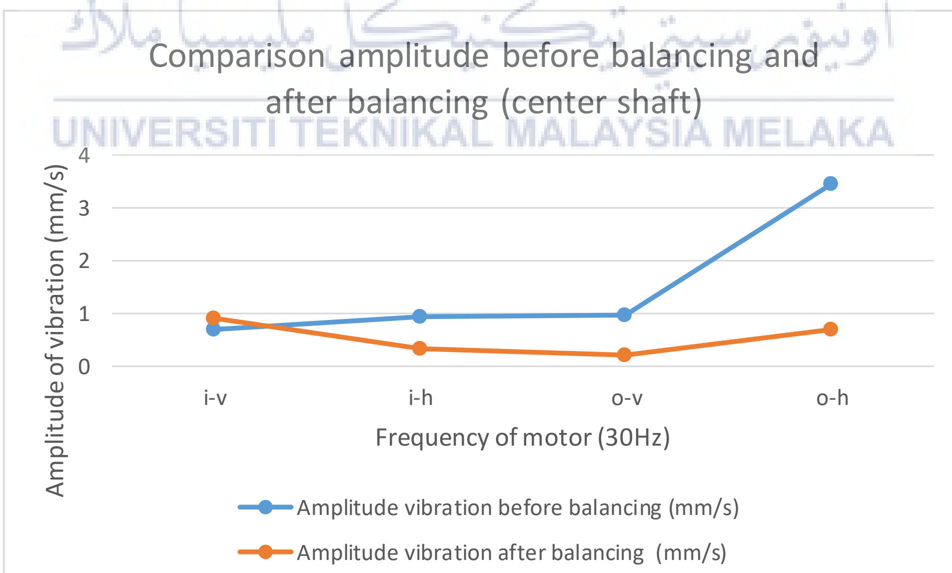


Figure4.14: Graph of amplitude before balancing and after balancing pulley at center shaft

Type of bearing	Amplitude original unbalanced (mm/s)	Phase angle (θ)	Amplitude with $7g$ trial weight (mm/s)	Phase angle (θ)	Amplitude after balancing (mm/s)
Inboard vertical	0.697575	270.25	1.193746	242.58	0.890627
Inboard horizontal	0.946097	37.18	1.091706	326.93	0.313255
Outboard vertical	0.954208	21.12	1.107883	291.14	0.210245
Outboard horizontal	3.451151	34.78	2.264806	312.30	0.678517

Table 4.11: Results balancing with balance pulley at center shaft in vector method

According to the table 4.11a nd graph figure 4.12 above, the outboard horizontal bearing shows the higher amplitude. Therefore, the amplitude is chosen for this method. Take scalar of 1 mm/s equals to 1 cm. In this polar plot, it is showing the anti-clockwise direction which due to the rotation of pulley. The O (original unbalanced) obtained 3.45 cm followed by the phase angle of 34.78° and O+T (original and trial) is 2.26 cm followed by 312°. By applying the vector method, the result for T value is 4.3 cm.

By applying the ratio:

1 cm = 1 mm/s

Then, 4.3 cm = 7 g So, for 3.45 cm = 7/4.3 × 3.45 = 5.6 g

Then, the balancing weight attached is must be less 1.4 g the balancing weight which 7 g- 1.4 g = 5.6 g After running the pulley, the results of balancing is 0.67 mm/s. The percentage of balancing is [1 – 0.67/3.45] × 100% = **80%** This percentage is higher with the 70% which is the standard in the industry.

4.5.2 SINGLE PULLEY (at inboard bearing)

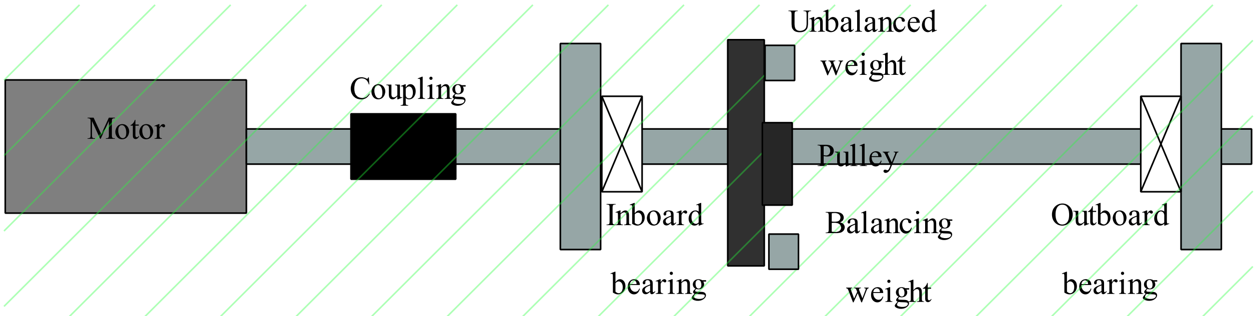


Figure 4.15: Schematic diagram of experiment runs with balance pulley at inboard bearing

4.5.2.1 Vector Method

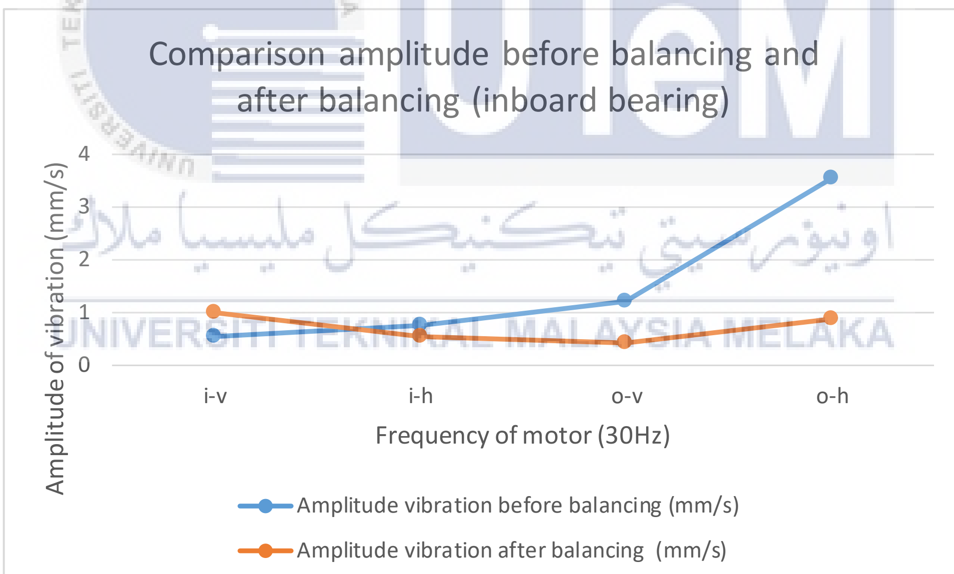


Figure 4.16: Graph of amplitude before balancing and after balancing pulley at inboard bearing

Type of bearing	Amplitude original unbalanced (mm/s)	Phase angle (θ)	Amplitude with $7g$ trial weight (mm/s)	Phase angle (θ)	Amplitude after balancing (mm/s)
Inboard vertical	0.531827	241.03	0.544304	220.23θ	0.988841
Inboard horizontal	0.762285	68.52	0.616944	97.16	0.532595
Outboard vertical	1.218200	54.90	1.268978	85.95	0.424666
Outboard horizontal	3.551396	56.11	3.301376	76.34	0.882360

Table 4.12: Results balancing with balance pulley at inboard bearing in vector method

According to the table 4.12 and graph figure 4.14 above, the outboard horizontal bearing shows the higher amplitude. Therefore, the amplitude is chosen for this method. Take scalar of 1 mm/s equals to 1 cm. In this polar plot, it is showing the anti-clockwise direction which due to the rotation of pulley. The O (original unbalanced) obtained 3.55 cm followed by the phase angle of 56.11° and O+T (original and trial) is 3.30 cm followed by 76.34°. By applying the vector method, the result for T value is 1.5 cm.

By applying the ratio:

1 cm = 1 mm/s

Then, 1.5 cm = 7 g So, for 3.55 cm = 7/1.5 × 3.55 = 16 g

Then, the balancing weight attached is must be add 9 g the balancing weight which 7 g +9 g = 16 g After running the pulley, the results of balancing is 0.88 mm/s. The percentage of balancing is [1 – 0.88/3.5] × 100% = **75%** This percentage is higher with the 70% which is the standard in the industry.

4.5.3 SINGLE PULLEY (at outboard bearing)

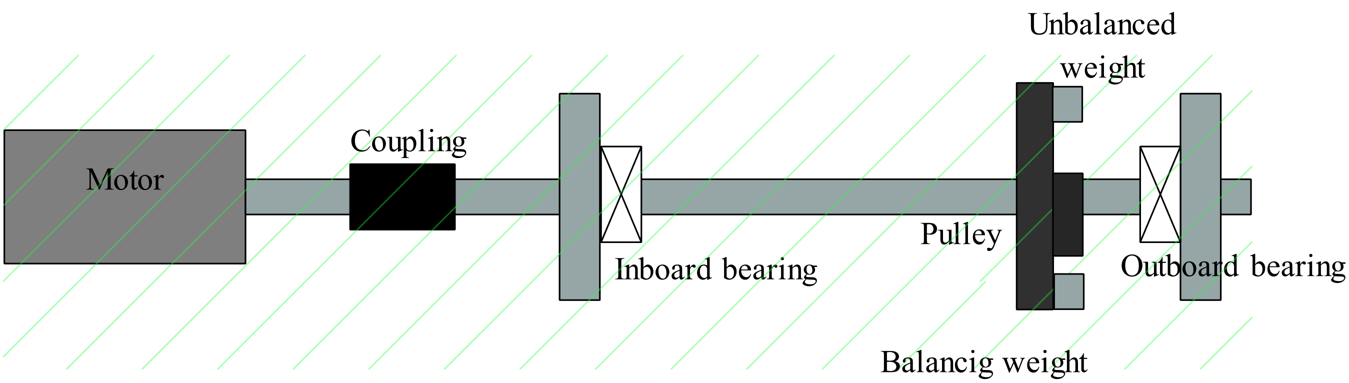


Figure 4.17: Schematic diagram of experiment runs with balance pulley at outboard bearing

4.5.3.1 Vector Method

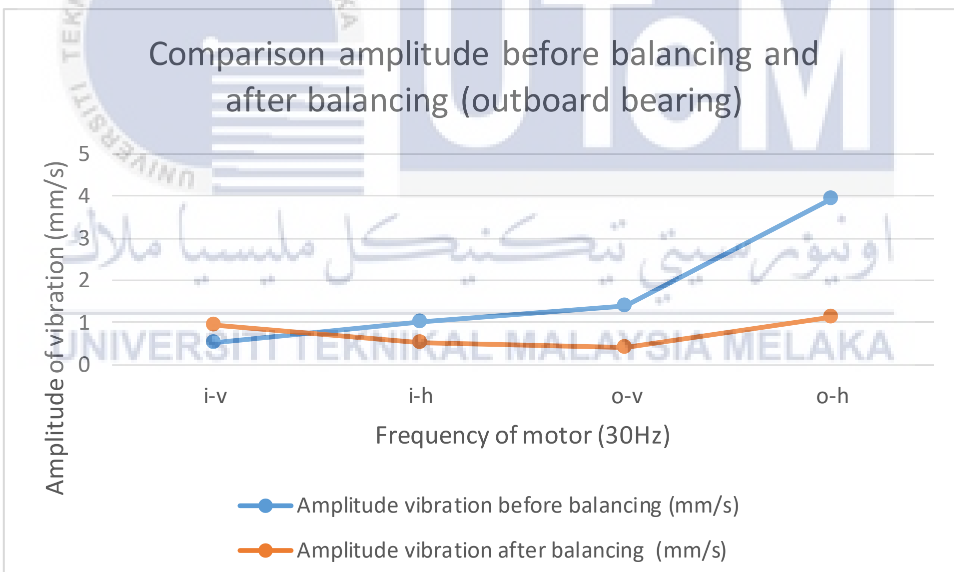


Figure 4.18: Graph of amplitude before balancing and after balancing pulley at outboard bearing

Table 4.13: Results balancing with balance pulley at outboard bearing in vector method

Type of bearing	Amplitude original unbalanced (mm/s)	Phase angle (θ)	Amplitude with $7g$ trial weight (mm/s)	Phase angle (θ)	Amplitude after balancing (mm/s)
Inboard vertical	0.527836	264.55	0.663227	238.88	0.954720
Inboard horizontal	1.014347	45.90	0.571434	45.91	0.531110
Outboard vertical	1.386937	29.13	0.823201	43.14	0.413541
Outboard horizontal	3.949582	36.93	2.704727	46.54	1.108302

According to the table 4.13 and graph figure 4.16 above, the outboard horizontal bearing shows the higher amplitude. Therefore, the amplitude is chosen for this method. Take scalar of 1 mm/s equals to 1 cm. In this polar plot, it is showing the anti-clockwise direction which due to the rotation of pulley. The O (original unbalanced) obtained 3.95 cm followed by the phase angle of 36.93° and O+T (original and trial) is 2.7 cm followed by 46.54°. By applying the vector method, the result for T value is 3.2 cm.

By applying the ratio:

1 cm = 1 mm/s

Then, 3.2 cm = 7 g So, for 3.95 cm = 7/3.2 × 3.95 = 9 g

Then, the balancing weight attached is must be add 2 g the balancing weight which 7 g +2 g = 9 g After running the pulley, the results of balancing is 1.11 mm/s. The percentage of balancing is [1 – 1.11/3.95] × 100% = **71%** This percentage is higher with the 70% which is the standard in the industry.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

Objective 1: "*To investigate unbalance pulley at various speed.*"

Based on this project, of unbalance pulley can be determined by carrying out some experiments which takes the parameter of different place of pulley at the shaft. From the result, as can be seen the percentage balancing for pulley at center of shaft is higher than pulley at inboard bearing and outboard bearing with 80% compared to 75% and 71% percentage balancing at inboard and outboard bearing. This is because the load attached from the pulley is at the center of shaft whereby the load attached is same to the inboard and outboard bearing that make the percentage balancing of pulley at center shaft is high. The purpose is to do research based on how a unbalanced pulley occur. It is explained that this unbalanced occurs when the inertial axis of a rotating mass is parallel to the axis of rotation. So, the findings of the experiment gives an empirical evidence that unbalance occur when a mass is added at the rotor. Besides that, the results shows the higher (1x) spectrum appear at the DAQ system. This result shows the unbalance occur when the amplitude of unbalance is higher than amplitude of baseline. From the figure 4.3 graph of amplitude against frequency of motor from 10 Hz to 50 Hz, it shows that unbalance might become more significant as seen from the figure that the amplitude increase while the frequency also increases that can detect where state of frequency the resonance is happen.

Objective 2: *"To do balancing on pulley using vector method"*.

The methods which for balancing is the vector method is applied. The results show the amplitude of vibration in balancing is lower than the (1x) spectrum of unbalanced pulley. This shows that the balancing experiment have successfully balanced the unbalanced pulley. In this project is to solve the problem by using the balancing method. Since the problem statement clarifies difficulties to find the unbalance at low RPM and the unbalance might become significant if the speed is increase to 3000 RPM therefore the pulley is runs with high frequency and RPM. There are also cases when the unbalanced mass is attached on the pulley in the same position and opposite direction. The position of balancing weight also need to be focused on. As conclusion the unbalance problem can effect and disrupted the performance and contribution of machine to the output product of some work process.

In the future study, there is need to focus more on the unbalanced and balancing in other conditions such as other material than pulley that are used in this experiment for example gear, sprocket,rotor disk and other. Other than that, use another parameter such as double rotor disk, pulley or other and adjust the distance between the rotor to find the unbalance and balance amplitude.

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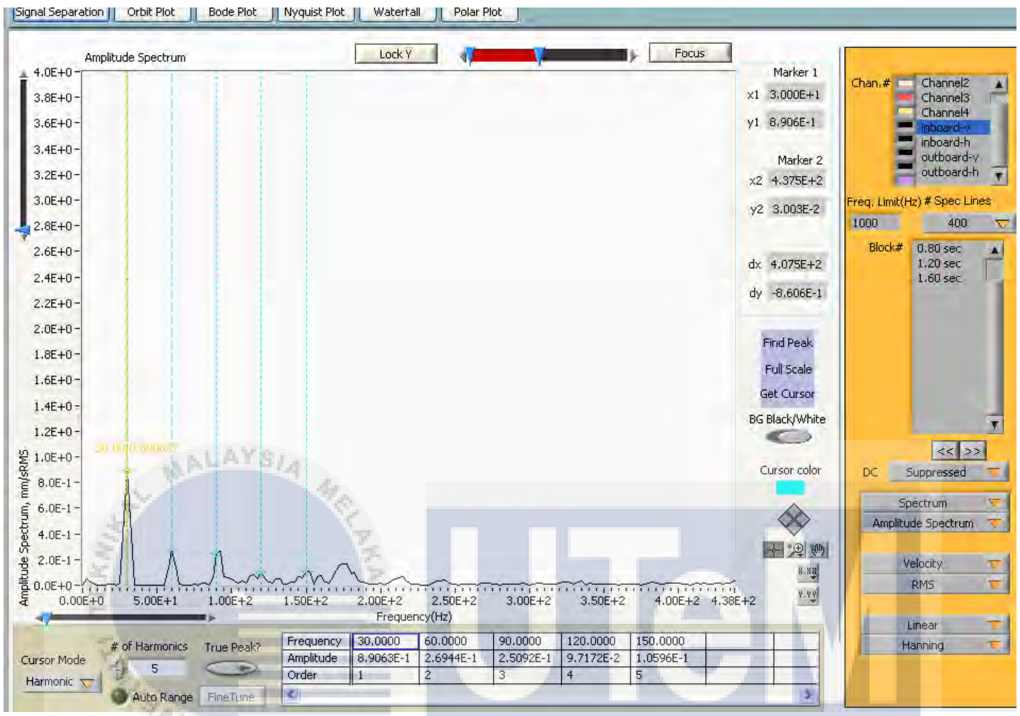
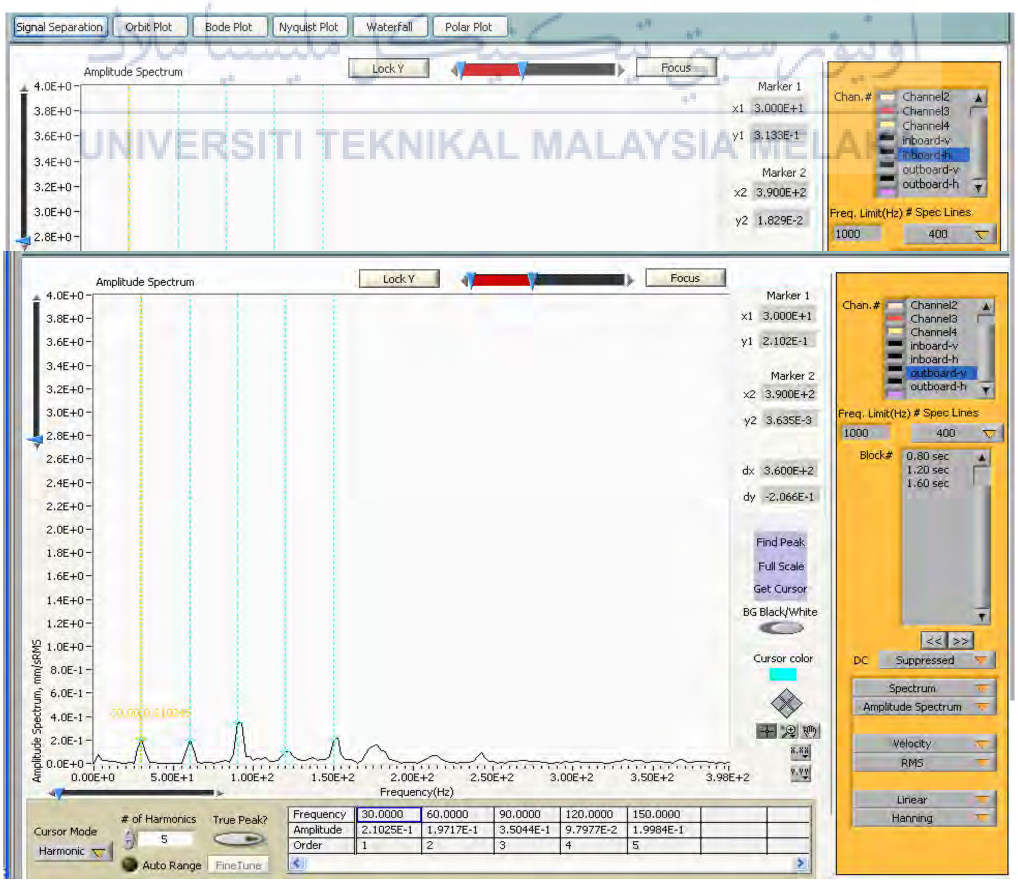
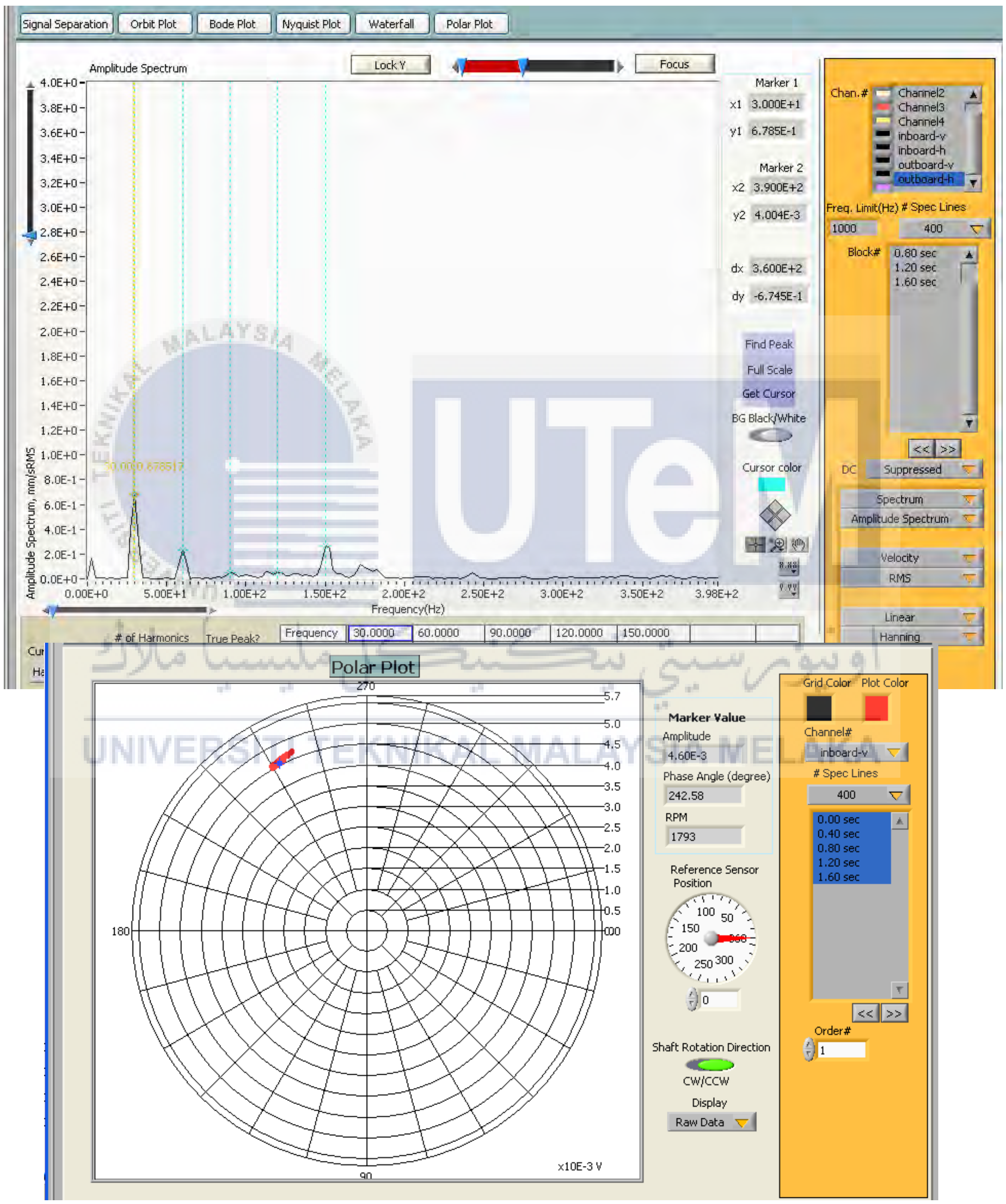
APPENDIX A

- Spectrum and polar plot diagram for balancing pulley at center shaft






Signal Separation
Orbit Plot
Bode Plot
Nyquist Plot
Waterfall
Polar Plot
Amplitude Spectrum
Lock Y
Focus
Marker 1
x1 3.000E+1
y1 6.785E-1
Marker 2
x2 3.900E+2
y2 4.004E-3
dx 3.600E+2
dy -6.745E-1
Find Peak
Full Scale
Get Cursor
BG Black/White
Cursor color
Chan.#
Channel2
Channel3
Channel4
inboard-v
inboard-h
outboard-v
outboard-h
Freq. Limit(Hz)
Spec Lines
1000
400
Block#
0.80 sec
1.20 sec
1.60 sec
DC
Suppressed
Spectrum
Amplitude Spectrum
Velocity
RMS
Linear
Hanning
Amplitude Spectrum, mm/sRMS
Frequency(Hz)
# of Harmonics
True Peak?
Frequency
30.0000
60.0000
90.0000
120.0000
150.0000
Polar Plot
270
180
90
x10E-3 V
Marker Value
Amplitude
4.60E-3
Phase Angle (degree)
242.58
RPM
1793
Reference Sensor Position
Shaft Rotation Direction
CW/CCW
Display
Raw Data
Grid Color
Plot Color
Channel#
inboard-v
Spec Lines
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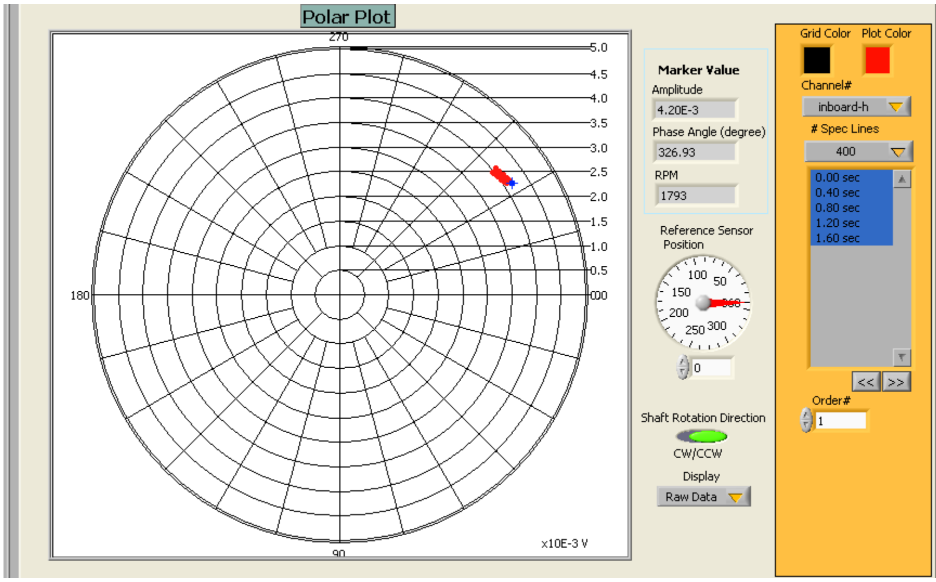
0.40 sec
0.80 sec
1.20 sec
1.60 sec
Order#
1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180
90
x10E-3 V
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4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
Marker Value
Amplitude
4.20E-3
Phase Angle (degree)
326.93
RPM
1793
Reference Sensor Position
Shaft Rotation Direction
CW/CCW
Display
Raw Data
Grid Color
Plot Color
Channel#
inboard-h
Spec Lines
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0.80 sec
1.20 sec
1.60 sec
Order#
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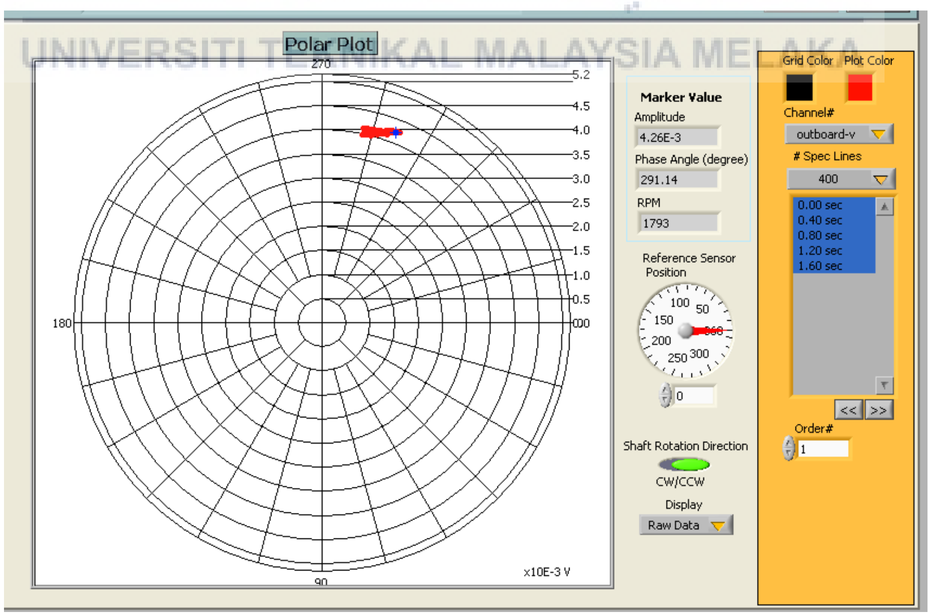
Polar Plot
270
180
90
x10E-3 V
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4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
Marker Value
Amplitude
4.26E-3
Phase Angle (degree)
291.14
RPM
1793
Reference Sensor Position
Shaft Rotation Direction
CW/CCW
Display
Raw Data
Grid Color
Plot Color
Channel#
outboard-v
Spec Lines
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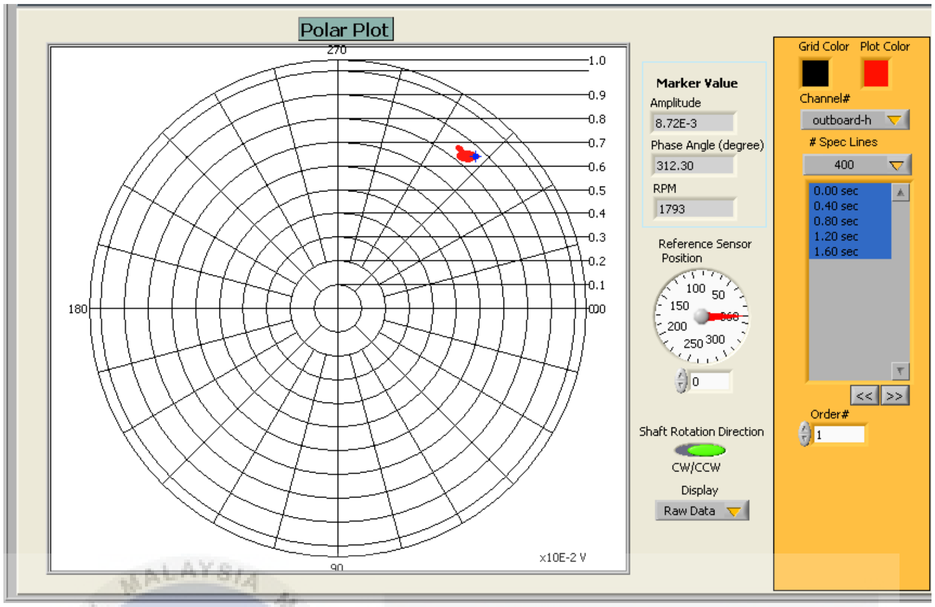
0.40 sec
0.80 sec
1.20 sec
1.60 sec
Order#
1



APPENDIX B

- Gant Chart for PSM 1

TASK WEEK	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Literature review															
Study of balancing for pulley equipment design requirements and its methodology															
Progress report submitted															
Progress of design pulley															
Analysis of balancing testing															
Report PSM 1 writing															
Report PSM 1 submitted															
Seminar presentation															

- Gant Chart for PSM 2

TASK WEEK	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Fabrication of pulley															
Running experiment with MFS machine															
Running until to 50 Hz to know whether the pulley can stand or not															
Determine whether there are resonance in the vibration															
Runs with three different parameter															
Analysis the spectrum diagram and use the right balancing method to balance it															
Writing report															
Submit the draft report															
Seminar PSM 2															
Complete report submit															

APPENDIX C

- Experiment setup