Vibration Control of Rotor-Shaft System With Electromagnetic Exciters

By

Muhamad Muaz Bin Maslan

Dissertation submitted in partial fulfillment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)
SEPTEMBER 2013

UniversitiTeknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

Vibration Control of Rotor-Shaft System With Electromagnetic Exciters

by

Muhamad Muaz Bin Maslan

A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

Approved by,

(Dr. Varaha Venkata Lakshmi Narasimha Rao Tad)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK

September 2013

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

MUHAMAD MUAZ BIN MASLAN

ABSTRACT

Reduction of rotor vibration is very important for safe and efficient functioning of rotating machines. Active vibration control system is used to control transverse vibration of rotating shaft equipment. Electromagnetic exciters are mounted on the stator plane, away from the conventional support location, around the rotor shaft for applying suitable force of actuation over an air gap to control transverse vibration. Electromagnetic used for vibration control do not levitate the rotor and facilitate the bearing action. Which is provided by conventional bearings.[4] By varying the control current in the exciters, suitable force of actuator can be achieved. This provides control force over an air gap and hence is free from difficulties of maintenance, wear and tear and power loss. Preliminary theoretical simulation using linearized expression of electromagnetic force and the accompanying example show good reduction in transverse response amplitude, postponement of instability caused by viscous form of rotor internal damping as well as great reduction of support forces.

ACKNOWLEDGEMENT

First and foremost, I would like to thank my project supervisor, Dr. V.V.L.N Tadimala Rao for guiding me incompleting this project. Apart from him, I would also like to thank my Final Year Project I external examiner Dr Setyamartana Parman who had evaluated my presentation and gives comments and suggestion for the bettermerment of my project.

Not to forget Dr. Nagarajan Thirumalaiswamy who had evaluated my Final Year Project II poster presentation and gave comments to improve my project upon completion. Besides, I would also like to thank everyone that I have worked with especially my colleagues for assisting me during completion of the project.

Lastly, I would like to thank lecturers from UTP Mechanical department who had thought me throughout the undergraduate studies and without them I won’t be able to complete my Final Year Project successfully.

TABLE OF CONTENTS

CERTIFICATION OF APPROVAL ii
CERTIFICATION OF ORIGINALITY iii
ABSTRACT iv
ACKNOWLEDGEMENT v

CHAPTER 1 : INTRODUCTION

1.0 Introduction 1
1.1 Background Study 3
1.2 Problem Statement 4
1.3 Objectives 6

CHAPTER 2 :LITERATURE REVIEW

2.0 Magnetic Exciters Background 8
2.1 Working Principle of Magnetic Exciters 12
2.2 Application of magnetic exciters in industry 16
2.3 Calculation Involved 17
2.4 Vibration Analysis 20

CHAPTER 3 : METHODOLOGY

3.0 Research Methodology 23
3.1 Key milestone 25
3.2 Project Gantt Chart 26

CHAPTER 4 : RESULT AND DISCUSSION

4.0 MATLAB Simulation 28
4.1 SIMULINK Simulation 32

CHAPTER 5 : CONCLUSION AND RECOMMENDATION

5.0 Conclusion 34
5.1 Recommendation 35
REFERENCES 36

LIST OF FIGURES

Figure 1 : Stator, Wirecoils, and Rotor Configuration

Figure 2 : Components of The Active Magnetic Actuator

Figure 3 : Inertial Disk and Eddy Current Probe Displacement Sensor

Figure 4 : Keyphase Sensor Used to Determine The Phase

Figure 5 : Test Rig Diagram

Figure 6 : Position of Magnetic Exciters Housing on V-shape Mount

Figure 7 : Magnetic Exciters Control System Block Diagram

Figure 8 : Spring Mass Damper Model and Feedback Control System Model on Rotating Eq.

Figure 9 : Physical Model of The Rotor

Figure 10 : Model of Unbalance Force (Rotational String-Stone)

Figure 11 : Slow Rotation Rotor Configuration

Figure 12 : Bode Plot Diagram

Figure 13 : Phase Diagram for Slow Rotating Rotor

Figure 14 : Phase Diagram for Approaching Resonance Rotor Speed

Figure 15 : Air Gap Between Stator and Rotor During Approaching Resonance Rotor Speed

Figure 16 : Project Methodology Flow Chart

Figure 17 : Project Key Milestone Flow Chart

Figure 18 : Amplitude vs. Time Graph Without Magnetic Exciters

Figure 19 : Amplitude vs. Time Graph With Magnetic Exciters

Figure 20 : Bode Plot Without Magnetic Exciters

Figure 21 : Bode Plot With Magnetic Exciters

Figure 22 : SIMULINK Block Diagram

Figure 23 : Amplitude vs. Time With Magnetic Exciters

Figure 24 : Amplitude vs. Time Without Magnetic Exciters

LIST OF TABLE

Table 1 : Project Gantt Chart

CHAPTER 1

INTRODUCTION

1.0 Introduction

The history of rotor dynamics begin back in the year 1869 where W.J.M Rankine first performed an analysis of a spinning shaft which he replete with the interplay of theory and practice. Unfortunately, his model was not adequate and cannot attained supercritical speeds as he predicted. Later on the year 1895 published an experimental paper describing supercritical speed which cannot be attained by W.J.M Rankine. In 1889, Gustaf de Laval, a Swedih engineer ran a steam turbine to supercritical speeds in 1889 and Kerr published a paper hat showed the experimental evidence of a second critical speed in 1916.

Royal Society of London has commissioned Henry Jeffcott to solve the conflict between theory and practice. Later he published a paper which now is considered classic in the Philosophical Magazine in 1919 in which he confirmed the existence of stable supercritical speed.

World War II start between the work of Jeffcott and Prohl and Nils Otto Myklested model technique culminating which led to the Transfer Matrix Method (TMM) for analyzing rotors. Today the most common method we used for rotordynamics analysis is the Finite Element Method.

The history of controlling rotor-shaft vibration using magnetic exciters begin with the invention of magnetic bearing as an alternative for the conventional bearing. Jesse Beams from the University of Virginia filed some of the earliest active magnetic bearing patents during World War II. However, magnetic bearings did not mature until the invention of modern art computer based control technology. The first commercial application of active magnetic bearing was in turbomachinery. With the evolution of active magnetic bearing, the research of the usage of magnetic exciters to control rotor shaft vibration has been developed.

Reduction of machine vibration is very important for safe and efficient functioning of turbomachinery in process industry in which fault free operation is demanded. As an example, the impellers of a centrifugal compressor impart work to the gas to increase its pressure. Ensuring stability is critical to the cost effective installation and operation of these machines in industry. Many new compressor designs have experienced unexpected and damaging instabilities, resulting in significant production downtime and production loss. As the gas pressure in the compressor increasing, the dynamic behaviors of shaft and impeller seals, axial thrust balance pistons, and impellers under the pressure become more significant. Although the accurate prediction of centrifugal compressor stability continues to be an important area of interest in the oil and gas industry, current industry standards and tools for the prediction of impeller destabilizing forces are based on empirical methods that, to date, have served fairly well for systems with reasonable stability margins. However, as stability margins are decreased, use of a more effective controlling method that is physics based and can better improve the stability margins is required

The usage of magnetic exciters to control vibration involve the usage of convention bearing such as oil film, roller and ball bearing to support the rotating shaft. Due to the unbalance force on the rotating shaft on conventional bearing, the magnetic exciters take action to eliminate the unbalance force which later will be discussed on the next chapter

1.1 Background Study

In this paper, the cause of vibration, the purpose to reduce vibration and method to reduce vibration using magnetic actuator on rotor-shaft system will be explained. In the problem statement section, explaination regarding the problem facing by rotating equipment especially vibration, will be presented. The history of the development od magnetic exciters used in rotor-shaft sytem will be presented to ensure w that appreciate the revolutionaries contribution of previous engineers and researches. Magnetic exciters have been used widely before in many kind of engineering field. With the development of computational method and engineering tools, complicated equation and problem regarding vibration can be simulated and solved. Magnetic exciters will be used to reduce the vibration and facilitate the conventional bearing. Analytical result based on the simulation is presented on graph and discussed in the next chapters onwards. Amplitude vs time graph and bode plot will be analyzed based on the input parameters of actual test rig. Simulation has been conducted using MATLAB and Simulink software to simulate the effect of magnetic exciters in controlling vibration of rotor-shaft system. From the result, we can see clearly the need of vibration control system in order to reduce the vibration for rotor-shaft system. The coding and modeling is improvised from the existing coding to simulate this system. Methodology of how this simulation is conducted is presented in flow chart diagram and at the end of this paper, recommendations are suggested on how to improve the simulation and the improvement that can be made on the coding and model.

1.2 Problem Statement

Vibration can result from a number of conditions, acting alone or in combination. Keep in mind that vibration problems may be caused by auxiliary equipment, not just the primary equipment.. Typically, 80% of the machinery problems experienced can of the be classified as either imbalance or misalignment. Imbalance and misalignment can lead to premature bearing, coupling, shaft seal, and gear wear. Most of the problems can be rectified by simply improving maintenance standards and procedures and by eliminating careless or sloppy work. Also, imbalance and misalignment do not only occur in establishment equipment over a period of time, they can be present after initial installation of a new piece of machinery.

Besides, it is important to note that the bearings in a machine-train are the primary limiting factor for operating life. The first indication of machinery problems often develops in the vibration signature of the machine bearings. However, the bearings are typically not the only cause of the problem. But since bearings are the link in most machinery, it usually the first to fail. Vibration checks at points other than the bearings are also taken to check for structural problems.

These are few major causes of vibration :

Imbalance – A “heavy spot” in a rotating component will cause vibration when the unbalanced weight rotates around the machine’s axis, creating a centrifugal force. Imbalance could be caused by manufacturing defects (machining errors, casting flaws) or maintenance issues (deformed or dirty fan blades, missing balance weights). As machine speed increases, the effects of imbalance become greater. Imbalance can severely reduce bearing life as well as cause undue machine vibration.

Misalignment/shaft run out – Vibration can result when machine shafts are out of line. Angular misalignment occurs when the axes of (for example) a motor and pump are not parallel. When the axes are parallel but not exactly aligned, the condition is known as parallel misalignment. Misalignment may be caused during assembly or develop over time, due to thermal expansion, components shifting or improper reassembly after maintenance. The resulting vibration may be radial or axial (in line with the axis of the machine) or both.

Wear – As components such as ball or roller bearings, drive belts or gears become worn, they may cause vibration. When a roller bearing race becomes pitted, for instance, the bearing rollers will cause a vibration each time they travel over the damaged area. A gear tooth that is heavily chipped or worn, or a drive belt that is breaking down, can also produce vibration.

Looseness – Vibration that might otherwise go unnoticed may become obvious and destructive if the component that is vibrating has loose bearings or is loosely attached to its mounts. Such looseness may or may not be caused by the underlying vibration. Whatever its cause, looseness can allow any vibration present to cause damage, such as further bearing wear, wear and fatigue in equipment mounts and other components.

Many methods have been developed to reduce the unbalance-induced vibration by using different devices such as active balancing devices, electromagnetic bearings, active squeeze film dampers, lateral force actuators, pressurized bearings and movable bearings.[9] Active magnetic actuator changes the dynamical properties of the system by using actuators or active devices during instantaneous operating conditions measured by the appropriate sensors. The main advantage of active control (compared to passive control) is the versatility in adapting to different load conditions, perturbations and configurations of the rotating machinery and hence, extending the system's life while greatly reducing operating costs.[8]

1.3 Objective

The objective of this project is to control the vibration of rotor-shaft system with electromagnetic exciters. Active vibration control system is used to control transverse vibration of rotating shaft equipment. Electromagnetic exciters are mounted on the stator plane, away from the conventional support location, around the rotor shaft for applying suitable force of actuation over an air gap to control transverse vibration. Electromagnetic used for vibration control do not levitate the rotor and facilitate the bearing action. Which is provided by conventional bearings. By varying the control current in the exciters, suitable force of actuator can be achieved. This provides control force over an air gap and hence is free from difficulties of maintenance, wear and tear and power loss.[6] Preliminary theoretical simulation using linearized expression of electromagnetic force and the accompanying example show good reduction in transverse response amplitude, postponement of instability caused by viscous form of rotor internal damping as well as great reduction of support forces.[5]

1.4 Scope of Study

In order to accomplish the project within the given time frame, there are few scope of study which are :

a. To understand what causes vibration on rotor-shaft rotating system.
b. Method to reduce the vibration on rotor-shaft rotating system by using electromagnetic exciters.
c. To understand the basic working principle of electromagnetic exciters.
d. Application of electromagnetic exciters on rotating equipment.
e. Simulation of rotor shaft system with and without using magnetic exciter (MATLAB and Simulink).

By listing out the scope of study, it will be easier to focus on the project development. For this project, we have to know first, what is the cause of vibration on rotor-shaft rotating system. Then, we can proceed to focus on the method to reduce vibration by using magnetic exciters. This project requires understanding on the working principle of electromagnetic exciters. By understanding the working principle, we can know the advantages of this magnetic bearing compared to the conventional bearing in order to reduce vibration on rotor-shaft system.

In industries nowadays, a lot of rotating equipment use magnetic exciters in order to reduce vibration. In this project, the application of magnetic exciters will be highlighted together with the engineering field that use it.

After understand the principle of magnetic exciters, only then we can workout on the operation and simulation.

CHAPTER 2

LITERATURE REVIEW

2.0 Magnetic Exciters Background

Magnetic bearing is a type of electromagnetic bearing that use the principle of levitation to support load without any physical contact. This levitation principle levitate a rotating shaft and enable relative rotation with very low friction and no frictional wear. Currently develop magnetic bearing has no maximum relative speed and can support the highest speeds of all kinds of bearing. [2]

This magnetic actuator works on the theory of conversion from electrical signal to a mechanical output (displacement). Thus, the mechanical output will provide levitation to the rotor shaft. Electromagnetic actuators can be used to apply forces to a rotor without contact. These forces may be controlled by an independent external source, or by rotor vibration. Thus the actuators can be used as a contactless excitation device as an active damper or as an active damping.

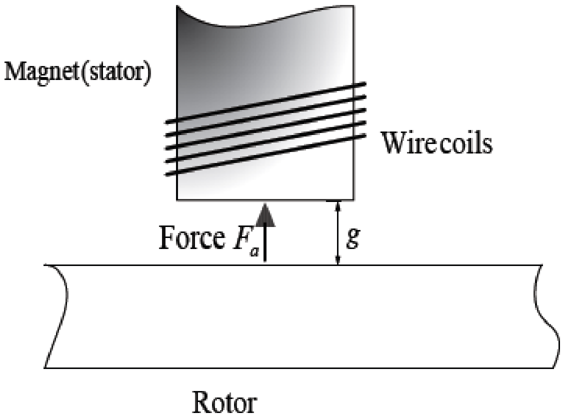


Figure 1 : Stator, Wirecoils, and Rotor Configuration

The AMA system consists of a stator, a rotor with an internal collect, position sensors, a control system and power amplifiers. The rotor, which is made of laminated iron, is attached the shaft with the collet. The magnetic actuators and sensors are located on opposite sides of the rotor in two perpendicular control axes. The control system and amplifiers are located in a separate housing and connected by wires. The stator is also made of laminated steel with poles on the internal diameter. Wire coils are wound around each pole, so that the actuator is divided into four quadrants, each having two poles. In our case, quadrants are aligned strictly vertically and horizontally. Opposing quadrants constitute an axis and, therefore, each actuator can be described by two perpendicular axes. So each actuator's axis has a pair of amplifiers to provide current to generate an attractive magnetic force to correct the position of the rotor along that particular axis. The amplifiers, which are of pulse with modulation (PWM) type, are high voltage switches that are turned "on" and "off" at a high frequency, to achieve a current in the coils requested by the controller. The stator and rotor are the active bearing elements used to apply force to the shaft.[1]

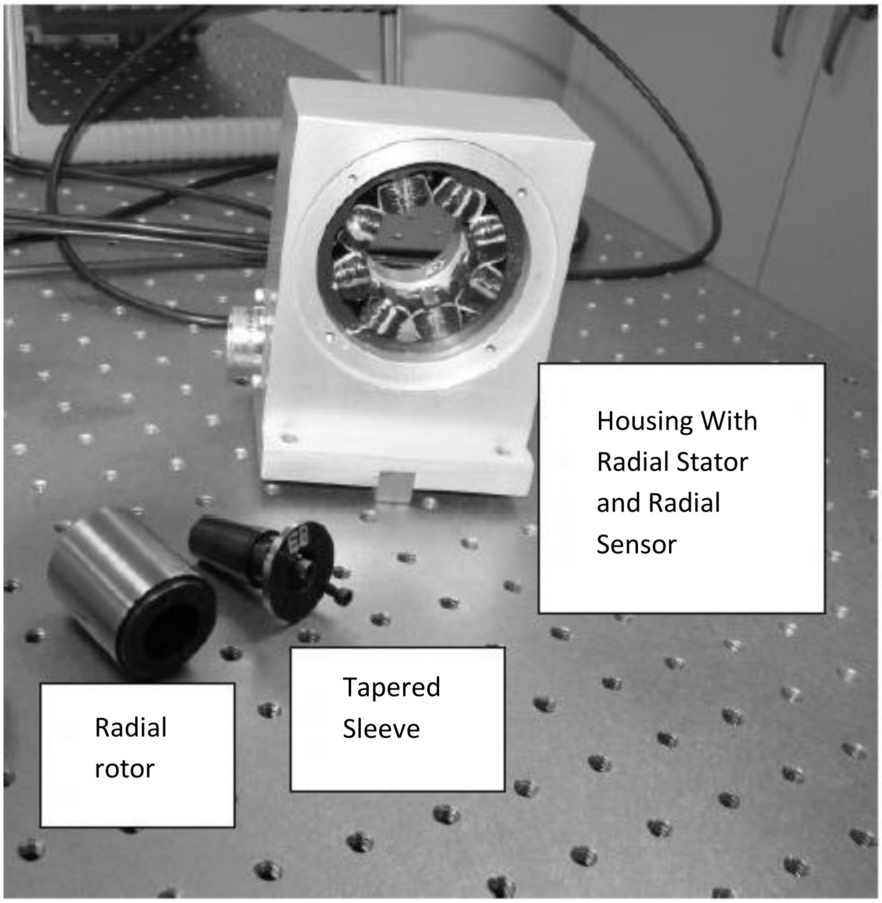


Figure 2 : Components of The Active Magnetic Actuator [1]

The application of magnetic actuators is based upon the principle that an electromagnet will attract ferromagnetic material. The sensor ring measures radial position of the shaft and is mounted as close to the actuator as possible. The sensors feed information about the position of the shaft to the controller in the form of an electrical voltage. See Figure, which shows the components disassembled.

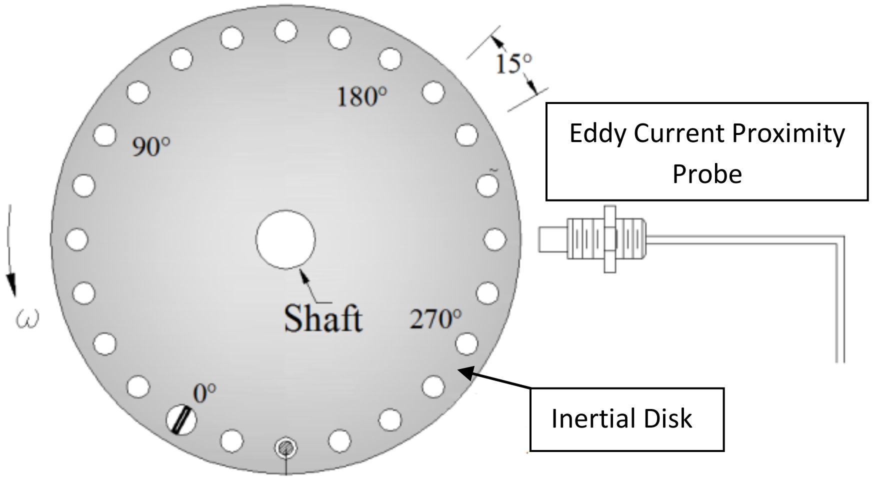


Figure 3 : Inertial Disk and Eddy Current Probe Displacement Sensor

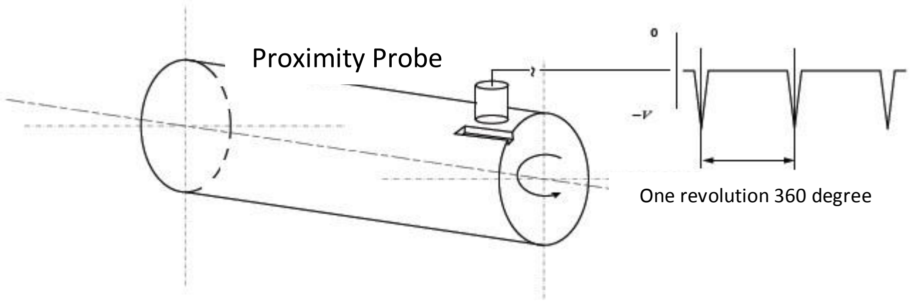


Figure 4 : Keyphase Sensor Used to Determine The Phase

The speed is controlled by feedback pulses from speed sensors, which observes a 20- notch wheel mounted on the rotor coupling.[10] The motor controller let us choose desired rotational speed that can be selected from 250 rpm to 10,000 rpm. In addition, the speed set point can be set to ramp up or downward at a rate of up to 15,000 rpm/min. The radial displacement of the shaft was measured in two planes (vertical and horizontal) with eddy current transducers or proximity probes. Proximity probes measure distances between 0.254 mm (10 mils) and 2.28 mm (90 mils). The proximity probes signal is generated by measuring voltage changes in the proximity probes circuit. The Keyphasor let us know the angular location (phase) of the shaft vibration response in relation to the physical location of the event (Figure 4).[8]

2.1 Working Principle of Magnetic Exciters

Rotor is supported by two bearings. The rotor is driven by an electrical motor with a separate controller. The rotor is attached to the driver and in this case electric motor is used by a flexible aluminium coupling, which also incorporates speed sensors for motor and a Keyphasor for rotor's angle determination.

The speed of the rotor is controlled by feedback pules from the speed sensors, which observes a 20-notch wheel mounted on the rotor coupling. The motor controller will let the user to choose desired rotational speed that can be selected. The radial displacement of the shaft due to vibration is measured in two planes (vertical and horizontal) with eddy current transducers or proximity probes. Proximity probes measure distances between 0.254mm and 2.28 between the rotating shaft and stator. The proximity probes signal is generated by measuring voltage changes in the proximity probes circuit. The Keyphasor sensor will let us know the angular location (phase) of the shaft vibration response in relation to the physical location of the rotating shaft.

The active magnetic exciter system consists of :

a. stator
b. rotor
c. position sensor
d. touchdown bearing
e. control system
f. power amplifier

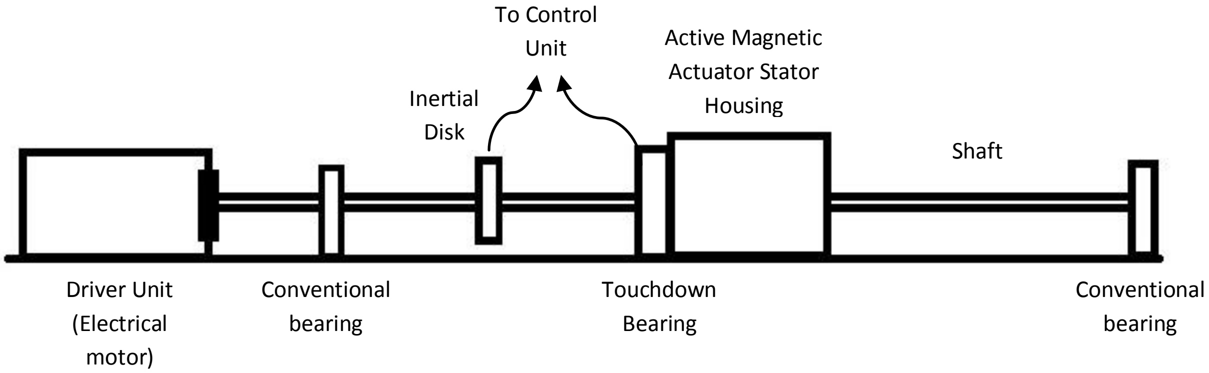


Figure 5 : Test Rig Diagram

The rotor, which is made up of laminated iron, is attached the shaft with the collet. The magnetic actuators and sensors are located are located on opposite sides of the rotor in two perpendicular control axes while the control system and amplifiers are located is separate housing and connected by wires. The stator is also made of laminated steel with poles on the internal diameter. Wire coils are wound around each pole and the actuator is divided into four quadrant, each having two poles. In this case, those four quadrants are aligned vertically and horizontally to apply force on the rotating shaft when feedback signal sends the whipping amplitude. Each actuator has a pair of amplifiers to provide current to generate an attractive force in order to correct the position of the rotor along that two x and y perpendicular axis. The puls-width-modulated (PWM) type are high voltage switch that are turned "on" and "off" at a high frequency, to achieve a current in the coils requested by the controller. Fundamentally, the application of magnetic exciters are based on the principle that an electromagnet will attract ferromagnetic material. In order to measure the radial position of the shaft, sensor ring is mounted as close to the exciter as possible. The sensor function to feed information about the position of the shaft to the controller in the form of an electrical voltage. The other part for safety and to protect the magnetic poles when shaft vibration exceeds the admissible level is the touchdown bearing. The bearing type used for touchdown bearing is deep groove ball bearing which is mounted in the actuator housing. The clearance between the shaft and the touchdown bearing is approximately half of that between the magnetic bearing rotor and stator.

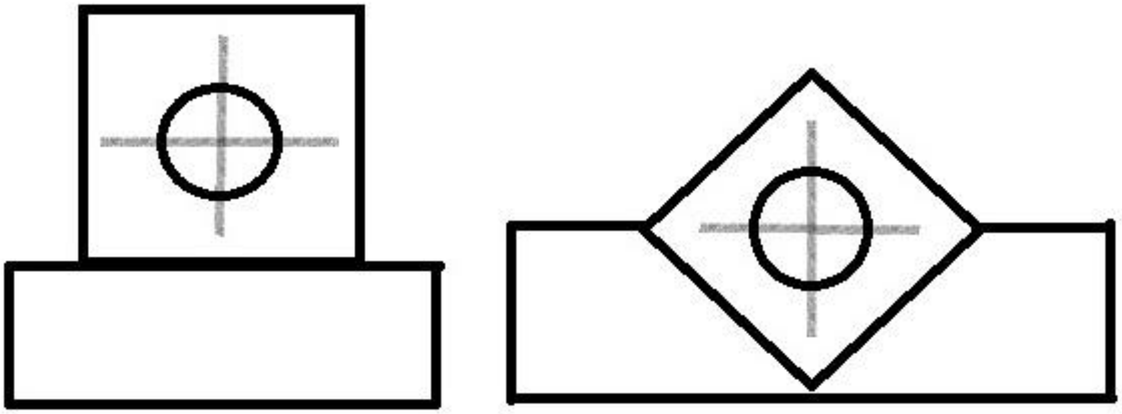


Figure 6 : Position of Magnetic Exciters Housing on V-shape Mount

A V-shape base mount is used for the mounting the magnetic exciter as the V-shape base is beig perturbed by an unbalance force in the X-Y plane, there always is a harmonic force applied in the vertical and horizontal directions, no matter where perturbation plane is. Since the V-shape base is symmetric about the Y-Z plane, only vertical and axial offsets will cause angular moments. The axial location of the unbalance force is due to the location of the components of the rotor. In order to minimize these moments as much as possible, the mounting was designed to keep the center of mass of the rig as close to the axis of rotation as possible.

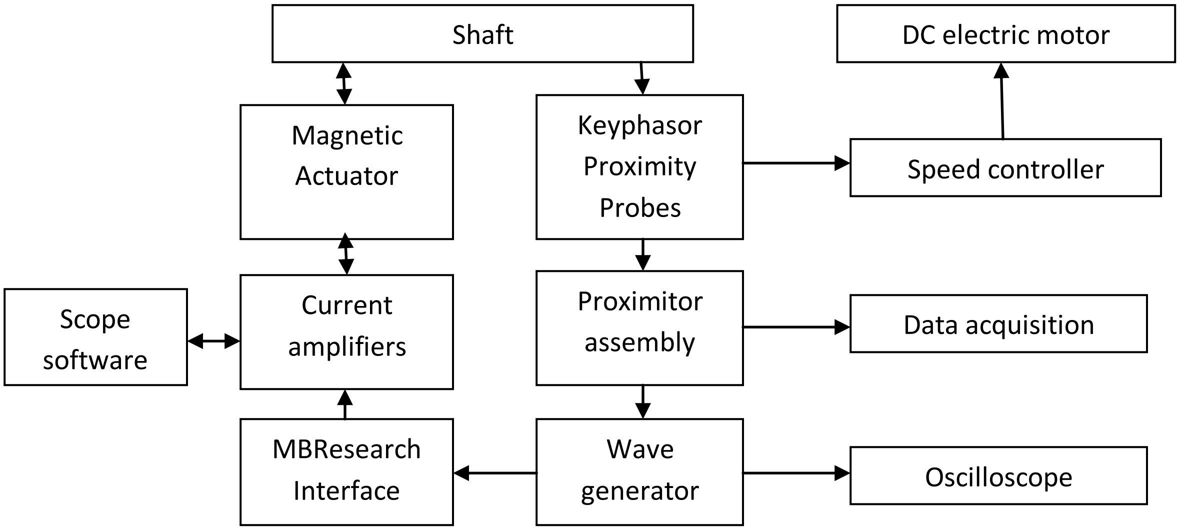


Figure 7 : Magnetic Exciters Control System Block Diagrem

Wave generator generate and inject sine waves in the vertical and horizontal directions. This kind of wave generator used is a precision source of sine, triangle square, positive pulse and negative pulse waveforms as well as a DC voltage supply. The waveform frequency is manually or remotely variable from 100μHz to 5μHz.

Wave generators are connected to the current amplifiers controller through MBResearch interface board. The MBResearch interface provides access to signals from the magnetic actuator controllers. The signals available include analog current and position signals and tp dead centre (TDC) pulse signal to monitor speed and phase. MBResearch also provides the ability to inject signals on all control axes for research purpose of control theory and rotor dynamics.

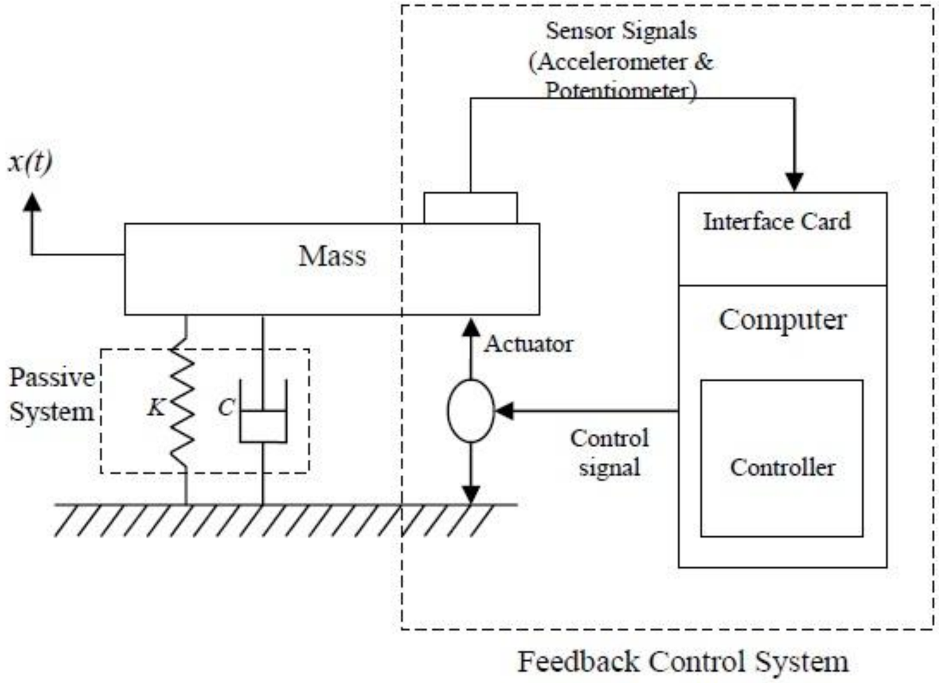


Figure 8 : Spring Mass Damper Model and Feedback Control System Model on Rotating Equipment

An active vibration control is a method that relies on the use of external power source called actuator. The actuator will provide a force or displacement to the system based on the measurement of the response of the system using feedback control system. An active vibration control system working principle starts with the measuring the response of the system using suitable sensors. The electronic circuit reads the sensors output, later which will convert the signal and sent it to the control unit. Based on the control law used, the calculated force signal is sent to the actuator and the controlled force is correspondingly applied to the system. The actuator force will eventually compensate the vibration force on the system. The passive system in this case represent the vibration on the rotating shaft which is denoted by the stiffness and damping of the oil film bearing.

2.2 Application of magnetic exciters in industry

The history of magnetic exciters to control rotor-shaft vibration starts with the invention and usage of Active Magnetic Bearing in industries back in the 90’s. The usage of AMB depends solely on magnetic bearing without the usage of conventional bearing. Instead, this magnetic exciters system still use conventional bearing and the magnetic exciters come into function in order to eliminate the rotor-shaft vibration.

AMB is used for rotor-shaft system that requires critical operation such as deepwater compressor, pump and high speed motor. Same goes to the magnetic exciters that are usd to eliminate rotor-shaft vibration. This application magnetic exciters on rotor-shaft is under research and development where testing is done on test rig. No clear paper, journal or any brochures in the market that manufacture and sell this kind of system so far.

Here is one numerical investigation that magnetic exciters can be potentially be implemented and applied. Ammonia is one of basic materials for industry and agriculture. Synthesis gas composed of nitrogen and hydrogen gas is pressurized through centrifugal compressor and sent to a reactor to produce ammonia. Because of its low density, more stage or high rotating speed is needed in this type of compressor than high density gas needed for reaching the same pressure ratio. Thus, rotors in these compressors are more flexible and consequently with lower stable margin. According to Alford equation, the dynamic behaviors of seal at shaft end and impeller eye, axial thrust balance pistons, become significant. They are related to pressure ratio, component of gas, and power of each stage. With low stable margin, any change in these three parameters may lead the compressor to unstable state which is characterized by unacceptable sub-synchronous vibration.

2.3 Calculation Involved

The vibration of rotating rotor shaft is subjected to to an external force is considered. In particular, the response to harmonic excitations, impulses and step forcing functions is examined.

In many environments, rotating machinery, motos, and so on cause periodic motions of structures to induce vibrations into the other mechanical devices and structures nearby. It is common to approximate the driving forces $F(t)$, as periodic of the form.

$$F(t) = F_o \sin \omega t$$

Where F_o represents the amplitude of the applied forc and ω represents the frequency of the applied force, or the driving frequency (rad/sec). On summing the forcs, the equation for the forced vibration of the system becomes.

$$m\ddot{x} + c\dot{x} + kx = F_o \sin \omega t$$

m= mass of the rotor

c= damping coefficient of the vibration system

k= spring constant of the vibration system

The attractive magnetic force, F applied to the rotor is determined by the formula :

$$F_a = \frac{\varepsilon \mu_0 A_g N^2 I^2}{4g^2}$$

ε = geometric correction factor

μ = permeability of the air gap

A_g = Single pole face area

N = Total number of wire coils in a horseshoe

I = Current in the coil

g = Gap air distance

The equations of motion, with present unbalance force, take the following format:

$$M\ddot{x} + D\dot{x} + Kx = Fcos\ \omega t - \alpha$$

$$M\ddot{y} + D\dot{y} + Ky = Fsin\,\omega t - \alpha$$

Where $M\ddot{x}, M\dot{y}$ represent inertia forces in two orthogonal lateral directions, $D\dot{x}, D\dot{y}$ - damping forces, Kx, Ky - stiffness and F - exciting force (in our case it is unbalance force). In order to cancel the rotor vibrations we need to cancel the exciting force.

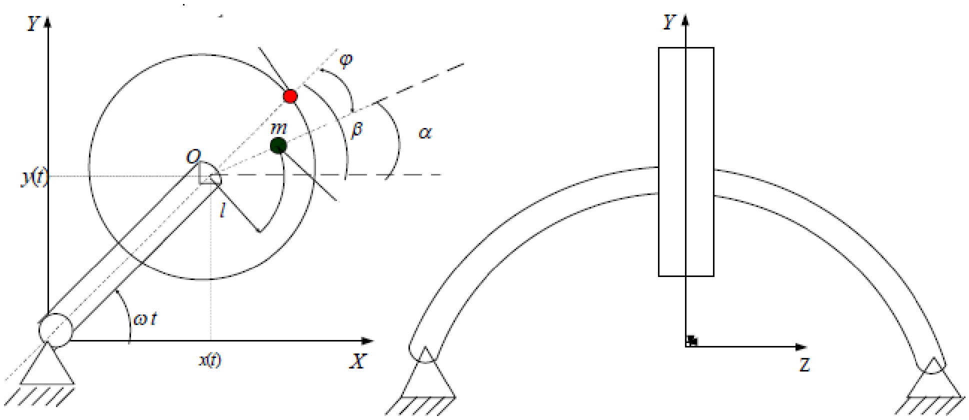


Figure 9 : Physical Model of The Rotor

Cancellation of the exciting force is possible by applying the same harmonic force but in opposite direction.[11] As an instrument to do this an active magnetic actuator (AMA) was chosen. An active magnetic actuator is a mechatronic device that uses electromagnetic fields to apply forces to a rotor without contact. The advantages of using active magnetic actuators are already well known. Their very low friction, virtually limitless life, insensitivity to surrounding environment and relatively large changes in temperature, and flexibility due to digital computer control, gives them extraordinary versatility. The effectiveness of the active magnetic actuators is based on the nature phenomena of attractive forces that are generated by magnets.[12]

This unbalance force on rotating equipment is caused due to the misalignment of the centre mass (inertia axis) and the centre of rotation (geometric axis). When the shaft is forced to spin about the fixed axis where the mass is not evenly distributed, then we have unbalance. Unbalance causes a moment which gives an object the wobbling movement characteristic of the vibration of rotating structures. The rotor can be considered as in unbalance condition when its centre of mass doesn’t coincide with the centre of rotation.

Calculating the imbalance applied force on rotor shaft :

$$F = I_m r \omega^2$$

F = Imbalance applied force

I_m = Mass of the rotor shaft

R = Distance from the pivot (shaft radius)

ω^2 = Shaft angular frequency (radian/sec)

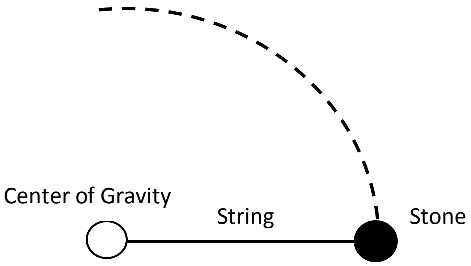


Figure 10 : Model of Unbalance Force (Rotational String-Stone)

It is seen that the force on the pivot is proportional to the distance from the center of rotation and to the speed squared. A rotor which has a heavy spot is not exactly equivalent to the stone on a string. In this case of the stone, the center of gravity of the system is the center of the stone itself, whereas the center of gravity of a rotor with imbalance is outside the imbalance mass and is near the axis of rotation of the rotor.

If the structure holding the bearings in the system is rigid, the center of rotation is constrained from moving, and the entripetal force that cause from the imbalance mass can be found from the above formula. The force is borne by the bearings.

2.4 Vibration Analysis

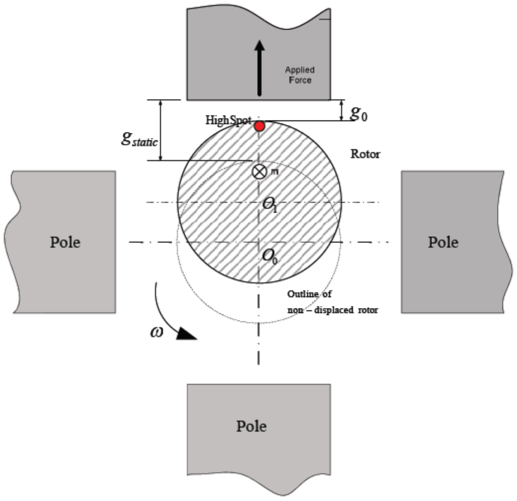


Figure 11 : Slow Rotation Rotor Configuration

Figure above shows the cross section of magnetic actuator with four poles. The rotor in the figure above shows that it displaced vertically from the center. During slow motion view, it can be seen the deflection of the rotor from its static position. Whirling orbit occur due to the unbalance force of the rotating shaft. The radius of this displaced orbit can be measured from the static position of the center of the rotor O_0 and the new position f the geometrical center of the rotor *O,* plus the radius of the rotor.

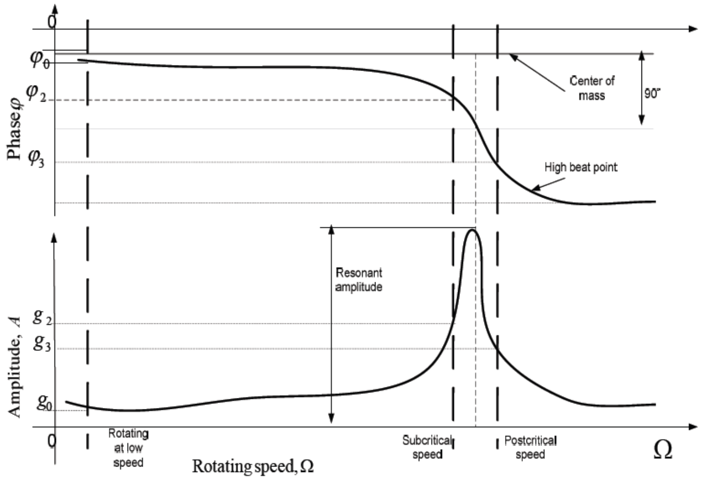


Figure 12 : Bode Plot Diagram

The Bode plot on the Figure above shows the position of the heavy spot has a fixed location while the line of response or the high spot increase as the rotating speed increase. Besides, the magnitude of the vibration of the rotor shaft increase as the resonance speed is approached. Therefore, in order to cancel the unbalance force, opposite force in the radial direction of the location of the center of mass is required. At a slow roll, this is also the location of the response (high beat point). The time base plot of the high beat point, center of mass and unbalance force at a slow roll will be in the same phase when the unbalanced force is cancelled.

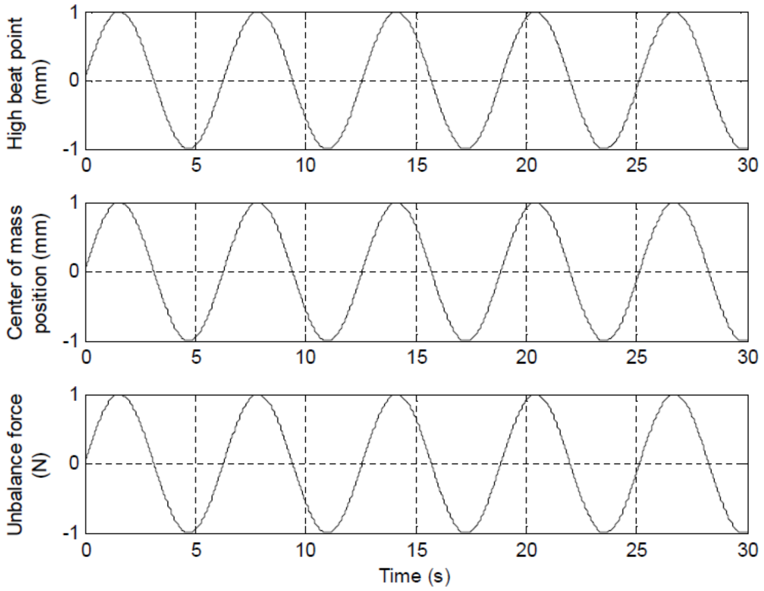


Figure 13 : Phase Diagram for Slow Rotating Rotor

In the figure above, it is clearly shown that for a slow rotational speed the high spot, center of mass and unbalance force are in the same phase or we can say that they are in the same radial line of the rotor. With increasing speed as shown on the figure below, we can see that the phase of center of mass and high spot become different in the phase angle of Φ_2 .

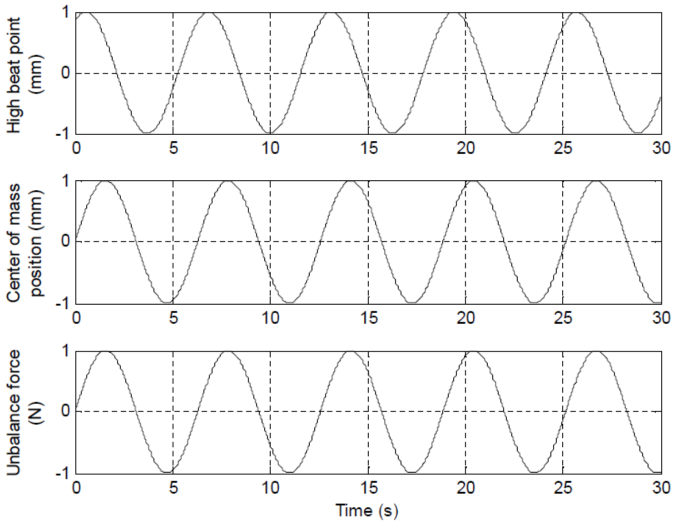


Figure 14 : Phase Diagram for Approaching Resonance Rotor Speed

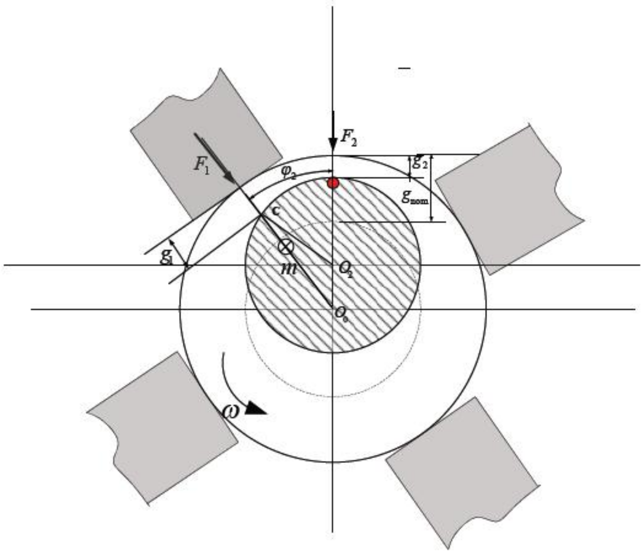


Figure 15 : Air Gap Between Stator and Rotor During Approaching Resonance Rotor Speed

In the Figure above, the rotor is at the speed close to resonance. The air gap between the stator and rotor are smaller and can be denoted by g_1 and g_2 . g_1 is the distance between the location of the center of mass and the stator while g_2 is the distance between air gap and the stator.[3]

CHAPTER 3

METHODOLOGY

3.0 Research Methodology

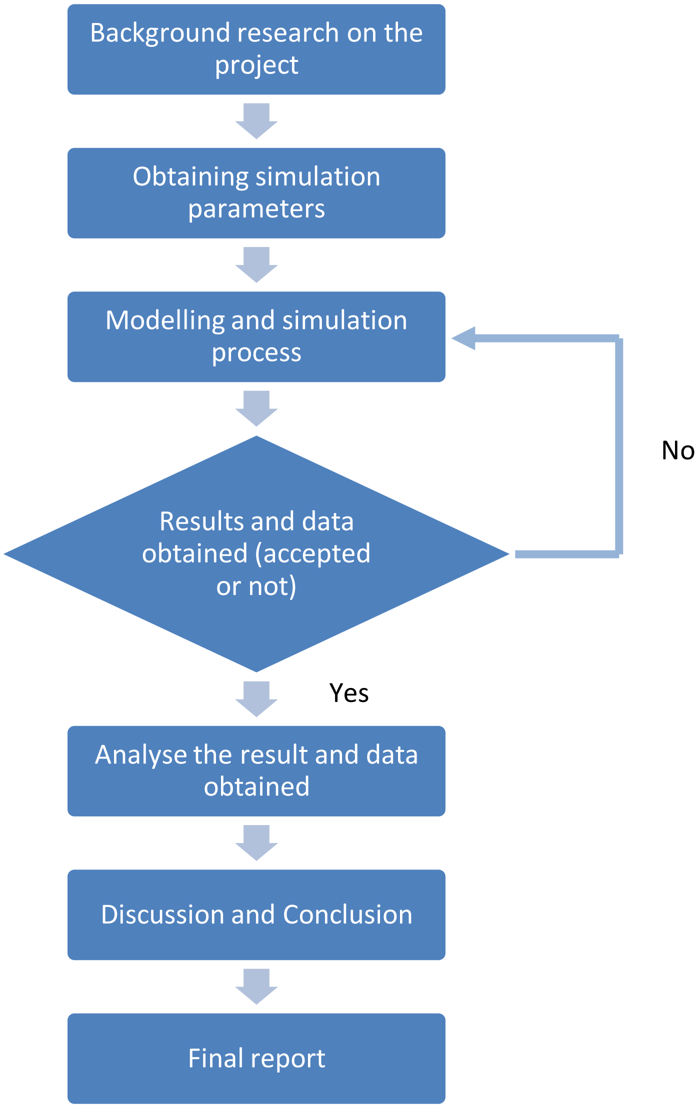


Figure 16 : Project Methodology Flow Chart

The project start with background research and study of the project. Reading materials are obtained from research paper, journal, product brochure and reference books. From the reading materials, a lot of information can be obtained and extracted to help more understanding of the project. Background operation of the magnetic exciters ha to be mastered first before this simulation and project can be conducted. Other than obtain information from reading journal, papers brochures and etc, discussion with experience rotating engineer regarding this project will be beneficial.

After the background research and study, now is the step to gather or collect data and parameters to conduct the simulation. As this simulation is only done without any test rig, it is important to obtain or run the simulation using actual test rig parameters in order to maintain the coherence of the data. So, from the reading and study from journal, papers, data and parameters to run the simulation can be obtained. Those data and parameters obtain is the actual operating test rig parameters. So what the author do is to run this simulation using the actual test rig parameters. This is the closest simulation that can be obtained in compared to the test rig.

The next step after obtaining the data and parameters is to conduct the experiment. This simulation is simulate using MATLAB and Simulink software. By using the MATLAB software, mathematical equations has to be converted into MATLAB coding to run the simulation. In this case, the author use spring –mass-damper model to simulate rotating rotor-shaft system. The MATAB coding has to be altered from the existing coding in order to suit the simulation. This Simulink software can also simulate vibration control of rotor shaft system using magnetic exciters by developing a model. All the key parameters have to be keyed in into the model to run the simulation.

After successfully running the simulation, all the results and data obtained has to be analyzed. There must be reasoning for all the data plotted result that has been obtained. Comparison of the result with and without the use of magnetic exciters has to be made. The discussion regarding the data obtained can be based on the information during the background study, but must be related to the project and data obtained during the simulation.

After finish with the result and discussion, full report has to be completed and checked by the supervisor. All the references has to be noted in the report for the final submission.

3.1 Key milestone

There are few key milestone need to be achieved in completing this project.

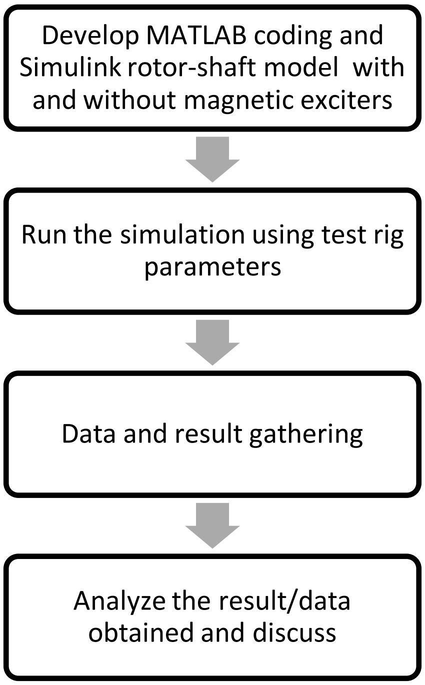
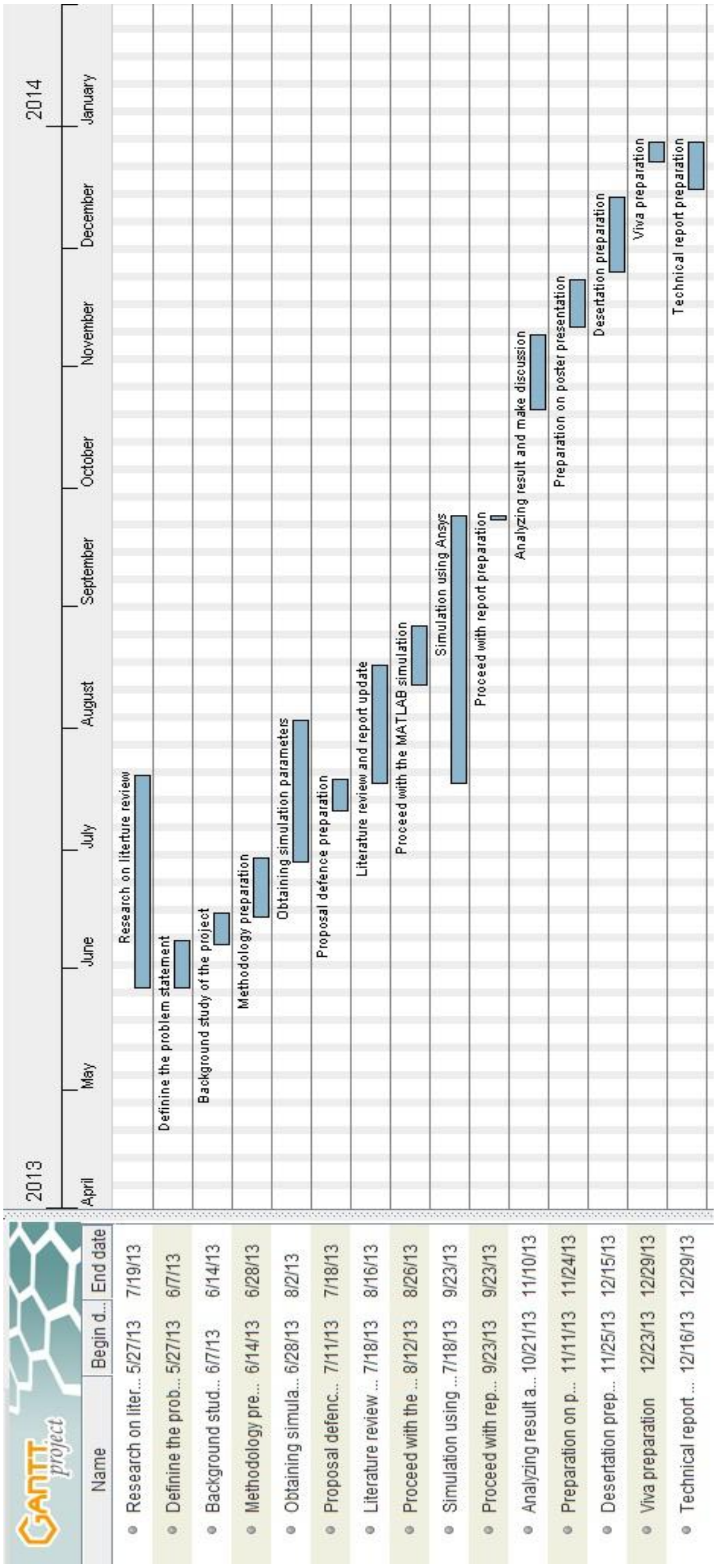


Figure 17 : Project Key Milestone Flow Chart

This project is mainly to simulate the effect of magnetic exciters on the vibration pattern of rotor-shaft system. Firstly, the model of rotor-shaft system has to be developed using MATLAB software by implementing ode45 solver which means it implements the Runge-Kutta(4,5) method. The Simulink can also be used to model the rotor-shaft system which use the spring-mass-damper model. This Simulink model is developed based on the flow diagram of spring-mass-damper itself.

The simulation is later on run using the obtained test rig parameters. All the parameters will later further discussed on the result and discussion section. Data and result is later plotted on graph. Amplitude vs time graph and bold plot graph is plotted with each represents the use and without usage of magnetic exciters. Data and result analysis will be conducted after the simulation has finished. The result is then compared with the actual test rig result. If the result shows closely similar to the test rig result, the model is accepted.

3.2 Project Gantt Chart



Name	Begin d...	End date
Research on liter...	5/27/13	7/19/13
Definine the prob...	5/27/13	6/7/13
Background stud...	6/7/13	6/14/13
Methodology pre...	6/14/13	6/28/13
Obtaining simula...	6/28/13	8/2/13
Proposal defenc...	7/11/13	7/18/13
Literature review ...	7/18/13	8/16/13
Proceed with the ...	8/12/13	8/26/13
Simulation using ...	7/18/13	9/23/13
Proceed with rep...	9/23/13	9/23/13
Analyzing result a...	10/21/13	11/10/13
Preparation on p...	11/11/13	11/24/13
Desertation prep...	11/25/13	12/15/13
Viva preparation	12/23/13	12/29/13
Technical report ...	12/16/13	12/29/13

Table 1 : Project Gantt Chart

CHAPTER 4

RESULT AND DISCUSSION

Simulation is conducted using MATLAB and Simulink to get vibration amplitude graph with the effect of magnetic actuator. The simulation is conducted based on the principle of spring mass damper. In order to reduce the vibration on rotor shaft, exciting force need to be eliminated and magnetic actuator take responsibilities.

Spring constant, k = 4000Ns/m

Damping, c = 1000Ns/m

Mass of rotor shaft system, m = 2kg

Frequency = 200 rad/sec

Unbalance force = 600N

In order to simulate spring mass damper using Simulink, first of all we have to know what is the equation involved.

$$M\ddot{x} + D\dot{x} + Kx = F\sin\omega t$$

From above equation, we can derive into this equation :

$$\ddot{x} = \frac{F\sin\omega t}{m} - \frac{c\dot{x}}{m} - \frac{kx}{m}$$

For multiplication, we use mathematical operation which in this case is the gain. 1/m gain is used to be multiplied with c and k . Then, integral is used to integrate $\ddot{x}$ (acceleration) and $\dot{x}$ (velocity). Gain k and Gain b is then used to be multiplied with x and $\dot{x}$. In order to plot the vibration pattern, we use scope and amplitude plotted graph will be shown. Input source we use sine wave to simulate exciting force. In this case magnetic actuator will be used to eliminate or reduce this exciting force.

4.0 Matlab Simulation

This MATLAB simulation use ode45 solver which means it implements the Runge-Kutta(4,5) method. Such method is suited for solving ordinary differential equations by predictions. The ode45 command is a variable step solver. This combination of 4th and 5th order of ode45 Runge-Kutta makes it very accurate.

The MATLAB code is computed using this code. First an m-file is created from the equation of motion given in first-order form :

```
function v=f (t,x)
m=2;k=4000;c=1000;Fo=10;w=200;
v=[x(2);x(1)*-k/m+x(2)*-c/m+Fo/m*sin(w*t)];
```

Then, the following command is typed on the command window :

```
>>clear all
>> xo=[0.01,1.5];
>> ts=[0 5];
>> [t,x]=ode45('f',ts,xo);
>> plot(t,x(:,1))
```

Initial Condition, velocity and amplitude

Simulation running time

Calling out ode45 function

Plot the function

This code will produce the plots given on the figure below.

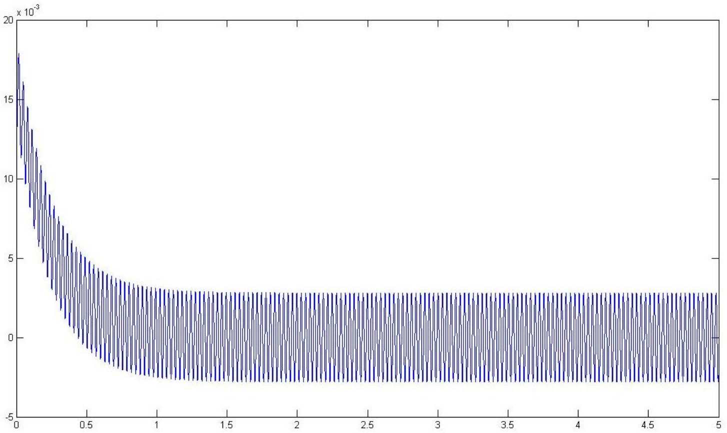


Figure 18 : Amplitude vs. Time Graph Without Magnetic Exciters

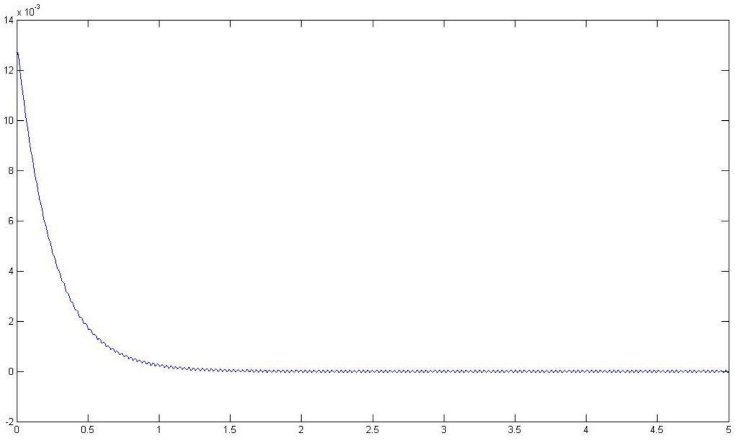


Figure 19 : Amplitude vs. Time Graph With Magnetic Exciters

Based on the response that has been plotted on the above figure, it is clearly shown that with the usage of magnetic exciters, unbalance force can be eliminated. Without the magnetic exciters, the vibration amplitude is about -3x10^{-3}mm to 4x10^{-3}mm. Although this amplitude value seems to be small, but it is significant for rotor shaft system. Even small vibration can cause catastrophic failure to the system.

With the usage of magnetic exciters, we can see that the plotted amplitude shows reduced amplitude which is about 0.5x10^{-3}mm to 1x10^{-3}mm. About 70% of the previous amplitude has been eliminated by exciters and this really shows that

The simulation is run for 5 seconds and it can be seen that initial amplitude during 0 second is higher and eventually decreasing at 1 second. This is due to the initial condition of the amplitude and rotating speed of the shaft. The motor which is the driver of the shaft is high torque and there might be slightly "over amplitude" during the initial condition.

The next MATLAB simulation is to simulate magnitude and phase of the system frequency response. Second order systems are commonly encountered in practice and is the simplest type of dynamic system to exhibit oscillations. In fact many real higher order systems are modeled as second order in order to facilitate the analysis.

The general form of the first order differential equation is as follows

$$m\ddot{y} + b\dot{y} + ky = f(t) \ \ or \ \ \ddot{y} + 2\zeta\omega_n\dot{y} + \omega_n^2 y = k_{dc}\omega_n^2\mu$$

$$G(s) = \frac{1}{ms^2 + bs + k} = \frac{k_{dc}\,\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2}$$

DC Gain

The DC gain, k_{dc} again is the ratio of the magnitude of the steady-state step response to the magnitude of the step input, and for stable systems it is the value of the transfer function when s=0. For second order systems,

$$k_{dc} = \frac{1}{k}$$

Damping Ratio

The damping ratio is a dimensionless quantity charaterizing the energy losses in the system due to such effects as viscous friction or electrical resistance. From the above definitions,

$$\zeta = \frac{b}{2\sqrt{k/m}}$$

Natural Frequency

The natural frequency is the frequency (in rad/s) that the system will oscillate at when there is no damping, ζ=0

$$\omega_n = \sqrt{k/m}$$

Coding below is used to plot the magnitude and phase of the system frequency response

```
>> k_dc = 2.5*10^-4
k_dc =2.5000e-04
>> w_n=44.72
w_n =44.7200
>> zeta=11.8
zeta =11.8000
>> s=tf('s');
>> G2 = k_dc*w_n^2/(s^2+2*zeta*w_n*s+w_n^2);
>> pzmap(G2)
>> axis([-20 1 -1 1])
>> bode(G2)
```

Define the value of k_{dc}, $\omega_n$and ζ

Define the S transfer function

Plug in the first order transfer function equation

Command to plot the Bode Plot

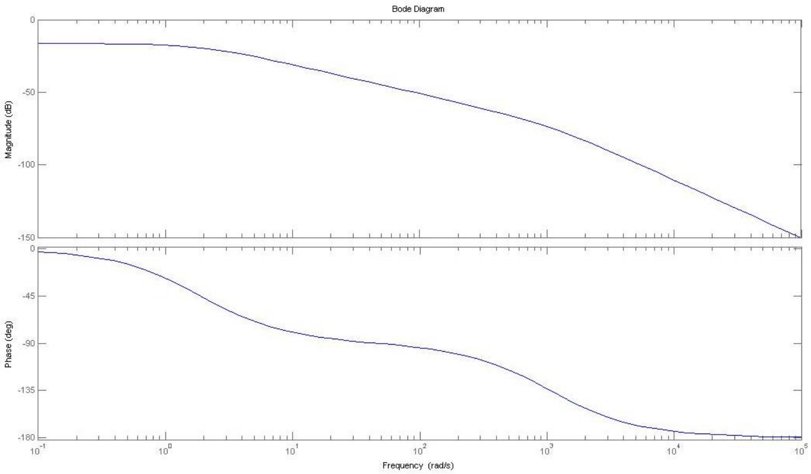


Figure 20 : Bode Plot Without Magnetic Exciters

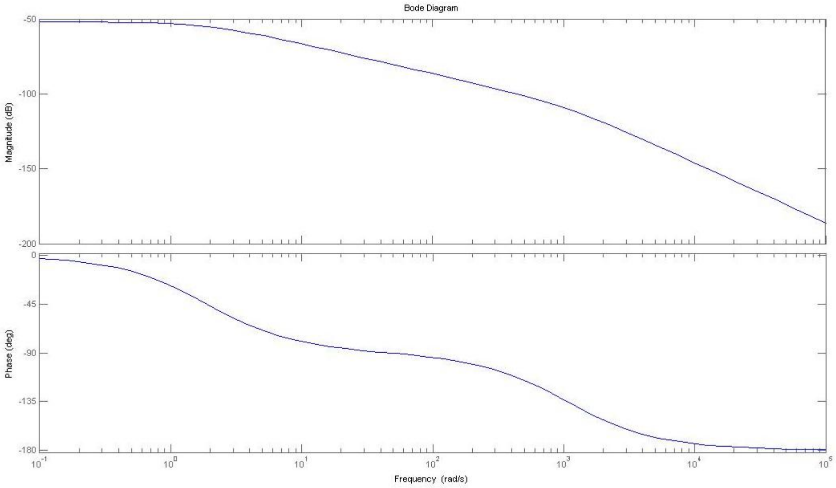


Figure 21 : Bode Plot With Magnetic Exciters

Based on the plotted graph of magnitude and phase of the system frequency response, for the system without magnetic exciters the maximum magnitude is -20db and decreases to -150db as frequency increases. Whereby, the system with magnetic exciters has lower maximum magnitude which is -50db which is clearly smaller than the system with magnetic exciters. This can be concluded that system with magnetic exciters applied has lower gain compared the system which doesn't apply magnetic exciters. The phase has no difference between system with and without magnetic exciters as both of the system rotate at the same frequency. As frequency increases, the phase change from 0degree to -180degree.

4.1 SIMULINK simulation

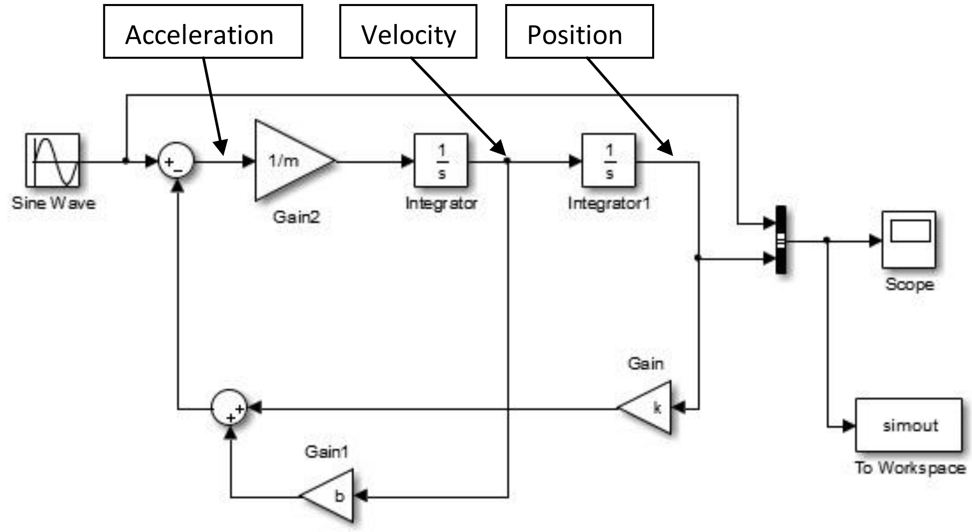


Figure 22 : Simulink Block Diagram

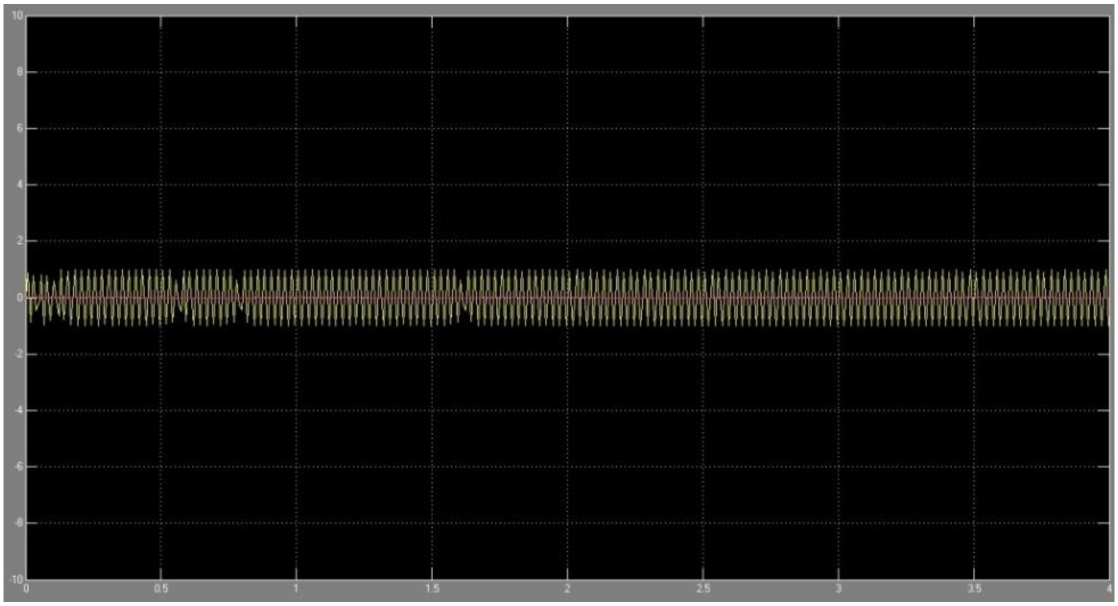


Figure 23 : Amplitude vs. Time With Magnetic Exciters

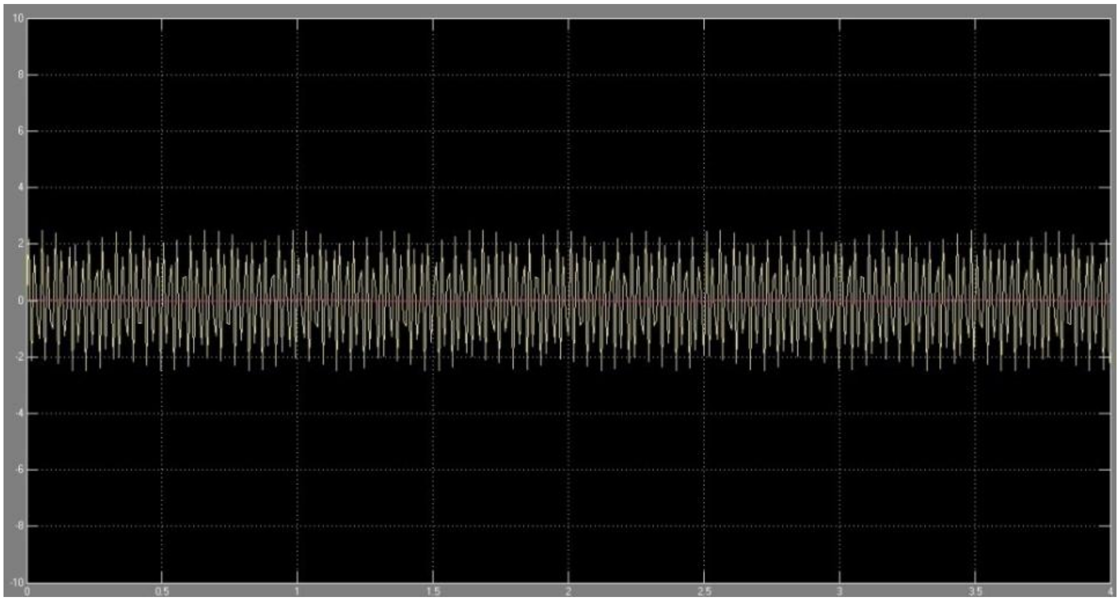


Figure 24 : Amplitude vs. Time Without Magnetic Exciters

This Simulink simulate the condition of rotating rotor-shaft system with and without magnetic exciters. Spring mass damper model is applied in this system. Sine wave is injected into the system to simulate rotating condition of the rotor-shaft. Gain and integrator is used to simulate mathematical expression and unknown which is the k, m and b. Integrator in this case is used to integrate the $\ddot{x}$ into $\dot{x}$ which will be multiplied with b which is the damping coefficient. $\dot{x}$ will then integrated to become x which will be multiplied with k which is the spring constant. k and b then multiplied with $1/m$ which will then satisfied this equation $\ddot{x} = \frac{Fsin\omega t}{m} - \frac{b\dot{x}}{m} - \frac{kx}{m}$ .The sine wave is then plotted using the scope.

Again using Simulink, it is clearly shown that with the usage of magnetic exciters, the amplitude of vibration can be reduce up to 50% to 70%. Magnetic exciters will eliminate the unbalance force that is caused by the rotating shaft. This simulation is run for 4 sec which means that the amplitude will stay as it is as long as the system is running.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.0 Conclusion

Vibration has become major problem for rotating equipment which requires critical operation. Even small unbalance force can lead to vibration and cause failure to the rotating system. Conventional bearing such as oil film and journal bearing doesn’t have the capabilities to eliminate this unbalance force and thus reduce the vibration. With the use of magnetic exciters, this unbalance force can be eliminated and lead to reduction in vibration.

The history of magnetic exciters implemented on rotor shaft system begins with the invention of active magnetic bearing. Continuous improvement on computers and software enables complicated calculation and simulation be conducted. This active magnetic bearing operates independently on magnetic bearing without the operation of conventional bearing. While magnetic exciters still operate on conventional bearing on both end with the feedback control system to control the amount of attractive forces to eliminate the unbalance force.

The parameters of simulation has been obtained from the actual test rig simulation. This is to ensure that this computer simulation can get the closest as it can with the test rig simulation to reduce any error and redundancy. The actual test rig used improvise SKF magnetic bearing with the shaft rotated on oil film bearing on both ends.

The usage of magnetic exciters to control vibration on rotor shaft really works and has been proven by result and simulation. Based on the simulation it is shown that with the usage of magnetic exciters, vibration can be reduced up to 50% to 70%. This result has clearly shown the advantage of using magnetic exciters to control rotor-shaft vibration.

5.1 Recommendation

For future work, improvement on the Matlab coding can be made to reduce redundancy on the result plotted. Coding for magnetic exciters can be developed to represent magnetic exciters and sync with the spring mass damper rotating shaft coding. Current coding is variably adjusted to the amount of unbalanced force to simulate the vibration pattern with and without the use of magnetic exciters.

Besides, the still have improvement on the result that is was obtained. For future research, test rig simulation can be conducted. With test rig simulation, actual experimental result on the vibration pattern can be compared with the computer simulation result. Using Matlab is only theoretical result, whereas in real situation, error is one of the major factor that has to be considered.

Therefore, I would like to suggest to UTP especially Mechanical Department to equip their laboratory with the active magnetic actuator test rig. This active magnetic actuator technology is not quite new, and have been used widely in the industries. With this advance technology test rig available, it will give opportunity to students to experience hands on experience in handling active electromagnetic actuator to control rotor shaft vibration.

REFERENCES

1. Mykhaylyshyn, V. (2011). Application of Active Magnetic Force Actuator For Control of Flexible Rotor System Vibrations, Cleveland State University.
2. A.S. Das, M.C Nighil, J.K. Dutt, H. Irretier. (2007) Vibration Control and Stability Analysis of Rotor-Shaft System With Electromagnetic Exciters, University of Kasel Germany. Retrieved October 11 2006.
3. Ahmed, A.H, Wahab, T.M.A. (2009). Active Magnetic Bearing Design Optimisation and Transient State Analysis Using ANSYS, University of Technology Baghdad.
4. SCHWEITZER, G & TRAXLER, A. Design of Magnetic Bearings, International Symposium on Design and Synthesis, The Japanese Society of Precision Engineers, Tokyo, July 1984.
5. BRADFLELD, C D, ROBERTS J B & KARUNENDIRAN, P. Performance of an electromagnetic bearing for the vibration control of a supercritical shaft. Proc. Instn. Mech. Engrs., Vol. 201, NoC3, I MechE, 1987.
6. BURROWS, C R & SAHINKAYA, M N. Vibration control of multi-mode rotor-bearing systems, Proc. Royal Soc, Lon. A386, 1983, p77–94.
7. Blanco, A.; Silva, G. and Gómez, J. C. (2003). Dynamic Stiffness Control and Acceleration
8. Scheduling for the Active Balancing Control of a Jeffcott-Like Rotor System, Proceedings of The tenth International Congress on Sound and Vibration, pp. 227-234, Stockholm, Sweden, July 7-10, 2003.
9. Beltrán, F.; Sira, H. and Silva, G. (2006). Adaptive-like Active Vibration Suppression for a Nonlinear Mechanical System Using On-Line Algebraic Identification, *Proceedings*. *Of the Thirteenth International Congress on Sound and Vibration*, Vienna, Austria, July 2-6, 2006.
10. Beltrán F.; Silva, G.; Sira, H. and Blanco, A. (2010). Active Vibration Control Using On-line Algebraic Identification and Sliding Modes. Computación y Sistemas. Vol. 13, No. 3, pp 313-330. ISSN 1405-5546.
11. Blanco, A.; Silva, G. and Gómez, J. C. (2003). Dynamic Stiffness Control and Acceleration Scheduling for the Active Balancing Control of a Jeffcott-Like Rotor System, *Proceedings of The tenth International Congress on Sound and Vibration*, pp. 227-234, Stockholm, Sweden, July 7-10, 2003.
12. Blanco, A.; Beltrán, F.; Silva, G. and Méndez, H. (2010). Control de Vibraciones en Sistemas Rotatorios, Revista Iberoamericana de Automática e Informática Industrial. Vol. 7, No. 4, (Octubre 2010.) pp. 36-43, ISSN 1697-7912.

13. MOORE J J, DAVID L R. Centrifugal compressor stability prediction using a new physics based approach [C/CD]//*Proceedings of ASME Turbo Expo 2009: Power for Land, Seal and Air*, Orlando, Florida, USA, GT200960229.
14. MANOJ K G, THOMAS A S, DARA W C. New steps to improve rotor dynamic stability predictions of centrifugal compressor [C/CD]//*Proceedings of ASME Turbo Expo 2007:Power for Land, Seal and Air*, Montreal, Canada, GT200727501.
15. BURROWS C R, SAINKAYA M N, TUTKAY O S. An adaptive squeeze film bearing[J]. *Journal of Tribology*, 1984, 106(1): 145–151.
16. ZHU Changsheng. Magnetic field controlling squeeze film damper for rotor system[J]. J*ournal of Zhejiang University (Engineering Science)*, 2007, 41(9): 1 561–1 566.
17. DESMIDT H A. Imbalance vibration suppression of a supercritical shaft via an automatic balancing device[J]. *Journal of Vibration and Acoustics*, 2009, 131(4): 1–12.
18. LUCAS Ginzinger, HEINZ Ulbrich. Efficient and robust rub control with an active auxiliary bearing[C/CD]//*Proceeding of ASME Turbo Expo 2007:Power for land, Sea and Air*, May 14–17, 2007, Montreal, Canada, GT200728229.
19. YAN Skladanek, JOHAN Der Hagopian, JARIR Mahfoud. Energy cost assessment of the active control of rotating machinery by using an electromagnetic actuator and a piezoelectric actuator[C/CD] *Proceeding of ASME Turbo Expo 2009: Power for Land, Sea and Air*, June 8–12, 2009, Orlando, Florida, USA, GT200959625.
20. AENIS M, KNOPF E, NORDMANN R. Active magnetic bearing for the identification and fault diagnosis in turbomachinery[J]. *Mechatronics*, 2002, 12: 1011–1021.
21. KASARDA M E F, MENDOZA H, KIRK R G, et al. Reduction of subsynchronous vibration in a singledisk rotor using an active magnetic damper[J]. *Mechanics Research Communications*, 2004, 31: 689–695.
22. MANI G, QUINN D D, KASARDA M. Active health monitoring in a rotating cracked shaft using active magnetic bearings as force actuators [J]. *Journal of Sound and Vibration*, 2006, 294: 454–465.
23. Muszynska, A. Whirl and Whip – Rotor / Bearing Stability Problems. Journal of Sound and Vibration (1986) 110(3), pp 443-462.
24. Šimek, J.: Test stand for active control of sliding bearings with the aim to suppress rotor instability, Technical report No. 07-405. TECHLAB Ltd., 2007 (in Czech)

25. Svoboda, R.: Theoretical analysis of influence of bearing bushing external excitation on dynamic properties of the rotor, Technical report No. 07-301. TECHLAB Ltd., 2007 (in Czech)
26. Rho, B-H, & Kim, K-W. (2002) The effect of active control on stability characteristics of hydrodynamic journal bearings with an axial groove. *Proceedings of the Institution of Me-chanical Engineers,* Part C: Journal of Mechanical Engineering Science, Volume 1. 216, Number 9 / 2002, 2002, pp. 939-946.