DYNAMICS OF ROTOR SUPPORTED BY MAGNETIC BEARING

by

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14780

Dissertation submitted in partial fulfilment of
the requirements for the
Degree of Study (Hons)
(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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Approved by,

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UNIVERSITI TEKNOLOGI PETRONAS

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January 2015

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

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Mohd Safwan Bin Mat Din

ABSTRACT

This paper presents a study of the dynamic behavior of a rotor system supported by magnetic bearing. Specifically, the work investigate the range of realistic operating parameters that evolve around the rotation pattern of magnetically supported rotor, the vibration that occurs during the operation and the results difference after adjustments of some input parameters. The simulations of rotating rotor with magnetic bearing supports will be executed using ANSYS software. By varying the damping ratio and stiffness coefficient, the simulation shall produce different results. With the simulation, the research can identify the limit to instability by varying the damping of bearing during operation, thus, suggest recommendations on improvement of stability of magnetic supported rotor system.

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CHAPTER 1

INTRODUCTION

1.1 Background Study

The blooming technology of rotating equipments whether in oil and gas industry or aeronautic industry, the number of equipments are keep increasing. Thus, the limit of current design continually pushed forward to meet new demands of the users. Concerning the performance of the equipment, a way to boost the performance of the system is by using magnetic bearings.

Magnetic bearing is used in developing smart machine, with its capability of controlling the stiffness of the bearing pedestals and suport structures. An ability or service for automatic diagnosis of faults united with capability to apply corrective loads of a machine is known as smart machine.

The magnetic bearings is designed with the expectation to overcome the common problem of conventional bearing. The current situation of the conventional bearing is that, they have significant weaknesses such as vibration, heat loss due to friction, the need for lubrication and short service intervals. These properties do affecting the drop of efficiency, costly maintenance and decrease in production.

Magnetic bearing can be divided into two types which are active magnetic bearing (AMB) and passive magnetic bearing (PMB). Active magnetic bearing is electrically controlled, while passive magnetic bearing has no control mechanism. Same with conventional bearing, magnetic bearing's function is to support the rotating shaft, but magnetic bearing support it without any physical contact by levitate the rotor in the air. The magnet can be electrically controlled magnetic force or permanent magnet. Fundamentally, magnetic bearing consist of an electromagnetic assembly, power amplifiers that will supply current to the electromagnets, position sensors, and a controller.

Auxiliary bearing will be match together with magnetic bearing. Auxiliary bearing is used to support the rotor during non-operational period and to protect the rotating assembly from being damaged during unstable rotation or loss of power during operation.

1.2 Problem Statement

Magnetic bearing is designed to overcome the common problem faced by conventional bearing. This research is carried out to study the dynamics of magnetic supported rotor. Thus, finding the ideal operating parameters of the magnetically supported rotor through simulation on ANSYS.

1.3 Objectives

Throughout this project, a few objectives are set to achieve.

I. To identify the limit to instability by varying the damping ratio during operation.

II. To suggest recommendations on improvement of stability of magnetic supported rotor system.

1.4 Scope Of Study

Throughout this study, author will be expose on the theory of two degree of freedom which is a forced vibration involving damping and stiffness of the bearing and smart rotor mechanism. The author need to learn and expert himself on ANSYS software in order to carry out simulation and analysis of the study.

CHAPTER 2

LITERATURE REVIEW

2.1 Smart Rotating Machines Trough The Use Of Active Bearing

Smart rotating machine design must have their basics application involved, which are the integration of controllable bearings, actuators, and sensors to measure a defined set of physical variables, an on-line adaptive controller, a control algorithm to attain the desired performance and an algorithm to reconfigure the system in the event of faults.

Compare to other bearings, the magnetic bearing does have lower load capacity. However, the magnetic bearings offers the potential for system diagnosis in addition to their primary function. Proportional-integral-derivative (PID) feedback control is used in active device like AMBs. With the ability to adapt or adjust control action to changes in internal structure or in the operational environment that make the device as smart machine.

Figure 2.1 show that magnetic bearing satisfied all the requirement.

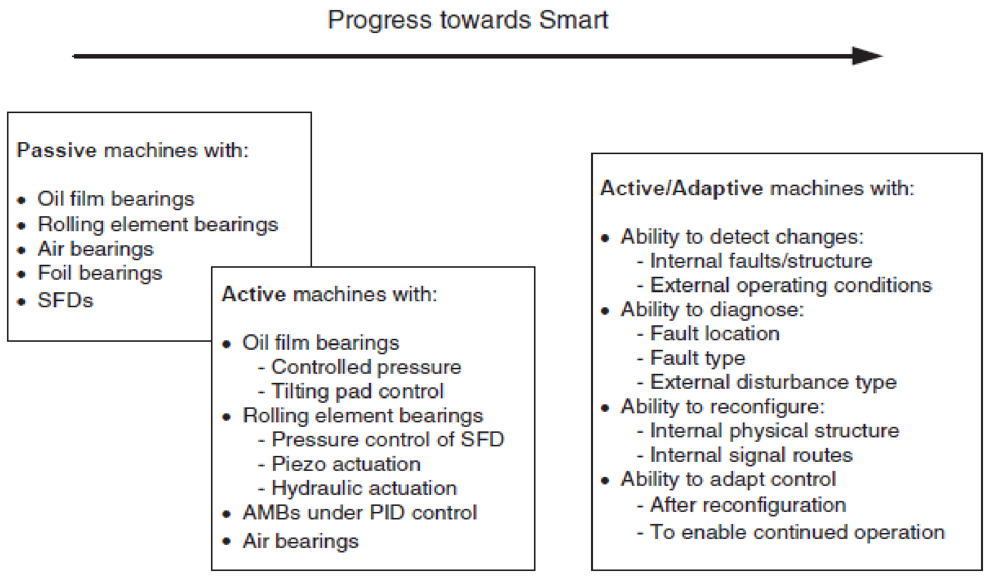

Figure 2.1: Development to enable smart machine performance and operation (Burrows et al, 2009)

2.2 Operation Mechanism Of Magnetic Bearings

The interesting advantage of magnetic bearing over the other conventional bearing is the bearings levitate the rotors, thus, there are no physical contact with rotors (Setiawan, 2006). Just like normal mechanical bearing, the magnetic bearing works are same apart from the bearing task is to exploit magnetic field and levitate the rotor in magnetic field (Mukhopadhyay et al, 2004).

The authors also stated that magnetic bearing process usually using the repulsive forces between the rotor permanent magnet and stator to levitate the rotor. However, in nature, the system is unstable. To keep the rotor in the required position, the controlled electromagnet is used. Most of magnetic bearing application use Active Magnetic Bearing (AMB) where the device use solely on electromagnets.Thus, this study will be focus on the mechanism of electromagnet.

2.2.1 Active Magnetic Bearing (Amb)

Based on Iannello (2009), an active magnetic bearing composed of a stator and a rotor. A stator accommodate the electromagnets and the position sensors, while the rotor, which rotates with the shaft. The rotor will be placed centered in correlate with stator in a operating bearing, so that no contact occurs. Closed-loop feedback system is used to controlled the position of the shaft. The displacement between shaft and bearing will be detected by position sensors where it will send these signal to a digital controller. These signal is processed by the controller then calculate the required currents in the electromagnets to maintain the centered position of the shaft. The current in the electromagnets will be alter according the calculation by power amplifiers. This sequence is repeated almost 15,000 times per second. Same as other types of bearings, the magnetic bearing come up with stiffness and damping except the stiffness and damping vary as a function of disturbance frequency. The bearing can be said as a transfer function which consist of amplitude and phase that differ with frequency. The transfer function is crucial to ensure the performance of the bearing has sufficient stability and force rejection over a range of frequency. The transfer function does have the power in controlling the stiffness and damping of the bearing.

2.3 Advantage Of Magnetic Bearing

Rama (2012), described magnetic bearing as a non-contacting technology, where the magnetic bearings have zero friction, no wear and high reliability. The speed that was unachievable by other bearings can be achieve by magnetic bearing. Iannello (2009), also said that magnetic bearing does have more benefits compare to other types of bearing in rotating machinery such as higher reliability with low maintenance, reduced frictional losses, no contaminating or flammable lubricants, reduced machine vibration, improved health monitoring and diagnostics.

Based on several study and journal, the advantages of magnetic bearing can be divided into 4 main aspects:

1) Maintenance

   Maintenance is action taken to preserve the condition of the equipment to be as original as it was. In maintenance, it is costly in term of part require in maintenance of an equipment, especially the lubrication. However, for magnetic bearing, the absenteeism of lubrication does get rid of mechanical maintenance, thus making it fit for vacuum and clean room operation (Setiawan, 2006).

2) Speed and Load

Table 2.1: Top Speed Comparison (X. Xie ,2004)

Bearings	Approximately maximum speed (mm*RPM)
Steel bearings	1,000,000
Ceramic bearings	2,000,000
Hydrostatic/Hydrodynamic bearings	1,000,000
Air bearings	4,400,000
Magnetic bearings	4,500,000

X. Xie (2004) stated that there are no limitation of speed of the component that was support by magnetic bearing. With having the properties of frictionless and programmable features, AMB can meet the higher demand for higher speed operation (Setiawan, 2006). X.Xie (2004) also stated that, practically, magnetic bearing can support any magnetic load, relying on the space and money that one wish to supply. Current that pass through the coils will be reduce and resultant heat generated when expand the amount of the load that is supported by permanent magnet.

Rolling element is superior than magnetic bearings and foil bearing in this case, magnetic bearing load capacities has increased further off assumption. The load capacitive were likely to increase with the advances of high strength magnetic materials (Clark et al, 2004). Based on Clark et al, (2004) also, the magnetic bearing does have higher operating speed compare to roller bearing and foil bearing.

Table 2.2: Comparison of top speed using various bearings (Clark et al, 2004)

Type of Bearing	Roller Bearing	Foil Bearing	Magnetic Bearing
Documented Operating Speed (RPM)	Less than 2 million	2 million	2.25 million

3) Frictionless

Setiawan (2006), stress that, compare to conventional bearing, friction is remarkably removed in magnetic bearing. This can grant the efficiency in energy and also having longer life. Iannello (2009), also agreed that magnetic bearing reduced frictional losses.

4) Operating temperature.

Operating temperature of the bearing must be considered, since heat produced by eddy current losses in the stator core and resistance heating in the electromagnetic coils. However, the past few years progress show that high strength ferromagnetic materials, like Hyperco 50HS, and heat treatment process has been introduced to reduced the temperature (Clark et al, 2004).

2.4 Backup System

Industrial turbomachinery equipments' magnetic bearing system usually be composed of two radial bearings, one axial or thrust bearing, position sensors, a control system and auxiliary system that include a landing system, and a purge system (Sears and Uptigrove, 2013). The function of the auxiliary landing system is to support the rotor when the system is delevitated. Auxiliary bearing's main purpose is to avoid any contact between rotor and stator (Cade et al, 2008). He also added that auxiliary bearing controller also used to reduce rotor and bearing contact force and maintain or restore the rotor to the non contacting position in running operation. The configuration of the both bearing on one shaft can be refer on figure 2.2.

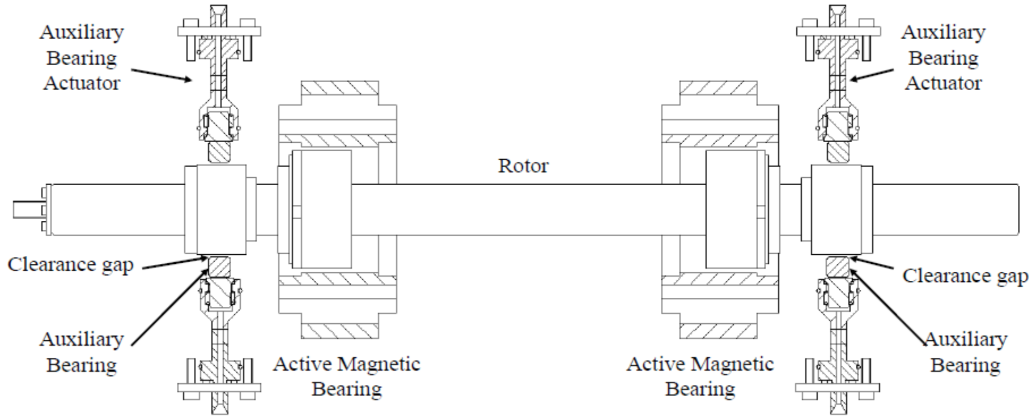

Figure 2.2: Auxiliary Bearing System schematic diagram (Cade et al, 2008)

2.5 Control System

Weise and Pinckney (1989) explain that active magnetic bearing use a closed loop operation. In processing the position signal and power the appropriate bearing coils, control electronics are necessary. Position sensors will determined the exact shaft location, and a DC voltage is produced depend on the rotor displacement. An error signal will be generated if there are any difference between the current shaft position and centered position (where the shaft should be) and this signal is used to maintain control of the rotor. This signal will be amplified, filtered, and conditioned, in order to command the specified power amplifier. Based on the position of the shaft, the current in the suitable bearing coils will be increased or decreased to maintain equilibrium position of rotor. Figure 2.3 show the basic block diagram of the closed loop, servo control.

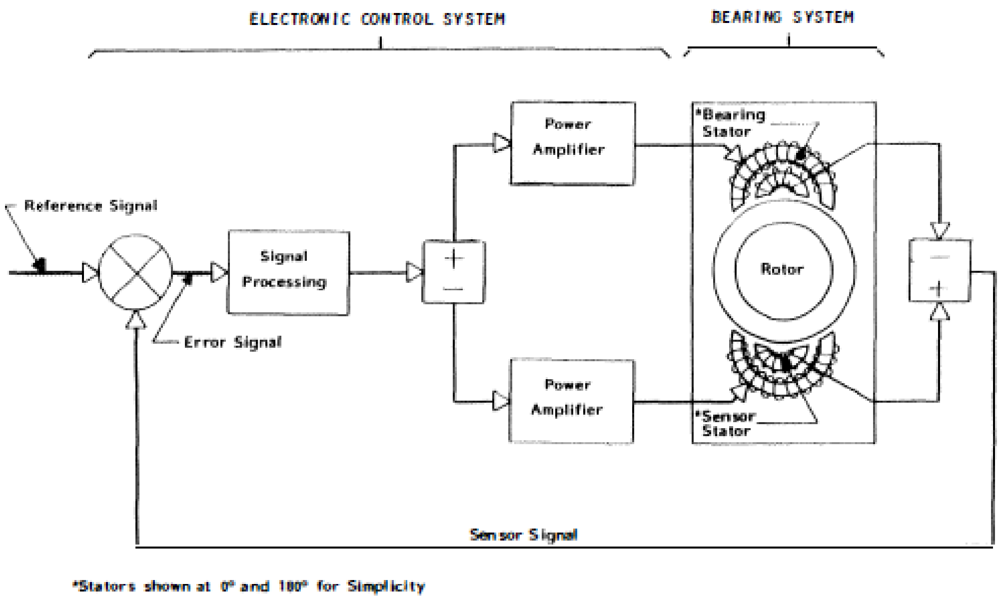

Figure 2.3: Basic Control Loop Diagram (Weise and Pinckney, 1989)

2.6 Campbell Diagram In Rotor-Bearing System Analysis

In understanding the dynamic behaviour of the rotating machines, the Campbell diagram is one of the most important tools. It basically consists of a plot of the natural frequencies of the system as functions of the spin speed. Eventhough based on complete linearity, the Campbell diagram of the linearized model can yield many important information concerning a nonlinear rotating system (Dumitru et al, 2009).

Aubrey et al (2000) stated that there are two type of waves in campbell diagram, which are, forward waves and backwards wave. Forward waves is produced when the rotation speed is added to the propagation speed of the travelling wave. Backward waves is formed when subtracted from the propagation speed of the travelling wave in the opposite direction. This leads to a splitting of each mode shape into two different frequencies. Only mode shapes with certain nodal diameters are affected.

Therefore, the frequencies of the forward modes increase and the frequencies of the backward modes decrease in relation to the rotation speed.

$$f_f = f_s + i\frac{\Omega}{60} \qquad f_b = f_s - i\frac{\Omega}{60} \qquad {f_s}^2 = f^2 + K(\frac{\Omega}{60})^2$$

where f_f and f_b respectively are the frequencies associated with the forward and the backward waves; i is the number of nodal diameter; Ω is the speed in rpm; K is dimensionless value and f is the natural frequency when the rotor is not rotating.

One approach for determining critical speeds is to generate the whirl speed map, include all excitation frequency lines of interest, and graphically note the intersections to obtain the critical speeds associated with each excitation. It is common practice to plots both whirl speed maps, i.e., the whirl frequencies and damping ratio ξ_i (or in terms of the logarithmic decrement δ_i) versus the rotor spin speed Ω. In order to obtain a correct plotting of the Campbell diagrams, in case these diagrams are drown for a finite number of rotations, we should achieve a correspondence between the modal forms and the eigenvalues.

$$\xi_i = -\frac{\alpha_i}{\sqrt{\alpha_i^2+\omega_i^2}} \cong -\frac{\alpha_i}{\omega_i}$$

$$\delta_i = 2\pi\xi_i$$

Noted that, α_i is the damping constant and ω_i is the damped natural frequency of whirl speed.

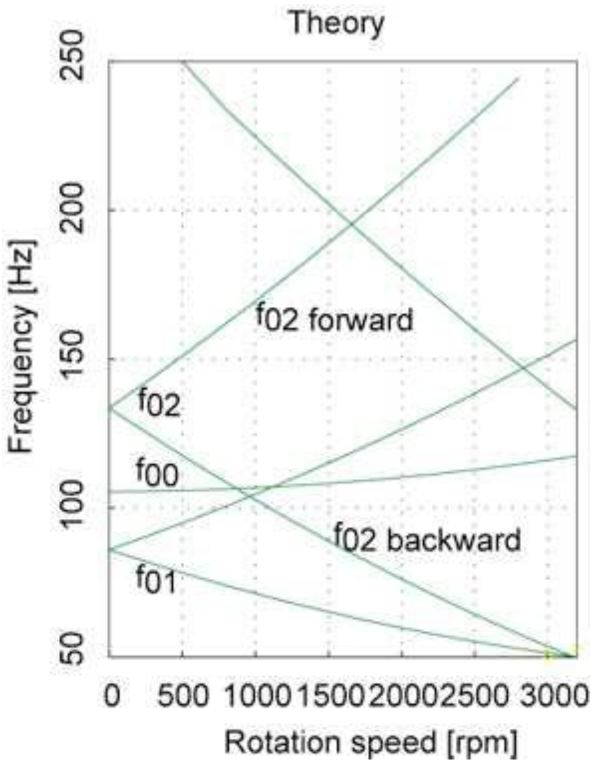

Figure 2.4: Example of Campbell Diagram (Aubrey et al, 2000)

2.6 Three Disk Shaft Analysis

Dumitru et al (2009) did his analysis on a three disk shaft with ball bearing was install at both ends of it. The analysis was done by plotting the campbell diagram of the rotor itself. Table 2.3 show the properties of the rotor. Based on figure 2.5, the campbell diagram there are several mode was forward and backward. However, there are not stated there whether the rotor was stable or not, and if any critical speed detected in the analysis.

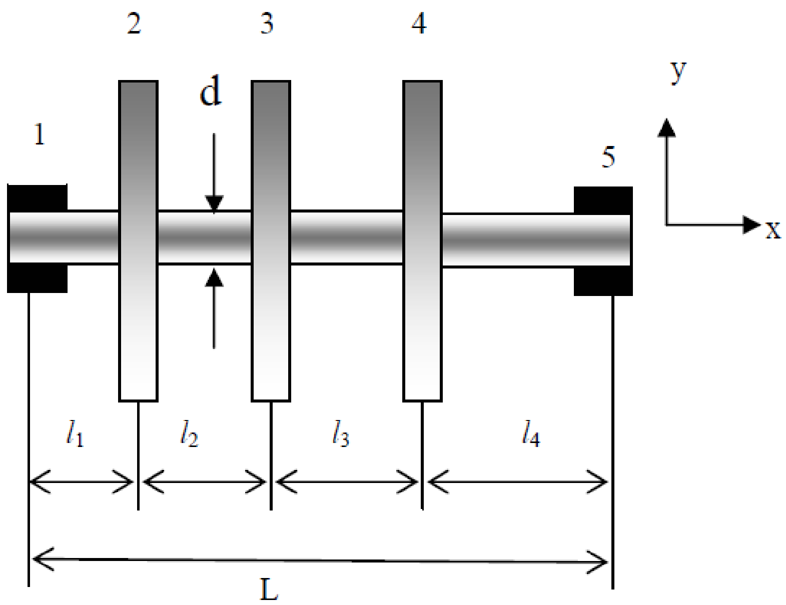

Figure 2.5: Rotor configuration (Dumitru et al, 2009)

Table 2.3: Rotor data (Dumitru et al, 2009)

Shaft	Disk	Bearings
$l_1 = 0.2$ m, $l_2 = 0.3$ m	$m_2 = 14.58$ Kg, $m_3 = 45.94$ Kg, $m_4 = 55.13$ Kg	Station 1 $k_{yy} = 7e7$ N/m $k_{zz} = 5e7$ N/m
$l_3 = 0.5$ m, $l_4 = 0.3$ m	$J_{T2} = 0.064$ Kg m^2 $J_{T3} = 0.498$ Kg m^2 $J_{T4} = 0.602$ Kg m^2	$c_{yy} = 7000$ Ns/m $c_{zz} = 4000$ Ns/m Station 5
d = 0.1 m ρ=7800 Kg/m^3 E=2e11 N/m^2	$J_{P2} = 0.123$ Kg m^2 $J_{P3} = 0.976$ Kg m^2 $J_{P4} = 1.171$ Kg m^2	$k_{yy} = 6e7$ N/m $k_{zz} = 4e7$ N/m $c_{yy} = 6000$ Ns/m $c_{zz} = 5000$ Ns/m

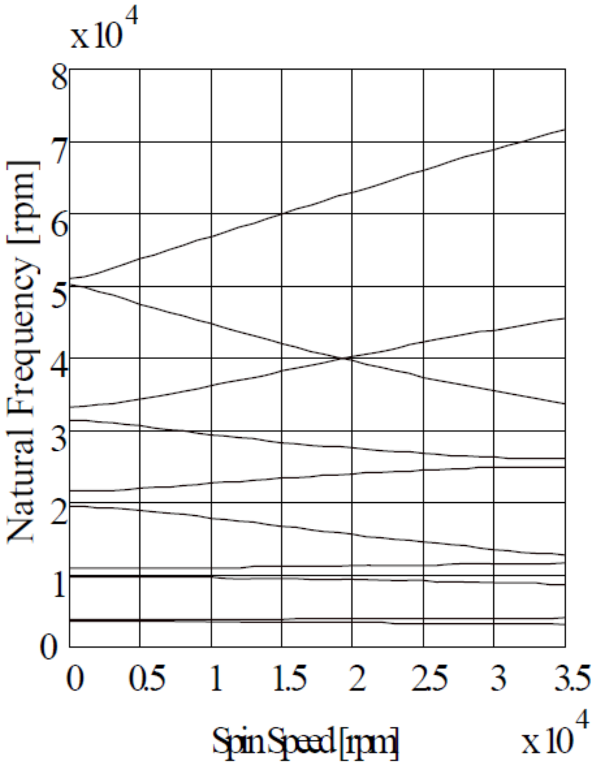

Figure 2.6: Campbell diagram (Dumitru et al, 2009)

2.7 Vibration Damping for Modal Analysis

Vibration characteristics which are the natural frequencies and mode shapes of a vibration system can be determined by using modal analysis (Cai et al, 2002). Modal analysis result can be futher applied for another dynamic analysis through mode superposition method. In modal analysis, Alpha (mass) damping, Beta (stiffness) damping, Material-dependent damping ratio (input on MP, DAMP command) and element damping (applied via element real constant) can be specified. Modal damping equation are:

$$\xi_r = \left(\frac{\alpha}{2\omega_r}\right) + \left(\frac{\beta}{2}\omega_r\right) + \xi + \xi_{mr}$$

where α is uniform mass damping multiplier (input on ALPHAD command); β uniform stiffness damping multiplier (input on BETAD command); ξ is constant damping ratio (input on DMPRAT command) and ξ_{mr} is the modal damping ratio (input on MDAMP command).

CHAPTER 3

METHODOLOGY

3.1 Project Flow

This is the project flow that was planned by the author in order to achieve the objective.

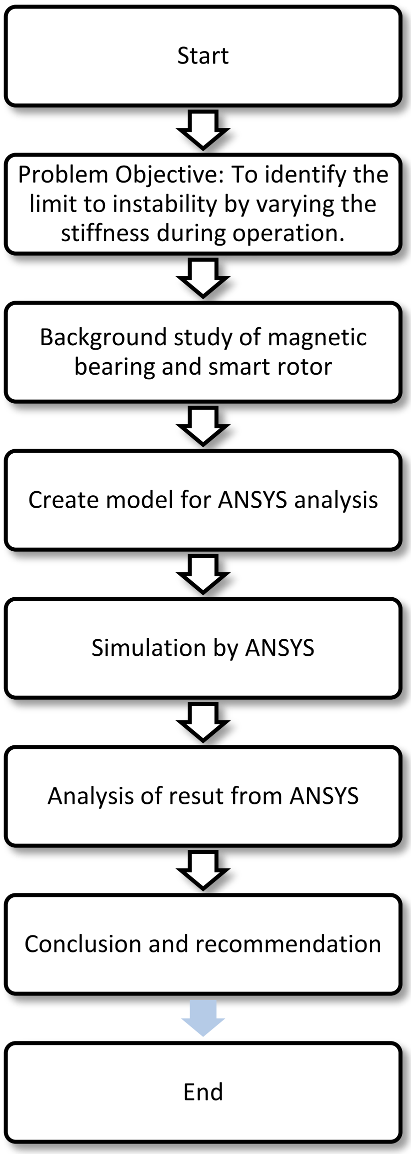

Figure 3.1: Overall Project Flow

3.2 Rotor-Bearing System Analysis

The shaft design was with bearing were installed at both ends of the shaft, with the three disk at the middle part of the shaft. The geometry datum was based on the study done by Dumitru et al (2009).

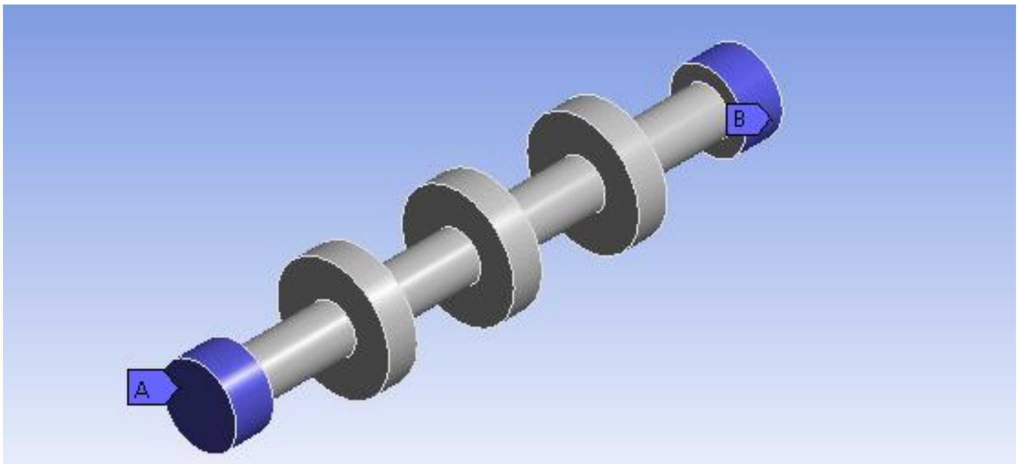

Figure 3.2: Shaft with support at both ends

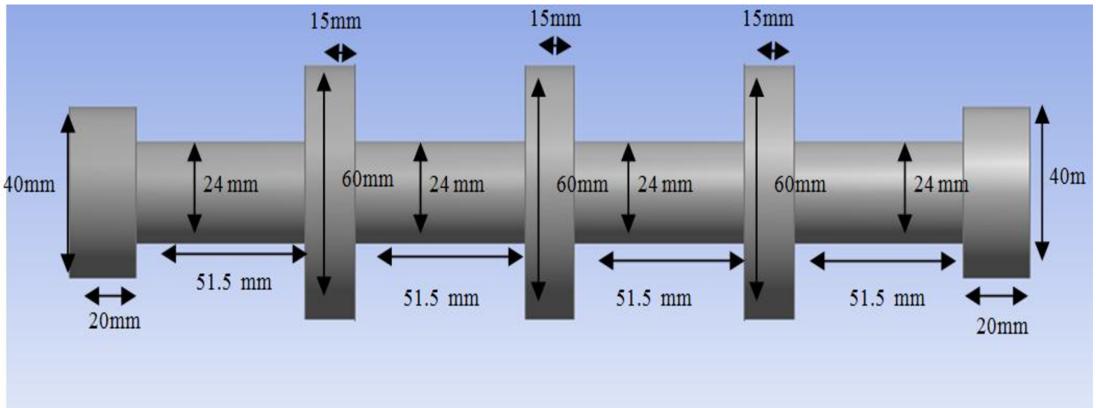

Figure 3.3: Side view of the shaft with dimensions

One approach for determining critical speeds is to generate the whirl speed map, include all excitation frequency lines of interest, and graphically note the intersections to obtain the critical speeds associated with each excitation. It is common practice to plots both whirl speed maps, i.e., the whirl frequencies and damping ratio ξ_i (or in terms of the logarithmic decrement δ_i) versus the rotor spin speed Ω. In order to obtain a correct plotting of the Campbell diagrams, in case these diagrams are drown for a finite number of rotations, we should achieve a correspondence between the modal forms and the eigenvalues.

$$\xi_i = -\frac{\alpha_i}{\sqrt{\alpha_i^2+\omega_i^2}} \cong -\frac{\alpha_i}{\omega_i} \tag{1}$$

$$\delta_i = 2\pi\xi_i \tag{2}$$

Noted that, α_i is the damping constant and ω_i is the damped natural frequency of whirl speed. The logarithmic decreament can be analyze through the ANSYS simulation and shown in the simulation result table with the changes in damping ratio.

Campbell Diagram Plotting:

Based on Aubrey et al (2000), the equation in plotting the campbell diagram will be:

$$f_f = f_s + i\frac{\Omega}{60} \quad (3)$$

$$f_b = f_s - i\frac{\Omega}{60} \quad (4)$$

Where f_f and f_b respectively are the frequencies associated with the forward and the backward waves; i is the number of nodal diameter; f_s is constant natural frequency when the rotor is not moving; Ω is the speed in rpm; K is dimensionless value and f is the natural frequency when the rotor is not rotating.

3.3 Simulation Using Ansys Software

ANSYS Workbench will be used in this project for simulation.

There are five simulations will be done to study the effect of changes in damping ratio and stiffness coefficient toward stability of the rotor.

List of Simulation:

1. Simulation of ball bearing on three disk shaft
   a) Damping ratio: 0.3
   b) Stiffness coefficient: 9.5493e-005
2. Simulation of active magnetic bearing on three disk shaft
   a) Damping ratio: 0.3
   b) Stiffness coefficient: 9.5493e-005
3. Simulation of active magnetic bearing on three disk shaft
   a) Damping ratio: 0.5
   b) Stiffness coefficient: 1.5915e-004
4. Simulation of active magnetic bearing on three disk shaft
   a) Damping ratio: 0.01
   b) Stiffness coefficient: 3.1831e-006
5. Simulation of active magnetic bearing on three disk shaft
   a) Damping ratio: 0.001
   b) Stiffness coefficient: 3.1831e-007

Basic Procedure of ANSYS Software:

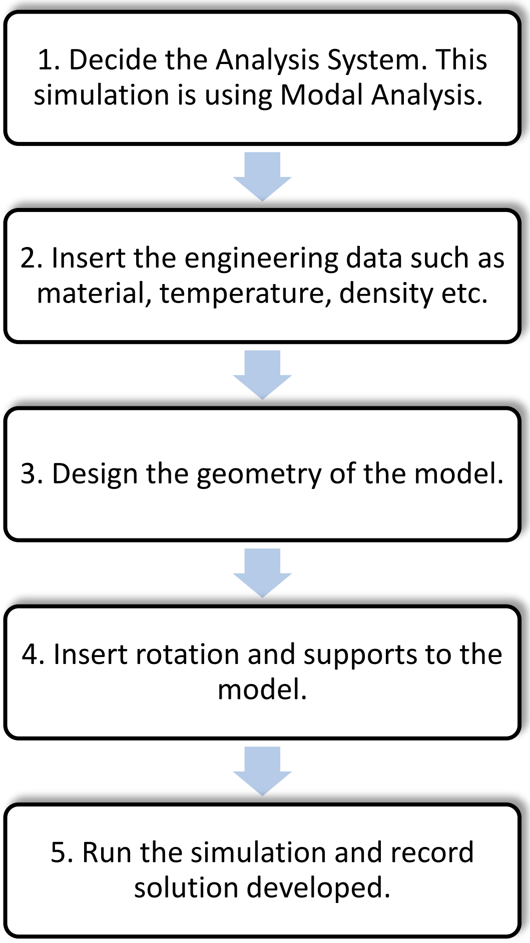

Figure 3.4: Basic Procedure Of ANSYS Software

ANSYS workbench was used for the simulation.

Steps for ANSYS simulation:

1) Decide the analysis system Modal Analysis were used for the simulation.

The Modal Analysis is the set of prescribed logic and mathematical operation in ANSYS that is designed to satisfy the user's interest. Modal analysis includes the study of vibration and stability of the model.

2) Engineering data of the shaft is set.

The engineering data of the shaft are the materials, density, temperature and other properties. For this simulation the material chosen is structural steel with density of 7850 kg/m^3.

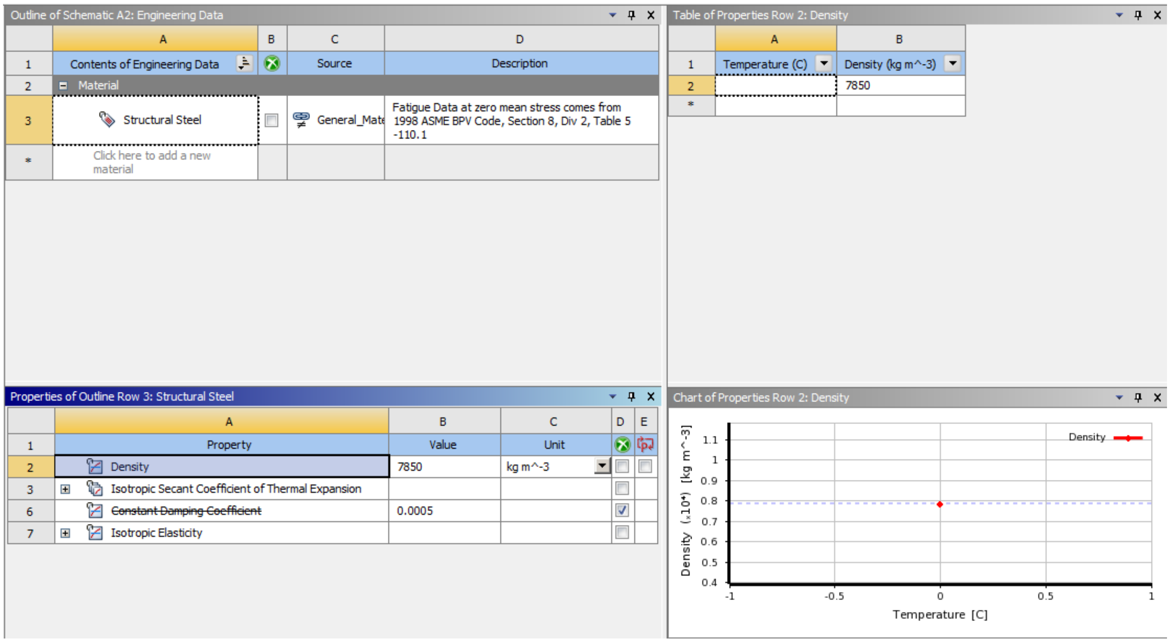

Figure 3.5: Engineering data list of the shaft

3) Design the geometry of the model.

The shaft design was with bearing were installed at both ends of the shaft, with the three disk at the middle part of the shaft. The geometry datum was based on the study done by Dumitru et al (2009).

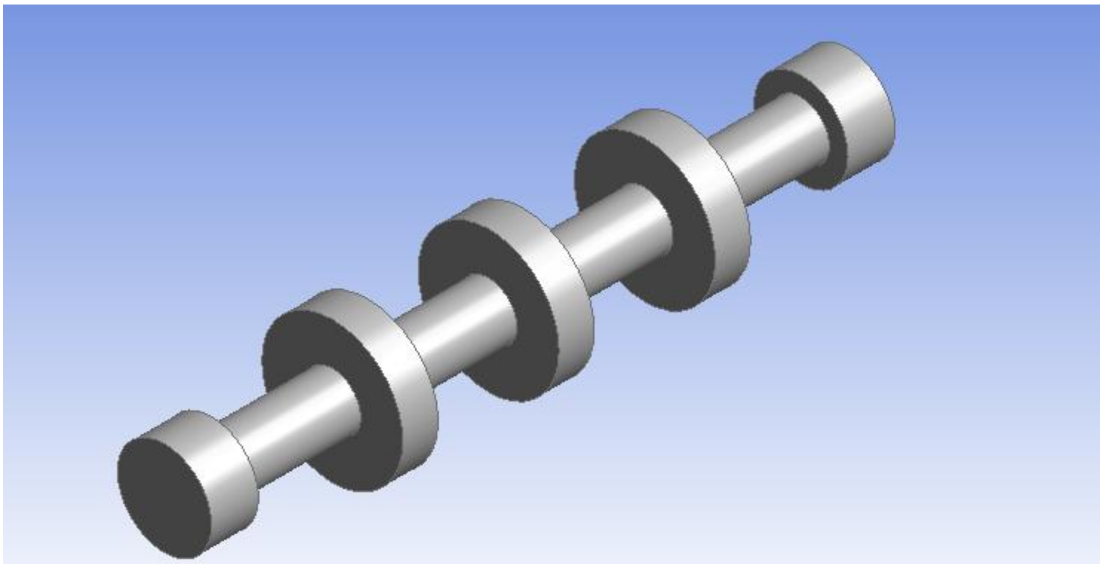

Figure 3.6: Isometric view of the shaft

4) Insert rotation and support to the model.

Bearing supports were installed at both ends of the shaft. Since there are two analyses with comparing the ball bearing supports with the magnetic bearing support. These are the properties of the shaft:

a) Volume: 2.7069e-004 m^3

b) Mass: 2.1249 kg

c) Rotation speed: 1000 rad/s to 4000 rad/s

5) Run the simulation and record the result developed.

The results of the simulation will be recorded and analyzed.

3.4 Project Gantt Chart And Key Milestone

3.4.1 Project Gantt Chart And Key Milestone for FYP 1

Table 3.1: Gantt chart for Final Year Project 1

Final Year Project 1															
No.	**Detail works/Week**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**
1	Project title selection and meeting with FYP coordinator and supervisor.														
2	Preliminary Research work and proposal preparation														
3	Extended proposal submission														
4	Proposal defence														
5	Project work continue														
6	Draft report submission														
7	Final report submission														

Legend: Process Suggested milestone

3.4.2 Project Gantt Chart And Key Milestone for FYP 2

Table 3.2: Gantt chart for Final Year Project 2

No	Activity	Week													
		1	2	3	4	5	6	7	8	9	10	11	12	13	14
	Final Year Project 2 (FYP 2)														
1	Simulation Preparation														
1.1	ANSYS exposure														
2	Simulation Progress														
2.1	ANSYS sketches of shaft with properties														
3	Simulation Result														
3.1	Acquire result from ANSYS														
3.2	Assess acceptable result from simulation														
4	Progress Report														
4.1	Final progress report & submission							▲							
5	Result & Analysis														
5.1	Result analyse and discussed														
6	Final Report														
6.1	Final report submission (soft bound)											▲			
7	Technical presentation														
7.1	Final presentation (VIVA)														▲
8	Final Report Submission (hard bound)														▲

Legend: Process ▲Suggested milestone

CHAPTER 4

RESULT AND DISCUSSION

4.1 Simulation Using Ansys

4.1.1 Simulation Of Ball Bearing On Three Disk Shaft

1) Damping ratio: 0.3
2) Stiffness Coefficient: 9.5493e-005

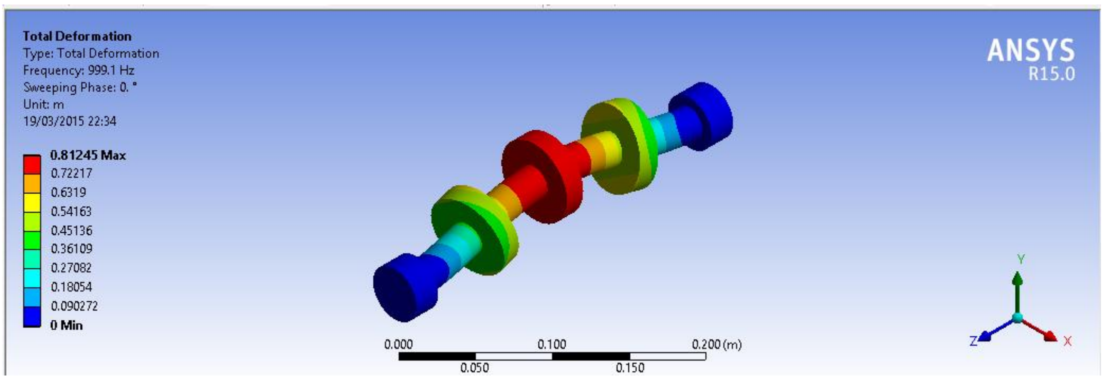

Figure 4.1: Total deformation simulation 1

Table 4.1: Damped frequency and modal damping ratio simulation 1

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	999.1	-334.3	0.31731
2	1	2	1007.1	-337.03	0.31735
3	1	3	1599.1	-1197.1	0.59927
4	1	4	0	-2306.3	N/A
5	1	5	1527	-2288.3	0.83181
6	1	6	1551.2	-2325	0.83186
7	2	1	995.15	-332.98	0.31731
8	2	2	1011.1	-338.35	0.31733
9	2	3	1599.1	-1197.1	0.59927
10	2	4	0	-2306.3	N/A
11	2	5	1515.1	-2270	0.83174

12	2	6	1563.9	-2343.3	0.83177
13	3	1	991.2	-331.65	0.3173
14	3	2	1015.2	-339.68	0.31732
15	3	3	1599.1	-1197.1	0.59927
16	3	4	0	-2306.3	N/A
17	3	5	1503.7	-2251.7	0.83161
18	3	6	1576.9	-2361.6	0.83163
19	4	1	987.27	-330.32	0.31729
20	4	2	1019.2	-341.02	0.3173
21	4	3	1599.1	-1197.1	0.59927
22	4	4	0	-2306.3	N/A
23	4	5	1492.6	-2233.5	0.83143
24	4	6	1590.3	-2379.8	0.83145

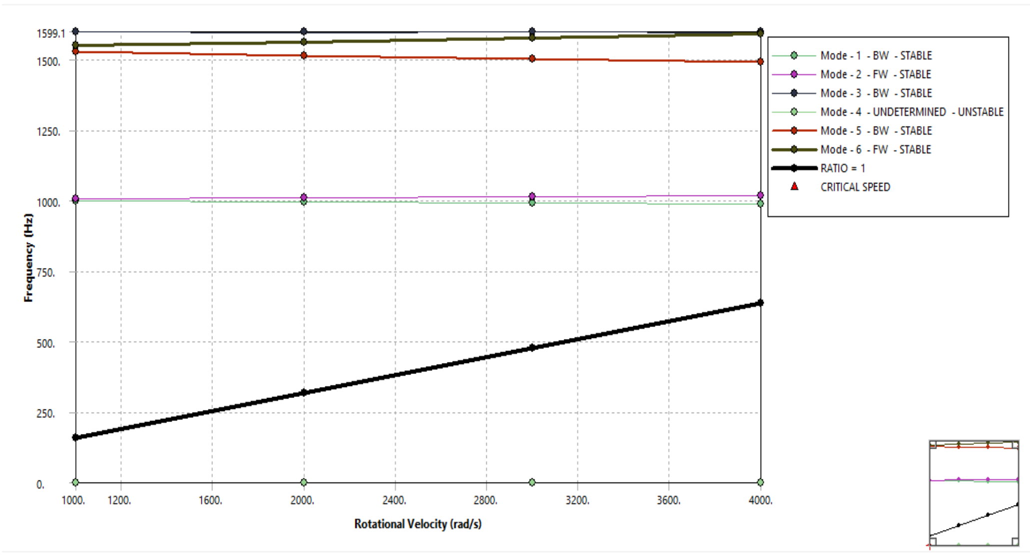

Figure 4.2: Campbell Diagram simulation 1

Table 4.2: Mode frequency simulation 1

Mode	Whirl Direction	Mode Stability	Critical Speed	1000. rad/s	2000. rad/s	3000. rad/s	4000. rad/s
1	BW	STABLE	NONE	999.1 Hz	995.15 Hz	991.2 Hz	987.27 Hz

2	FW	STABLE	NONE	1007.1 Hz	1011.1 Hz	1015.2 Hz	1019.2 Hz
3	BW	STABLE	NONE	1599.1 Hz	1599.1 Hz	1599.1 Hz	1599.1 Hz
4	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz	0. Hz	0. Hz
5	BW	STABLE	NONE	1527. Hz	1515.1 Hz	1503.7 Hz	1492.6 Hz
6	FW	STABLE	NONE	1551.2 Hz	1563.9 Hz	1576.9 Hz	1590.3 Hz

4.1.2 Simulation Of Active Magnetic Bearing

There are four case of simulation to study the effect of different value of damping ratio on the rotor supported with magnetic bearing.

Case One

1) Damping ratio: 0.3
2) Stiffness Coefficient: 9.5493e-005

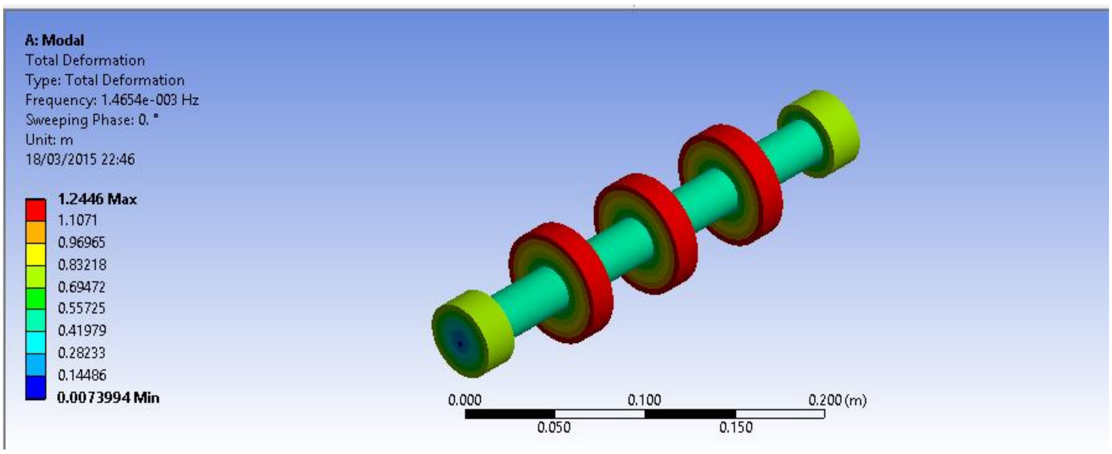

Figure 4.3: Total deformation simulation 2

Table 4.3: Damped frequency and modal damping ratio simulation 2

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	1.47E-03	-1.58E-10	1.08E-07
2	1	2	988.65	-326.49	0.31358
3	1	3	996.54	-329.13	0.31361
4	1	4	1659.9	-1516.1	0.67442
5	1	5	1544.4	-2245.9	0.82399
6	1	6	1568.2	-2280.9	0.82403

7	2	1	1.47E-03	-3.75E-10	2.56E-07
8	2	2	984.76	-325.2	0.31358
9	2	3	1000.5	-330.41	0.31359
10	2	4	1659.9	-1516.1	0.67442
11	2	5	1532.8	-2228.5	0.82393
12	2	6	1580.7	-2298.4	0.82395
13	3	1	1.47E-03	-3.95E-10	2.70E-07
14	3	2	980.88	-323.91	0.31357
15	3	3	1004.5	-331.71	0.31358
16	3	4	1659.9	-1516.1	0.67442
17	3	5	1521.5	-2211	0.82381
18	3	6	1593.5	-2315.8	0.82382
19	4	1	1.47E-03	-4.23E-10	2.89E-07
20	4	2	977.01	-322.61	0.31355
21	4	3	1008.4	-333	0.31356
22	4	4	1659.9	-1516.1	0.67442
23	4	5	1510.4	-2193.6	0.82364
24	4	6	1606.5	-2333.2	0.82365

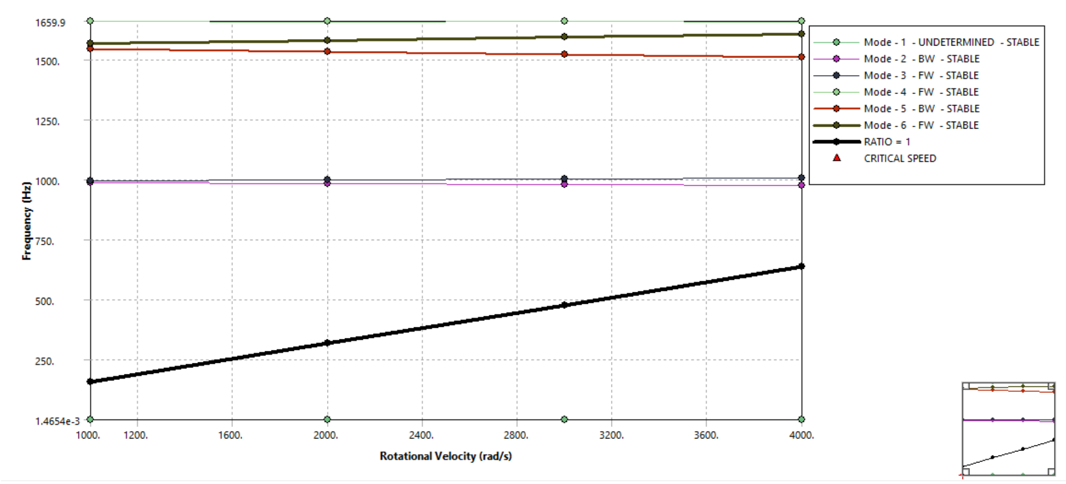

Figure 4.4: Campbell Diagram simulation 2

Table 4.4: Mode frequency simulation 2

Mode	Whirl Direction	Mode Stability	Critical Speed	1000. rad/s	2000. rad/s	3000. rad/s	4000. rad/s
1	UNDETERMINED	STABLE	NONE	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz
2	BW	STABLE	NONE	988.65 Hz	984.76 Hz	980.88 Hz	977.01 Hz
3	FW	STABLE	NONE	996.54 Hz	1000.5 Hz	1004.5 Hz	1008.4 Hz
4	FW	STABLE	NONE	1659.9	1659.9	1659.9	1659.9

				Hz	Hz	Hz	Hz
5	BW	STABLE	NONE	1544.4 Hz	1532.8 Hz	1521.5 Hz	1510.4 Hz
6	FW	STABLE	NONE	1568.2 Hz	1580.7 Hz	1593.5 Hz	1606.5 Hz

Case Two

1) Damping ratio: 0.5
2) Stiffness Coefficient: 1.5915e-004

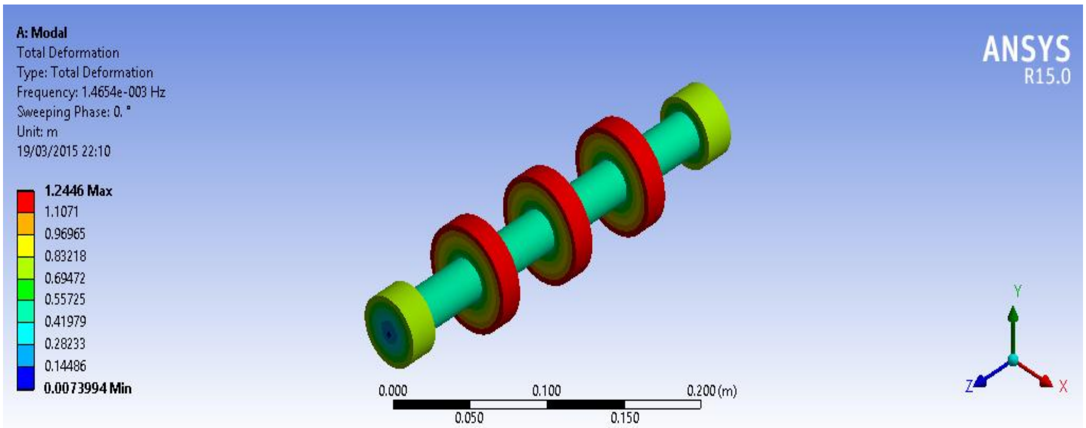

Figure 4.5: Total deformation simulation 3

Table 4.5: Damped frequency and modal damping ratio simulation 3

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	1.47E-03	-1.00E-09	6.83E-07
2	1	2	887.26	-543.91	0.52263
3	1	3	895.12	-548.79	0.52268
4	1	4	5.508	-1186.6	0.99999
5	1	5	0	-1373	N/A
6	1	6	0	-3680.8	N/A
7	2	1	1.47E-03	-1.00E-09	6.84E-07
8	2	2	883.37	-541.51	0.52263
9	2	3	899.08	-551.18	0.52265
10	2	4	11.023	-1186.5	0.99996
11	2	5	0	-1373	N/A
12	2	6	0	-3680.8	N/A
13	3	1	1.47E-03	-8.73E-10	5.96E-07
14	3	2	879.5	-539.11	0.52261
15	3	3	903.07	-553.58	0.52262
16	3	4	16.533	-1186.3	0.9999
17	3	5	0	-1373	N/A
18	3	6	0	-3680.8	N/A

19	4	1	1.47E-03	-9.73E-10	6.64E-07
20	4	2	875.66	-536.71	0.52257
21	4	3	907.08	-555.98	0.52258
22	4	4	22.039	-1186.1	0.99983
23	4	5	0	-1373	N/A
24	4	6	0	-3680.8	N/A

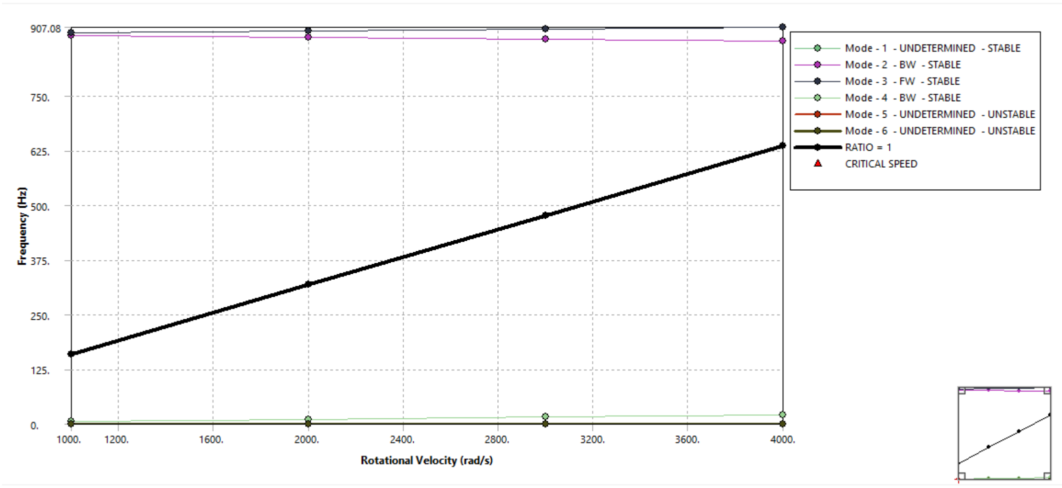

Figure 4.6: Campbell Diagram simulation 3

Table 4.6: Mode frequency simulation 3

Mode	Whirl Direction	Mode Stability	Critical Speed	1000. rad/s	2000. rad/s	3000. rad/s	4000. rad/s
1	UNDETERMINED	STABLE	NONE	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz
2	BW	STABLE	NONE	887.26 Hz	883.37 Hz	879.5 Hz	875.66 Hz
3	FW	STABLE	NONE	895.12 Hz	899.08 Hz	903.07 Hz	907.08 Hz
4	BW	STABLE	NONE	5.508 Hz	11.023 Hz	16.533 Hz	22.039 Hz
5	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz	0. Hz	0. Hz
6	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz	0. Hz	0. Hz

Case Three

1) Damping ratio: 0.01
2) Stiffness Coefficient: 3.1831e-006

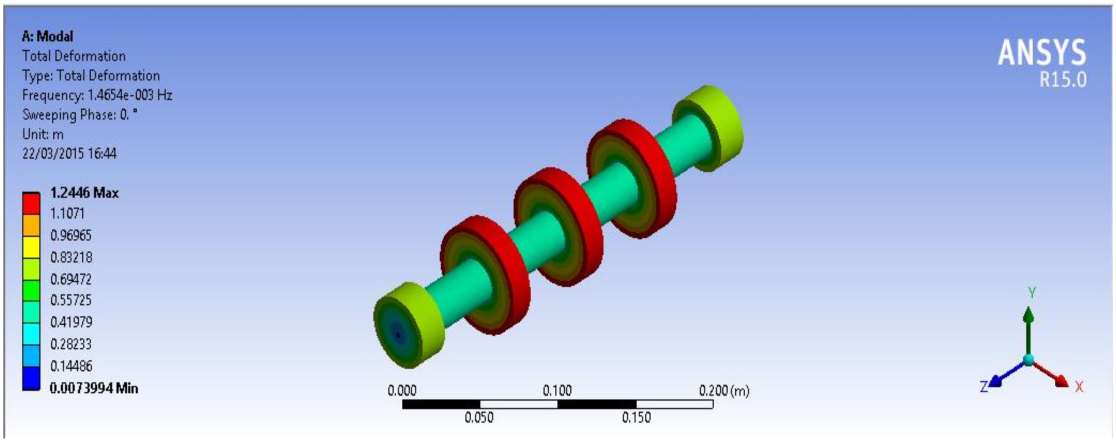

Figure 4.7: Total deformation simulation 4

Table 4.7: Damped frequency and modal damping ratio simulation 4

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	1.47E-03	0	0
2	1	2	1041.3	-10.885	1.05E-02
3	1	3	1049.2	-10.969	1.05E-02
4	1	4	2247.5	-50.538	2.25E-02
5	1	5	2733.7	-75.113	2.75E-02
6	1	6	2757.8	-75.783	2.75E-02
7	2	1	1.47E-03	-1.76E-11	1.20E-08
8	2	2	1037.4	-10.844	1.05E-02
9	2	3	1053.2	-11.009	1.05E-02
10	2	4	2247.5	-50.538	2.25E-02
11	2	5	2721.8	-74.785	2.75E-02
12	2	6	2769.9	-76.111	2.75E-02
13	3	1	1.47E-03	-2.04E-11	1.39E-08
14	3	2	1033.5	-10.803	1.05E-02
15	3	3	1057.1	-11.05	1.05E-02
16	3	4	2247.5	-50.538	2.25E-02
17	3	5	2709.9	-74.456	2.75E-02
18	3	6	2782	-76.44	2.75E-02

19	4	1	1.47E-03	-2.08E-11	1.42E-08
20	4	2	1029.7	-10.762	1.05E-02
21	4	3	1061.1	-11.091	1.05E-02
22	4	4	2247.5	-50.538	2.25E-02
23	4	5	2698.1	-74.126	2.75E-02
24	4	6	2794.2	-76.77	2.75E-02

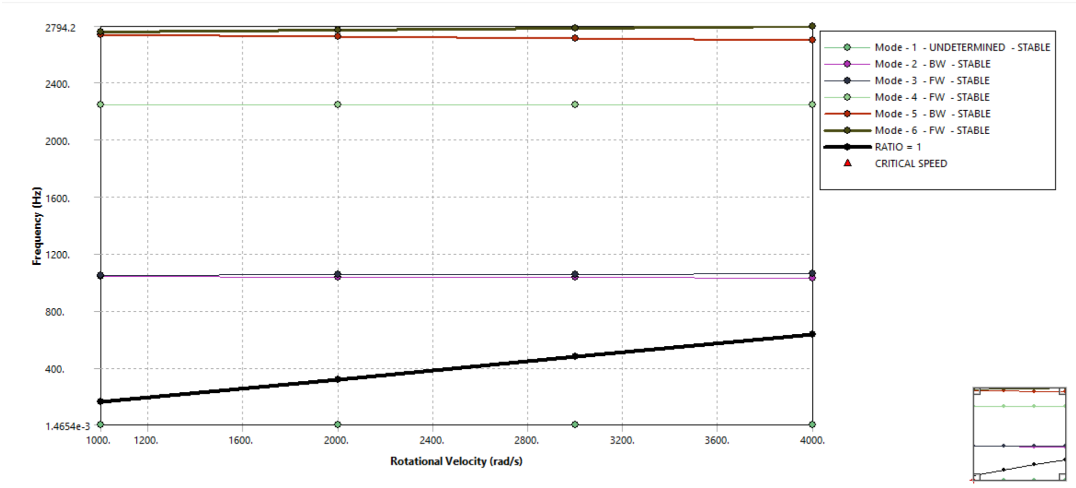

Figure 4.8: Campbell Diagram simulation 4

Table 4.8: Mode frequency simulation 4

Mode	Whirl Direction	Mode Stability	Critical Speed	1000. rad/s	2000. rad/s	3000. rad/s	4000. rad/s
1	UNDETERMINED	STABLE	NONE	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz
2	BW	STABLE	NONE	1041.3 Hz	1037.4 Hz	1033.5 Hz	1029.7 Hz
3	FW	STABLE	NONE	1049.2 Hz	1053.2 Hz	1057.1 Hz	1061.1 Hz
4	FW	STABLE	NONE	2247.5 Hz	2247.5 Hz	2247.5 Hz	2247.5 Hz
5	BW	STABLE	NONE	2733.7 Hz	2721.8 Hz	2709.9 Hz	2698.1 Hz
6	FW	STABLE	NONE	2757.8 Hz	2769.9 Hz	2782. Hz	2794.2 Hz

Case Four

1) Damping ratio: 0.001
2) Stiffness Coefficient: 3.1831e-007

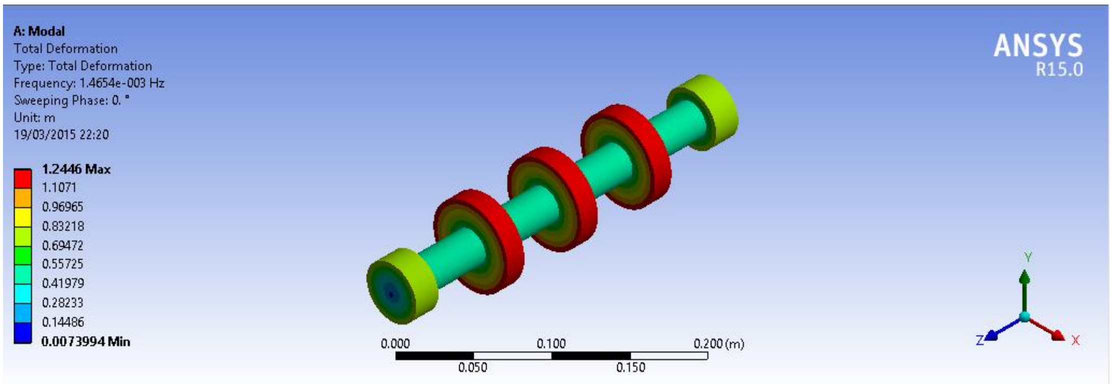

Figure 4.9: Total deformation simulation 5

Table 4.9: Damped frequency and modal damping ratio simulation 5

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	1.47E-03	0	0
2	1	2	1041.4	-1.0885	1.05E-03
3	1	3	1049.3	-1.0969	1.05E-03
4	1	4	2248.1	-5.0538	2.25E-03
5	1	5	2734.7	-7.5113	2.75E-03
6	1	6	2758.9	-7.5783	2.75E-03
7	2	1	1.47E-03	0	0
8	2	2	1037.5	-1.0844	1.05E-03
9	2	3	1053.2	-1.1009	1.05E-03
10	2	4	2248.1	-5.0538	2.25E-03
11	2	5	2722.8	-7.4785	2.75E-03
12	2	6	2770.9	-7.611	2.75E-03
13	3	1	1.47E-03	0	0
14	3	2	1033.6	-1.0803	1.05E-03
15	3	3	1057.2	-1.105	1.05E-03
16	3	4	2248.1	-5.0538	2.25E-03
17	3	5	2710.9	-7.4456	2.75E-03
18	3	6	2783.1	-7.644	2.75E-03
19	4	1	1.47E-03	0	0

20	4	2	1029.7	-1.0762	1.05E-03
21	4	3	1061.2	-1.1091	1.05E-03
22	4	4	2248.1	-5.0538	2.25E-03
23	4	5	2699.1	-7.4127	2.75E-03
24	4	6	2795.3	-7.6769	2.75E-03

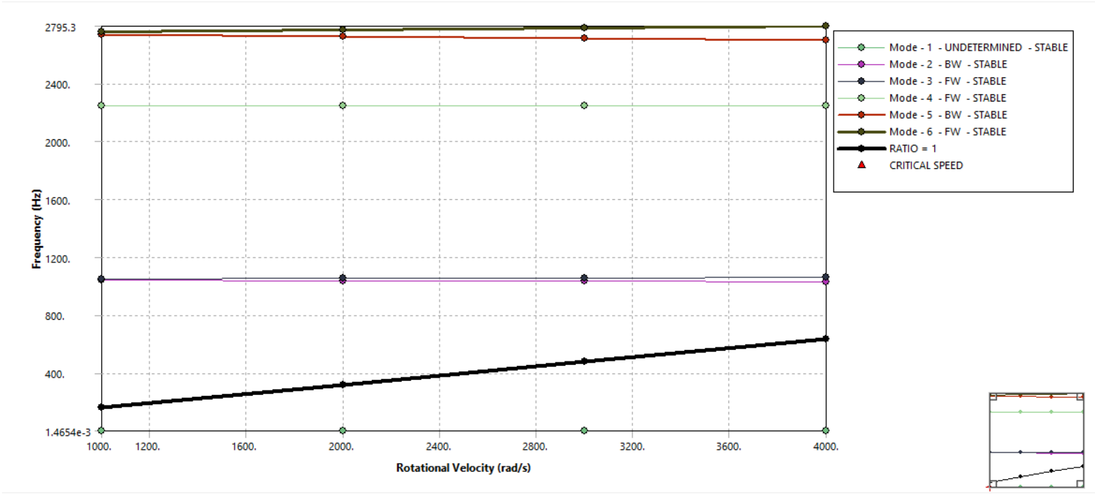

Figure 4.10: Campbell Diagram simulation 5

Table 4.10: Mode frequency simulation 5

Mode	Whirl Direction	Mode Stability	Critical Speed	1000. rad/s	2000. rad/s	3000. rad/s	4000. rad/s
1	UNDETERMINED	STABLE	NONE	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz	1.4654e-003 Hz
2	BW	STABLE	NONE	1041.4 Hz	1037.5 Hz	1033.6 Hz	1029.7 Hz
3	FW	STABLE	NONE	1049.3 Hz	1053.2 Hz	1057.2 Hz	1061.2 Hz
4	FW	STABLE	NONE	2248.1 Hz	2248.1 Hz	2248.1 Hz	2248.1 Hz
5	BW	STABLE	NONE	2734.7 Hz	2722.8 Hz	2710.9 Hz	2699.1 Hz
6	FW	STABLE	NONE	2758.9 Hz	2770.9 Hz	2783.1 Hz	2795.3 Hz

4.2 Discussion Of The Simulation

Table 4.11: Simulation result

Ball Bearing	Magnetic Bearing			
Damping ratio: 0.3 Stiffness Coefficient: 9.5493e-005	Damping ratio: 0.3 Stiffness Coefficient: 9.5493e-005	Damping ratio: 0.5 Stiffness Coefficient: 1.5915e-004	Damping ratio: 0.01 Stiffness Coefficient: 3.1831e-006	Damping ratio: 0.001 Stiffness Coefficient: 3.1831e-007
Mode 4 Unstable	All modes stable	Mode 5 & 6 Unstable	All modes stable	All modes stable
No Critical Speed Detected	No Critical Speed Detected	No Critical Speed Detected	No Critical Speed Detected	No Critical Speed Detected

There are two type of analysis where comparing the effect of using ball bearing and active magnetic bearing. The ball bearing analysis was done with the value of damping ratio 0.3 and stiffness coefficient of 9.5493e-005.

For active magnetic bearing, the simulation was done with three different values of damping ratio 0.3 with stiffness coefficient of 9.5493e-005, damping ratio 0.5 with stiffness coefficient: 1.5915e-004, damping ratio 0.01 with stiffness coefficient 3.1831e-006 and damping ratio 0.001 with stiffness coefficient 3.1831e-007.

The result developed were:

1) The total deformation of the model
2) The vibration frequency
3) Campbell diagram
4) The stability

With the same value of damping ratio and stiffness coefficient, the ball bearing simulation is unstable compare to active magnetic bearing. There is one mode which is unstable with the bearing damping ratio 0.3 and stiffness coefficient of 9.5493e-

005 which is mode 4. Magnetic bearing does stabilize the shaft with the same properties of the bearing. The total deformation also show huge different between both simulation, heat does make the shaft to deform compare to frictionless support of magnetic bearing.

For magnetic bearing the simulation was done with increament of damping ratio values, to analyze the effect of increasing the damping ratio values toward the stability of the rotor itself.

Based on the result, when the damping ratio is 0.001, 0.01 and 0.3, the rotor is stable and have no unbalance mode detected. However, when the damping ratio was set to 0.5, the rotor was unstable at mode 3 and 6. This show that the damping ratio of 0.5 is not suitable for the rotor to work on since the rotor will be unstable.

However, from all three simulation, there are none critical speed were detected that will affect the vibration of the rotor itself. The damping ratio does effect the stability of the system.

Not all of the system will be suitable to work under low damping ratio, the system have its best condition to be operate at. For this shaft, it is more suitable to work at lower damping ratio value, since, it is stable and the mode frequency is much lower compare to others. The average operating frequency will be higher, thus increasing the gap between the operating frequency than the critical line in the campbell diagram. Noted that, if any line of modes cross the critical line will result in critical speed that can eventually influence the vibration of the rotor itself. Thus, it would be safer to avoid any cross at the critical line during the rotor movement.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

Magnetic bearing does give an opportunity to the engineering world to break the limit from the inadequacy of conventional bearing. Same as other types of bearings, the magnetic bearing come up with stiffness and damping except the stiffness and damping vary as a function of disturbance frequency. By varying the damping ratio of the system, this study can show the instability during operation. This study show that the system does have their own best parameters during operation that can enhance the stability of the system. For this type of shaft, it would be safer to work with lower damping ratio and stiffness coefficient to avoid any critical speed detected. The lower the damping ratio and stiffness coefficient the higher the gap of operating mode line and the critical line in the campbell diagram.

5.2 Recommendation

For future study, there are many cases that can improve this study. In real world, there would be cases where the shaft have unbalance force, extra force, external force, flexible rotor and unbalance rotor. All these cases can cause unbalance towards the rotor itself. With further study, the imbalance rotor can be stabilize with magnetic bearing. Thus, to enhance the knowledge of the effect of magneric bearing in stabilize the rotor should include these new cases.

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