DYNAMICS OF MAGNETICALLY SUPPORTED RIGID ROTOR IN AUXILIARY BEARING

By

MUHAMMAD BIN HASAN AZHARI

FINAL PROJECT REPORT

A project dissertation submitted to the

Department of Mechanical Engineering

UNIVERSITI TEKNOLOGI PETRONAS

in Partial fulfillment of the requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

Universiti Teknologi PETRONAS

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CERTIFICATION OF APPROVAL

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_____

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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.

_____

MUHAMMAD BIN HASAN AZHARI

ABSTRACT

This paper presents a study of the dynamic behavior of a rotor system supported by magnetic bearing and auxiliary bearing. Specifically, the work will investigate the range of realistic operating parameters. That will 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 MATLAB and ANSYS software. By varying the damping ratio and stiffness coefficient, the simulation shall produce different results. The results will be recorded, analyzed and discussed.

ACKNOWLEDGEMENT

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CHAPTER 1

INTRODUCTION

1.1 BACKGROUND STUDY

In an ever changing world, the limits of current designs are constantly pushed further, requiring genuinely new ideas and technologies to meet new targets. Concerning about rotating equipments such as pumps, turbines and jet engine, an idea had come out to improve the system which is magnetic bearings.

The conventional ball bearings which are used nowadays have some significant weaknesses which are vibration, heat loss due friction, the need for lubrication system and short service intervals. These feature of conventional bearings leads to the efficiency drops, costly maintenance and decrease in production. Magnetic bearings are expected to overcome these issues.

Magnetic bearings is a system which supports a rotating shaft, without any physical contact by suspending the rotor in the air, with an electrically controlled magnetic force or permanent magnet. Magnetic bearing works on the principle of electromagnetic suspension. Basically, magnetic bearings will consist of an electromagnet assembly, a set of power amplifiers which supply current to the electromagnets, a controller, and gap sensors with associated electronics to provide the feedback required to control the position of the rotor within the gap.

This magnetic bearing operation is complemented by auxiliary bearing. Auxiliary bearing is the bearing installed in the magnetically supported rotating machines with the function of supporting the rotor during non-operational period and to protect the rotating assembly from being damaged during unstable rotation or loss of power during operation.

This technology of magnetic bearings with its features of high maximum speed, high operating temperature and high efficiency is a big potential for the development of rotating machines and engines in the future.

Figure 1.1: Eight-pole radial magnetic bearing

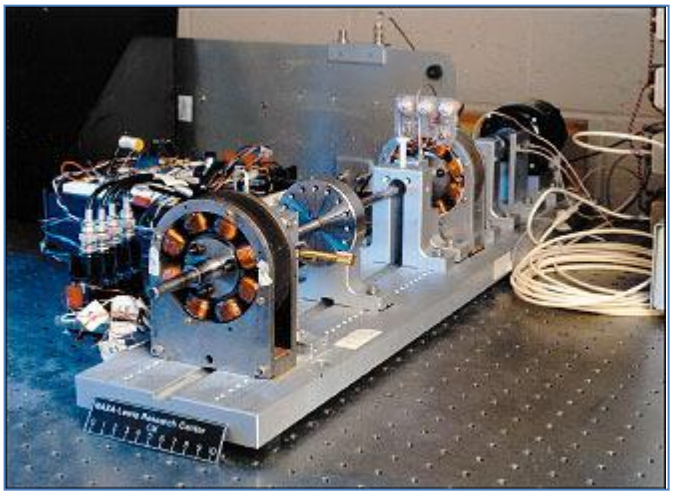


Figure 1.2: Apparatus for operating rotor with magnetic bearing support

1.2 PROBLEM STATEMENT

Magnetic bearing is the on blooming technology for rotating equipment application. The promising features of magnetic bearing are contact free rotor, friction-free rotation and lubricant-free operation. To further refine this new technology, this research is carried out. This research is objectively to study the dynamics of magnetic supported rotor. Thus finding the ideal operating parameters of the magnetically supported rigid rotor through simulations on ANSYS

1.3 OBJECTIVES

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

I. To study the vibration of magnetically of magnetically supported rotor.
II. To identify the limit to instability by varying the parameters during operation.
III. To study the construction and working principle of magnetic bearing support.
IV. To suggest recommendations on improvement of stability of magnetic supported rotor system.

CHAPTER 2

LITERATURE REVIEW

To progress with this study/project, I have done references to existing papers and journals that are related to the dynamics of magnetically supported rotor.

2.1 Bearing in Rotating Equipments

Rotating equipments are the machines that involve rotating operations. The examples of this kind of machine are pumps, turbines, jet engine and helicopter propeller. For all rotating equipments, there is a crucial part which is: bearing.

The use of bearings is essential to all types of machines, in that they provide the function of supporting another piece or component in a desired position. Two major types include radial and axial bearings. Furthermore, bearings are usually implied to be supporting a rotating object or shaft (Hillyard J., 2006).

Conventional type of bearing has an obvious limitations which are efficiency drop due to friction, heat generated from contact surfaces, limited speed, more vibration occur and the need for lubrication.

Figure 2.1: Conventional ball bearings

Figure 2.2: Pump in oil and gas industry

2.2 Magnetic Bearing: The Introduction

Magnetic bearing is the new technology developed from decades ago. Magnetic bearings have reached the point in their development where they can be considered a mature technology, and as such, should gain wide acceptance by the rotating equipment industry. While current industrial installation has proven durable and reliable, we have just begun to peer into the world of possibilities for advanced application of this technology to rotating equipment
(Alves P., 2012).

These magnetic bearing can be further classified as Active magnetic bearing and passive magnetic bearing. Active bearings are electrically controlled with some type of controller, whereas passive bearings are not electrically powered and thus have no control mechanism (Hillyard J., 2006). According to S.C.Mukhopadhyay, C. Gooneratne and G. Sen. Gupta (2004), Active Magnetic Bearing is using solely on electromagnets while Passive Magnetic Bearing us using in combination of electromagnets and permanent magnets.

2.3 Magnetic Bearing: The Mechanism of Operation

The magnetic bearing works just like normal mechanical bearing except the bearing functionality is achieved utilizing magnetic field. In magnetic bearing system the rotor is levitated in the magnetic field (S.C.Mukhopadhyay, C. Gooneratne and G. Sen. Gupta, 2004).

Electromagnetism

A magnetic field exists between two poles of opposite polarity, denoted as the north and south poles. The magnetic field lines are emitted radially from the north pole in every direction and end up at the south pole, then going through the object back to the north pole, creating a closed path. Such magnetic field lines can also be created by an electro

magnet, which functions by running current through a wire that is coiled around a piece of ferromagnetic material
(Hillyard J., 2006).

Most of the application of magnetic bearing is using Active Magnetic Bearing (AMB) which means the device will have fully electromagnets. As of that, I will focus on the mechanism of electromagnet in this study of magnetic bearing.

Now, let us take a look into how electromagnet is created.
Electromagnets are magnets that are created when there is electric current flowing in a wire. The simplest electromagnet uses a coil of wire, often wrapped around some iron. Since iron is magnetic, it concentrates the magnetic field created by the current in the coil.

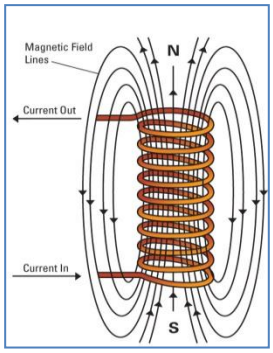


Figure 2.3: Electromagnet

Figure 2.4:
The simplest electromagnet arrangement (solenoid)

Active Magnetic Bearing (ABM)

The active magnetic bearing is a support system for a rotor/rotating shaft that uses magnetic force to hold the rotor in place. Same as other bearing types, the magnetic bearing can be characterized in terms of stiffness, damping, and load capacity. A magnetic bearing consists of multiple electromagnetic coils attached to a ferromagnetic stator. The coils are arranged such that opposite poles are adjacent, maximizing magnetic flux through the rotor. A ferromagnetic, laminated rotor stack is attached to the shaft to provide the flux path and attractive magnetic forces while minimizing eddy current formation (Clark D.J., Jansen M. J., Montague G.T., 2004).

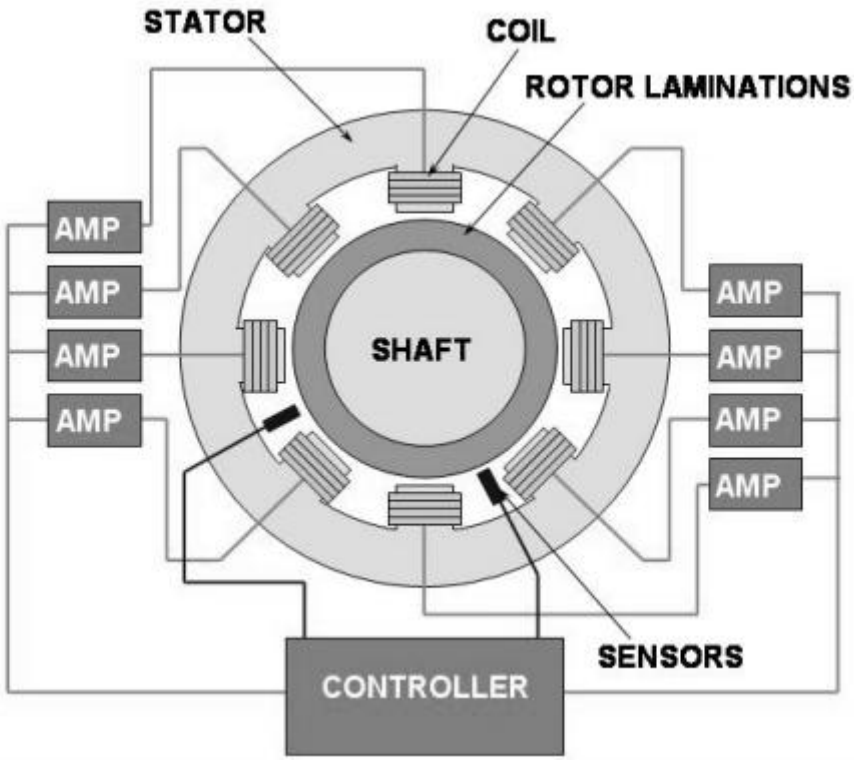


Figure 2.5: Basic arrangement of a magnetic bearing

Advantages of Magnetic Bearing

Generally, the invention and development of magnetic bearings is to improve the efficiency and effectiveness of rotating equipments. Its features and behaviors are expected to overcome the limitations of the conventional ball bearings.

Magnetic bearings are increasingly being used for a large variety of applications. Their unique features make them attractive for solving classical rotor-bearing problems in a new way and allow novel design approaches for rotating machinery. Classical limitations can be overcome and feature ranges can be extended (Schweitzer G., 2002).

In the review of the journals and papers available, we can come to an agreement that the advantages of magnetic bearing can be listed into several aspects which are:

1. Energy loss from friction
2. Lubrication and Maintenance
3. Speed
4. Load
5. Operating Temperature

I. Energy Loss from Friction

There are numerous advantages to using magnetic bearings, the most notable being contact-free, in that the magnetic force is used to support the object as opposed to contact between two surfaces (Hillyard J., 2006). Since it is a contact free operation, thus the energy loss due friction is highly reduced. The same finding is shared by Paul Alves (2012) as he stated in his paper that one or two percent increase of overall efficiency is achievable due to reduced friction loss.

II. Lubrication and Maintenance

Another significant advantage of magnetic bearing is that the system does not require lubrication system and the maintenance cost is minimal.

The magnetic bearing can work without usage of lubricant, without emission of particles due to friction and without wear and are moreover very reliable. A lot of systems are running for more than ten years without maintenance (Burdet L., 2006).

A few other physicists also supported this point. Magnetic bearing contributes to reduced weight, no bearing contact, no wear, and less maintenance, the subtraction of an oil, lube, and cooling system (Clark D.J., Jansen M.J., Montague G.T., 2004).

III. Speed

According to Paul Alves (2012), one of the advantages of magnetic bearing is that it is capable to operate the rotor/rotating shaft at higher speed. The achievable speed is higher than the speed of rotor supported by ball bearing.

Table 2.1: Comparison of top speed using various bearings
(Clark D.J., Jansen M.J., Montague G.T., 2004)

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

IV. Load

One aspect to look into in application magnetic bearing is the load it capable to operate. The load capacity depends on the arrangement and geometry of the electromagnets, the magnetic properties of the material, of the power electronics, and of the control laws (Schweitzer G., 2002).

Table 2.2: Comparison of load limit using various bearings (Clark D.J., Jansen M.J., Montague G.T., 2004)

Type of Bearing	Roller Bearing	Foil Bearing	Magnetic Bearing
Documented Load Capacity	Varies: wear, heating, lube breakdown Highest load capacity per square inch **(~300 psi)**	Proportional to rotation speed 0 to ~1000 lbs (4”dia.) max range for largest bearing size **(~100 psi)**	For entire speed and temperature range 1000 lb/axis **(116 psi)** (3” dia.)

Even though it is proven that maximum load that magnetic bearing is capable to carry is higher than the foil and ball bearings, there is a concern and a precaution in applying load.

Any applied load that exceeds the force that the magnetic bearings can provide will cause the rotor to break away from its centre position and touch down on its retainer bearings

(Schweitzer G., 2002).

V. Temperature

Magnetic bearing has a big potential for application in gas turbines and aircrafts. In order to utilize the full potential of active magnetic bearings, operation in gas turbine and aircraft engines, it is important for the magnetic bearing to work properly at high temperatures (Schweitzer G., 2002).

Figure 2.6: Equipment to test magnetic bearing at high temperature

According to Daniel J. Clarke et al (2004), magnetic bearing is capable to operate in extreme temperature.

Table 2.3: Comparison of operating temperature of various bearings (Clark D.J., Jansen M.J., Montague G.T., 2004)

Type of Bearing	Roller Bearing	Foil Bearing	Magnetic Bearing
Maximum Operating Temperature (°c)	**180-260**	**650**	**540**

2.4 Auxiliary Bearing

In order to provide a successful magnetic bearing system, it is also necessary to provide a reliable auxiliary bearing landing system (L.P. Tessier, 1997). Auxiliary bearing is a kind of bearing that is attached to the structure some clearance to the magnetically supported rotor. The purpose of this auxiliary bearing is to support the rotor during a delevitation coast down from full speed without excessive rotor deflections or damage to any of the rotating or stationary components (L.P. Tessier, 1997). In addition to this, auxiliary bearing also share momentary overloads which are caused by shock and blade loss with magnetic bearing (J.I. Inayat-Hussain, 2010)

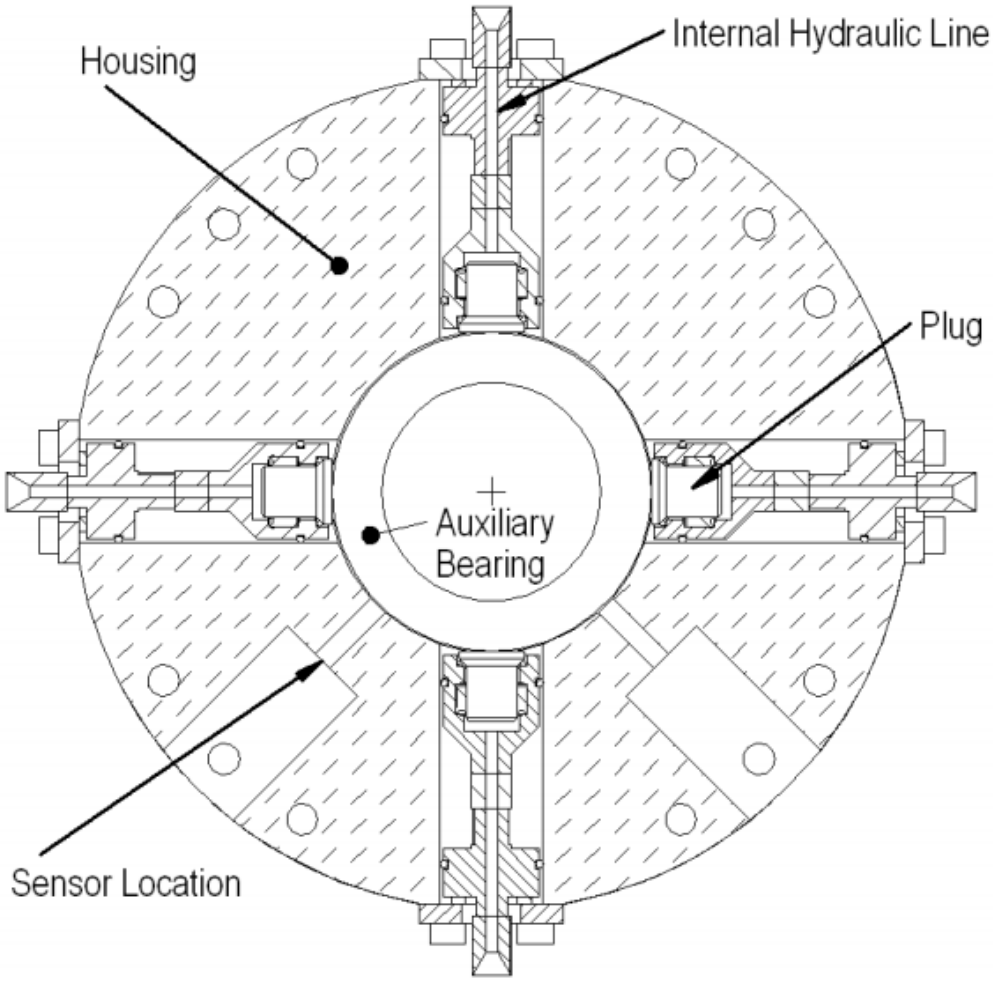


Figure 2.7: The Assembly of Auxiliary Bearing
(I.S. Cade, M.N Sahinkaya, C.R. Burrows, P.S. Keogh, 2008)

2.5 Control System

The AMB system is excited by unbalance forces of the rotor. Especially for the case of large rotor eccentricities, causing large rotor displacements. The behavior of the system is discussed. As of this respective situation, nonlinear phenomena may occur (N. Steischaden, H. Springer, 1999). Active Magnetic Bearings applied to rotating machines can be considered as a common mechatronic application. To regulate the rotor/shaft, a possible application is an electronic control unit which will command the electromechanical actuators through closed loop control architecture of displacement and current sensors (A. Tonoli, A. Bonfitto, M. Silvagni. L.D. Suarez, 2012). This closed loop consists of the rotor, position sensors, a digital controller, power amplifiers and electromagnets applying the supporting and operating forces (K.J. Hoffman, D. Laier, R. Markert, 1997). The idea of having a control system is to further reduce the vibration and thus making the rotation of rotor/shaft smoother and more linear.

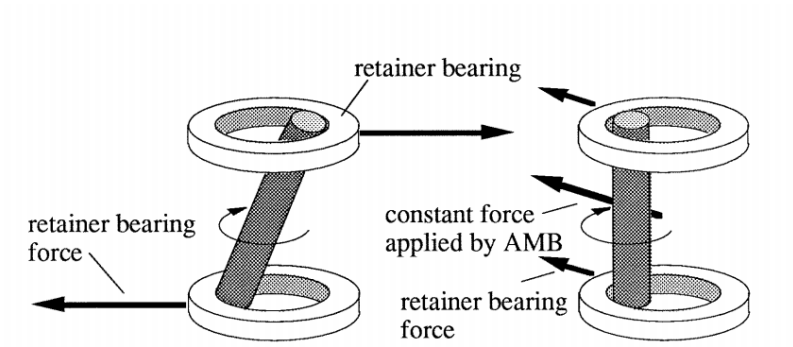


Figure 2.8: Non-Linearity that may occur.
(Larsonneur R., 1990)

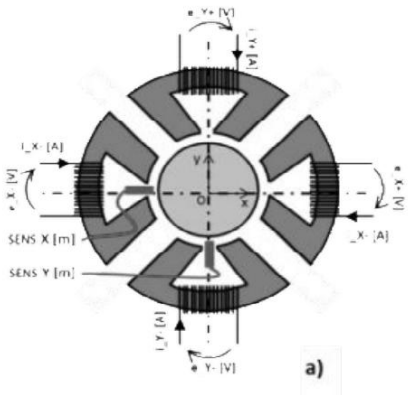


Figure 2.9: Position of sensors from the shaft
(A. Tonoli, A. Bonfitto, M. Silvagni. L.D. Suarez, 2012).

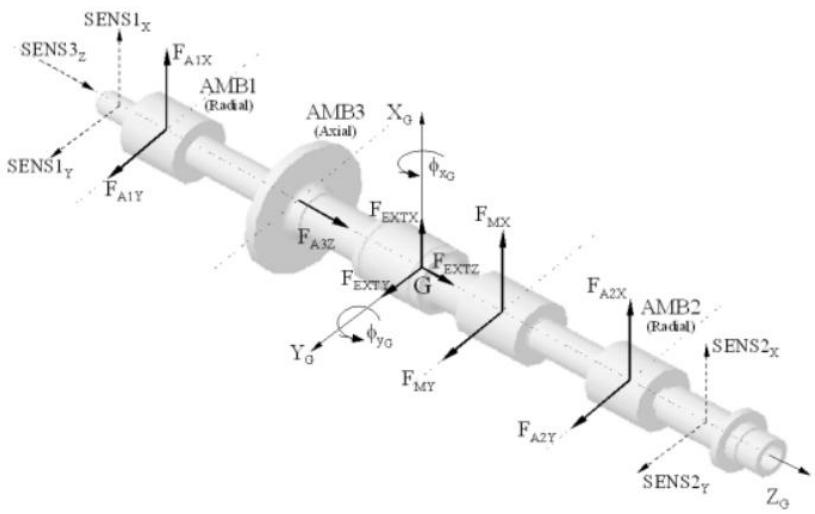


Figure 2.10: The location of sensors along the shaft.
(A. Tonoli, A. Bonfitto, M. Silvagni. L.D. Suarez, 2012).

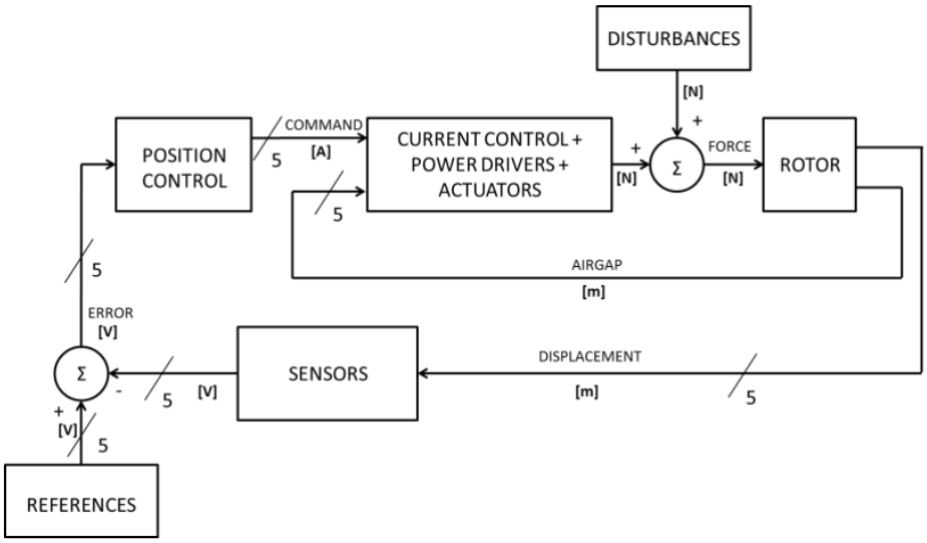


Figure 2.11: The architecture of control system.
(A. Tonoli, A. Bonfitto, M. Silvagni. L.D. Suarez, 2012).

CHAPTER 3

THEORY AND PRINCIPLES

The foundation of this research study is the two degree of freedom vibration theory. It is a forced vibration involving damping and stiffness.

The basic equation:

$$[\boldsymbol{m}]\ddot{x} + [\boldsymbol{c}]\dot{x} + [\boldsymbol{k}]\boldsymbol{x} = \boldsymbol{f}$$

$$[\boldsymbol{m}]\ddot{y} + [\boldsymbol{c}]\dot{y} + [\boldsymbol{k}]\boldsymbol{y} = \boldsymbol{f}$$

Where

[m] = mass matrices

[c] = damping matrices

[k] = stiffness matrices

x = displacement vector in x-direction

y = displacement vector in y-direction

f = force vector

The equations of motion, with the presence of weight of the rotor, magnetic bearing force and auxiliary bearing force will be as follows:

$$[\boldsymbol{M}]\ddot{\boldsymbol{x}} + [\boldsymbol{D}]\dot{\boldsymbol{x}} + [\boldsymbol{K}]\boldsymbol{x} = (\boldsymbol{Fm} + \boldsymbol{Fa})\boldsymbol{cos\ \omega t}$$

$$[\boldsymbol{M}]\ddot{y} + [\boldsymbol{D}]\dot{y} + [\boldsymbol{K}]\boldsymbol{y} = (\boldsymbol{Fm} + \boldsymbol{Fa})\boldsymbol{sin\ \omega t} - \boldsymbol{W}$$

$M\ddot{x}$ = inertia forces in x-direction. $M\ddot{y}$ = inertia forces in y-direction

$D\dot{x}$ = damping forces in x-direction. $D\dot{y}$ = damping forces in y-direction

Kx = stiffness in x-direction. Ky = stiffness in y-direction

Fm = magnetic force

Fa = auxiliary bearing force (only when rotor in contact with auxiliary bearing)

Force of magnetic bearing equation:

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

Where

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

Force of auxiliary bearing equation:

CHAPTER 4

METHODOLOGY

4.1 Simulation using MATLab Software Procedures:

To study the vibration amplitude of the rotor

Low speed rotation (50 rad/s):

1. The software is turned on.
2. The coding for the simulation is put in.
3. The unbalance force is set to 0.5 N (representing the usage conventional ball bearing).
4. The simulation is run.
5. The result graph is observed and analyzed.
6. Step 1 until step 5 is repeated by adjusting the unbalance force to 0.25 N (representing the usage of magnetic bearing).

High Speed Rotation (250 rad/s):

1. The software is turned on.
2. The coding for the simulation is put in.
3. The unbalance force is set to 0.5 N (representing the usage conventional ball bearing).
4. The simulation is run.
5. The result graph is observed and analyzed.
6. Step 1 until step 5 is repeated by adjusting the unbalance force to 0.25 N (representing the usage of magnetic bearing).

4.2 Simulation using ANSYS Software Procedures:

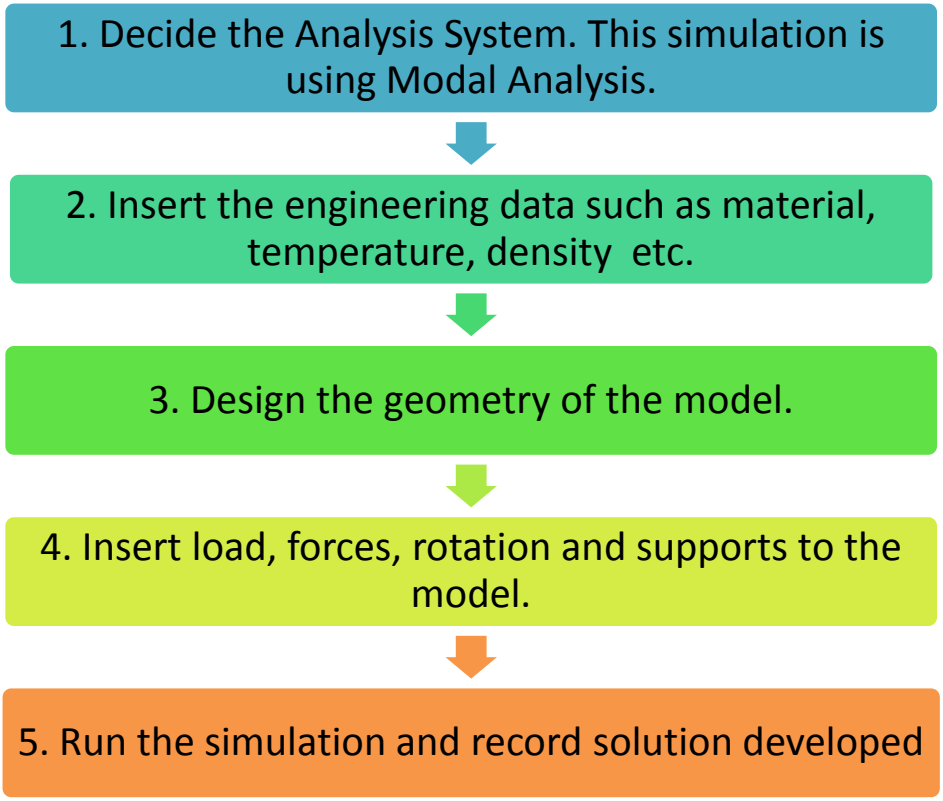
I used ANSYS Workbench to progress with the simulation. ANSYS Workbench is the latest version of the software. ANSYS Workbench combines the strength of the core product solvers with the project management tools necessary to manage the project workflow. In ANSYS Workbench, analyses are built as systems, which can be combined into a project. The project is driven by a schematic workflow that manages the connections between the systems. From the schematic, we can interact with applications that are native to ANSYS Workbench (called workspaces) and that display within the ANSYS Workbench interface and we can launch applications that are data-integrated with ANSYS Workbench, meaning the interface remains separate, but the data from the application communicates with the native ANSYS Workbench data. Native workspaces include Project Schematic, Engineering Data, and Design Exploration (Parameters and Design Points). Data-integrated applications include the Mechanical APDL application (formerly ANSYS), ANSYS FLUENT, ANSYS CFX, the Mechanical application (formerly Simulation), etc.

Step 1: Decide the Analysis System. This simulation is using Modal Analysis.

The Analysis System is the set of prescribed logic and mathematical operation in ANSYS that designed to satisfy the user's interest.

In my simulation of magnetic bearing in rotor, Modal System Analysis is used. Modal analysis includes the study of vibration and stability of the model.

Process Flow of Using ANSYS



Step 2: Insert the Engineering Data.

The engineering data is the information about the model. It includes the material, density, temperature etc.

For this simulation, the engineering data is as follows:

i. Material: Structural steel

ii. Density: 7850 kg/m^3

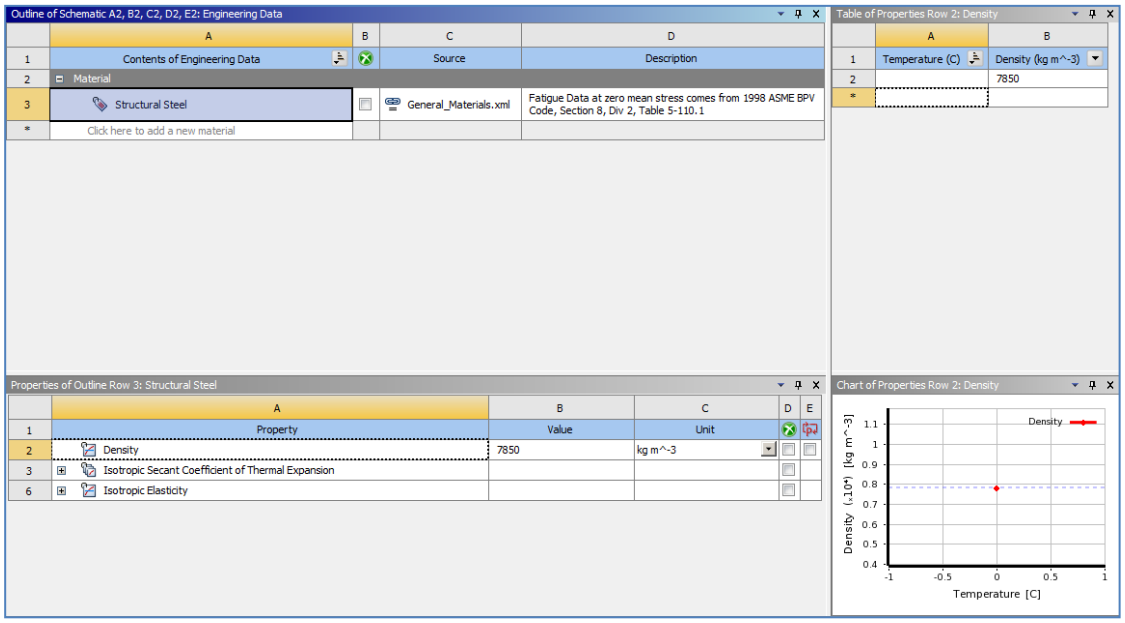


Figure 4.1: ANSYS Engineering Data Interface

Step 3: Design the Geometry of the Model.

The model I designed for this simulation is a shaft attached to a load and the bearing at both ends of it.

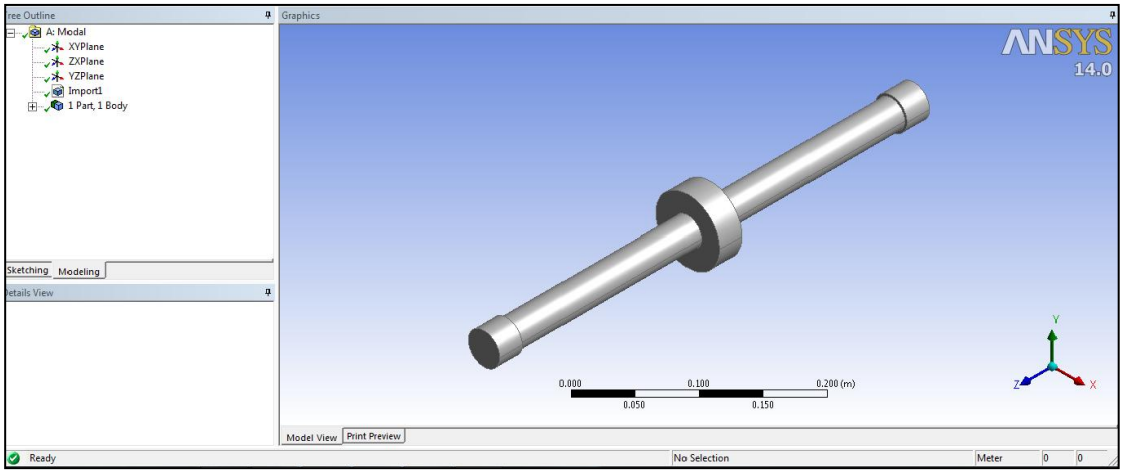


Figure 4.2: Bearing Design

Step 4: Insert load, forces, rotation and supports to the model.

Displacement supports are put to the model. Assuming that is the support from magnetic bearing.

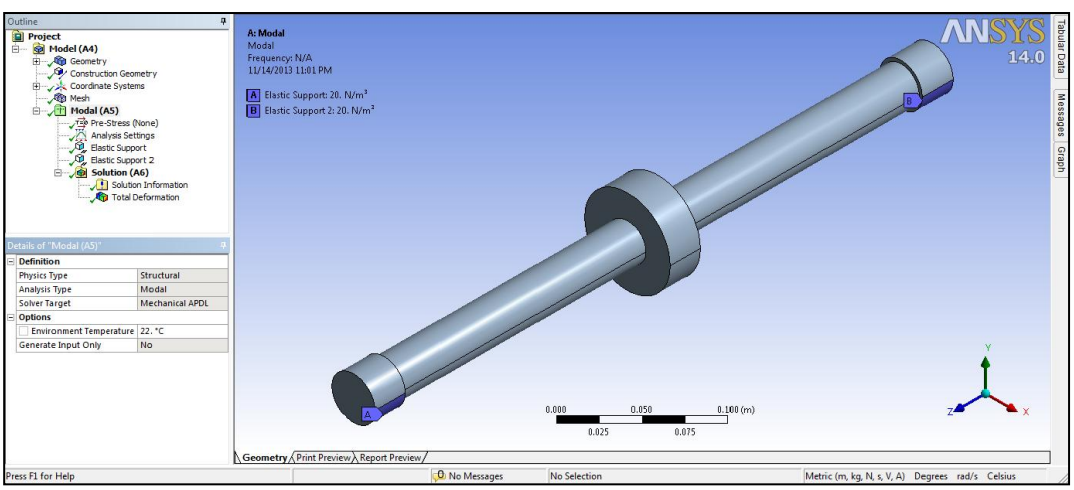


Figure 4.3: Bearing Design with Displacement Support

Step 5: Run the simulation and record the result developed.

The results of the simulation developed will be:

i. The deformation of the model
ii. Campbell Diagram
iii. The vibration frequency
iv. The stability

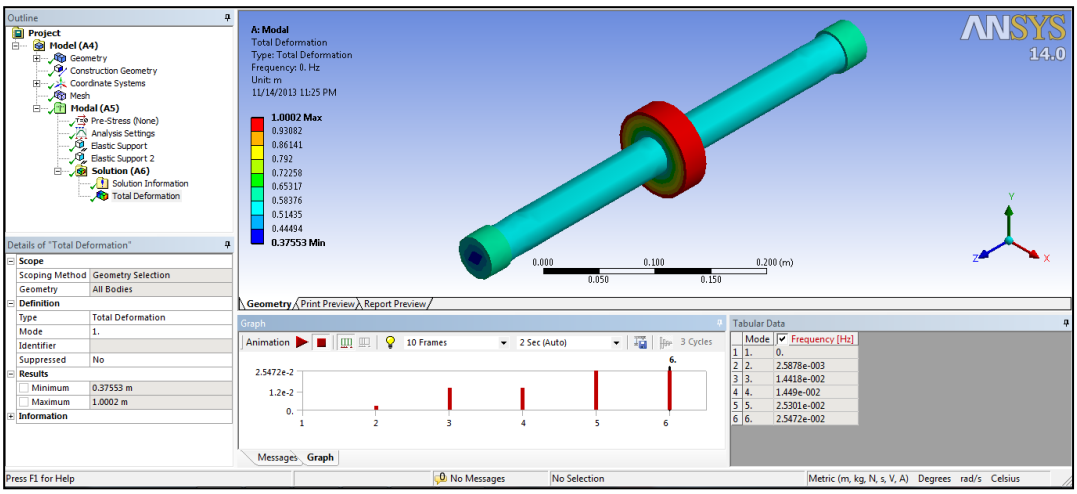


Figure 4.4:
Result of Modal Analysis Simulation showing Bearing Deformation and Vibration Frequency

CHAPTER 5

RESULT AND DISCUSSION

5.1 Simulation using MATLab: To study the vibration amplitude of the rotor

A simulation is run using MATLab. The result comprises of vibration amplitude graphs. Each of the graphs has description and discussion on how and why the result is so.

The basic coding of this simulation on MATLab:

for t=1:40 x(t)=(**ω***t-20); end y = **F***cos(x) plot(x,y)	*Where:* ω = *shaft rotational speed* *250 rad/s is the speed for high speed simulation* *50 rad/s is the speed for low speed simulation* ***F= Unbalance Force*** *0.5 is the force when using conventional ball bearing*

Simulation 1: Speed 50, Amplitude 50.

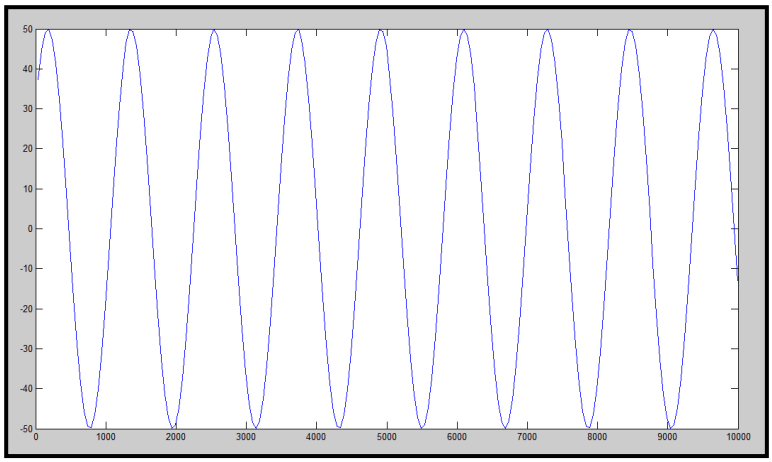


Figure 5.1: Speed versus Amplitude Graph 1

Description and discussion:

- Y- Axis is the amplitude of vibration. X-Axis is time.
- This simulation is considered a slow rotation.
- This is using conventional ball bearing. The amplitude is 50.

Simulation 2: Speed 50, Amplitude 25

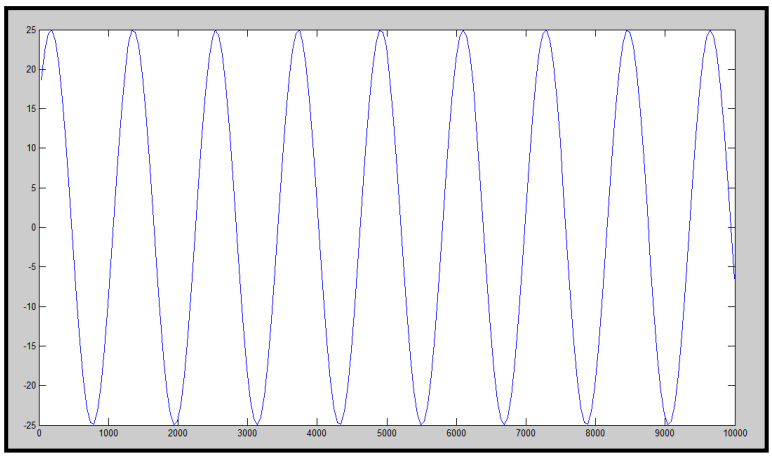


Figure 5.2: Speed versus Amplitude Graph 2

Description and discussion:

- Y- Axis is the amplitude of vibration. X-Axis is time.
- This simulation is considered a slow rotation.
- This is using magnetic bearing. The amplitude is 25.

Simulation 3: Speed 250, Amplitude 50.

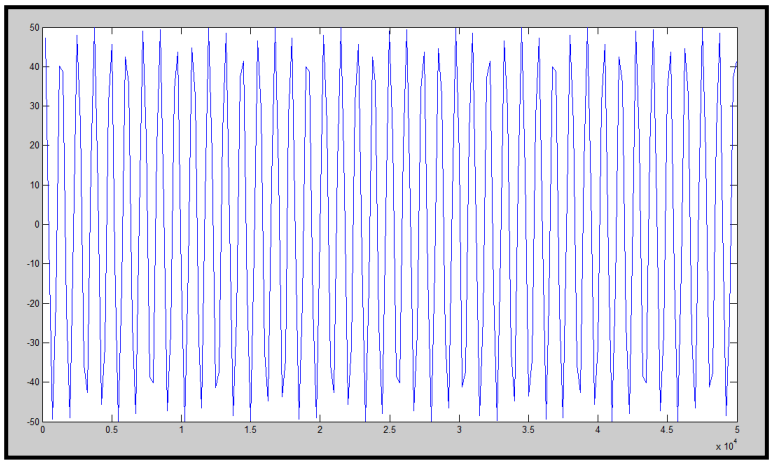


Figure 5.3: Speed versus Amplitude Graph 3

Description and discussion:

- Y- Axis is the amplitude of vibration. X-Axis is time.
- This simulation is using higher speed. The frequency of vibration increases.
- This is using conventional ball bearing. The amplitude is 50.

Simulation 4: Speed 250, Amplitude 25.

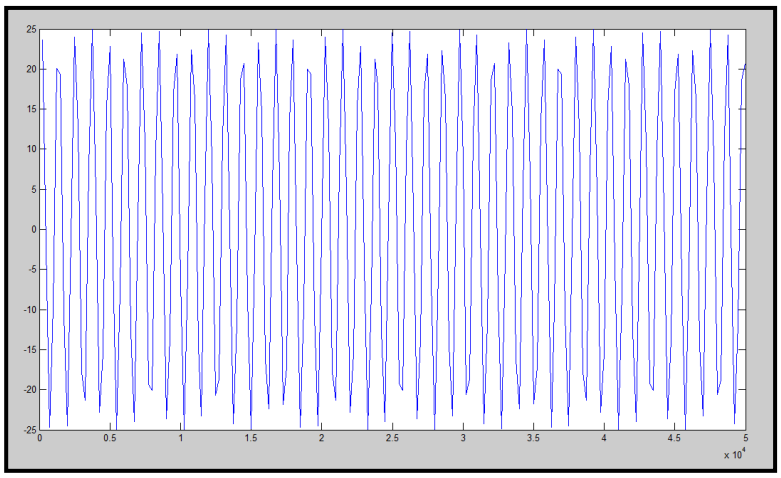


Figure 5.4: Speed versus Amplitude Graph 4

Description and discussion:

- Y- Axis is the amplitude of vibration. X-Axis is time.
- This simulation is using higher speed. The frequency of vibration increases.
- This is using magnetic bearing. The amplitude is 25.

5.2 Simulation using ANSYS

Simulation 1: Damping Ratio $5e^{-4}$

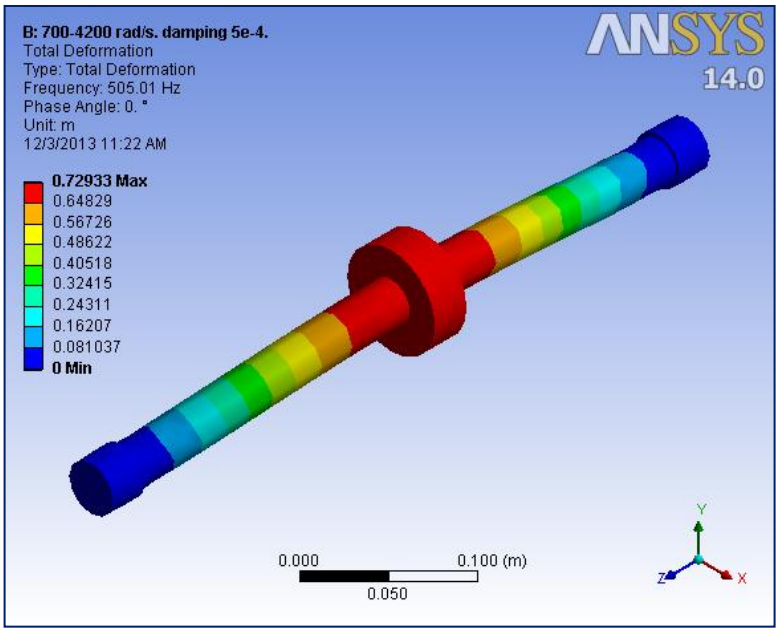


Figure 5.5: Total Deformation Simulation 1

Table 5.1: Damped Frequency and Modal Damping Ratio Simulation 1

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	505.01	-146.2	0.27808
2		2	505.61	-146.47	0.27824
3		3	0	-1138.6	N/A
4		4	924.9	-1358.7	0.82664
5		5	884.83	-1424	0.84938
6		6	898.48	-1446	
7	2	1	504.84	-146.17	0.27811
8		2	505.78	-146.5	0.27821
9		3	0	-1138.6	N/A
10		4	924.9	-1358.7	0.82664
11		5	878.27	-1413	0.8493

12		6	905.6	-1457	0.84931
13	3	1	504.64	-146.12	0.27813
14		2	505.98	-146.55	0.2782
15		3	0	-1138.6	N/A
16		4	924.9	-1358.7	0.82664
17		5	871.9	-1402	0.84918
18		6	912.91	-1468	
19	4	1	504.44	-146.06	0.27813
20		2	506.19	-146.6	0.27819
21		3	0	-1138.6	N/A
22		4	865.72	-1391	0.849
23		5	924.9	-1358.7	0.82664
24		6	920.41	-1478.9	0.84901
25	5	1	504.23	-146.01	0.27814
26		2	506.39	-146.66	0.27819
27		3	0	-1138.6	N/A
28		4	859.73	-1380.1	0.84878
29		5	924.9	-1358.7	0.82664
30		6	928.08	-1489.8	0.84878
31	6	1	504.02	-145.95	0.27814
32		2	506.6	-146.72	0.27818
33		3	0	-1138.6	N/A
34		4	853.91	-1369.2	0.84851
35		5	924.9	-1358.7	0.82664
36		6	935.94	-1500.7	0.84851

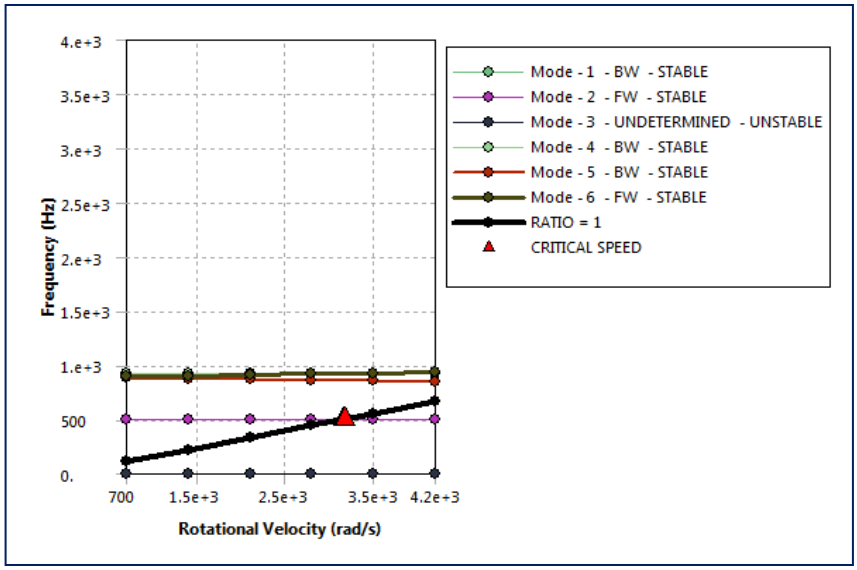


Figure 5.6: Campbell Diagram (Frequency) Simulation 1

Table 5.2: Mode Frequency Simulation 1

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	3168.8 rad/s	505.01 Hz	504.84 Hz	504.64 Hz	504.44 Hz	504.23 Hz	504.02 Hz
2	FW	STABLE	3181.2 rad/s	505.61 Hz	505.78 Hz	505.98 Hz	506.19 Hz	506.39 Hz	506.6 Hz
3	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz	0. Hz	0. Hz	0. Hz	0. Hz
4	BW	STABLE	NONE	924.9 Hz	924.9 Hz	924.9 Hz	924.9 Hz	924.9 Hz	924.9 Hz
5	BW	STABLE	NONE	884.83 Hz	878.27 Hz	871.9 Hz	865.72 Hz	859.73 Hz	853.91 Hz
6	FW	STABLE	NONE	898.48 Hz	905.6 Hz	912.91 Hz	920.41 Hz	928.08 Hz	935.94 Hz

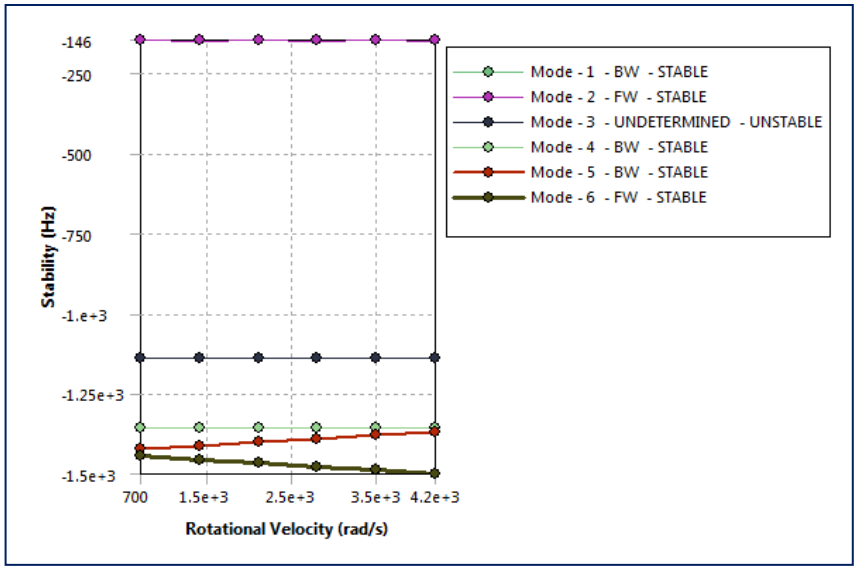


Figure 5.7: Campbell Diagram (Stability) Simulation 1

Table 5.3: Mode Stability Simulation 1

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	NONE	-146.2 Hz	-146.17 Hz	-146.12 Hz	-146.06 Hz	-146.01 Hz	-145.95 Hz
2	FW	STABLE	NONE	-146.47 Hz	-146.5 Hz	-146.55 Hz	-146.6 Hz	-146.66 Hz	-146.72 Hz
3	UNDETERMINED	UNSTABLE	NONE	-1138.6 Hz	-1138.6 Hz	-1138.6 Hz	-1138.6 Hz	-1138.6 Hz	-1138.6 Hz
4	BW	STABLE	NONE	-1358.7 Hz	-1358.7 Hz	-1358.7 Hz	-1358.7 Hz	-1358.7 Hz	-1358.7 Hz
5	BW	STABLE	NONE	-1424. Hz	-1413. Hz	-1402. Hz	-1391. Hz	-1380.1 Hz	-1369.2 Hz
6	FW	STABLE	NONE	-1446. Hz	-1457. Hz	-1468. Hz	-1478.9 Hz	-1489.8 Hz	-1500.7 Hz

Discussion

Setting the damping ratio to $10e^{-4}$, Frequency Campbell diagram and a Stability Campbell Diagram are generated.

From Frequency Campbell Diagram two *critical speeds* are obtained.

i. 3168.8 rad/s
ii. 3181.2 rad/s

Critical speed is that will produce resonance to the system and thus multiply the vibration that occurs. These two speeds are to be avoided from operating at.

From the Stability Campbell Diagram, it is evaluated that *mode* 1, 2, 4, 5 and 6 are stable, however, *mode* 3 in unstable.

Simulation 2: Damping Ratio $5e^{-5}$

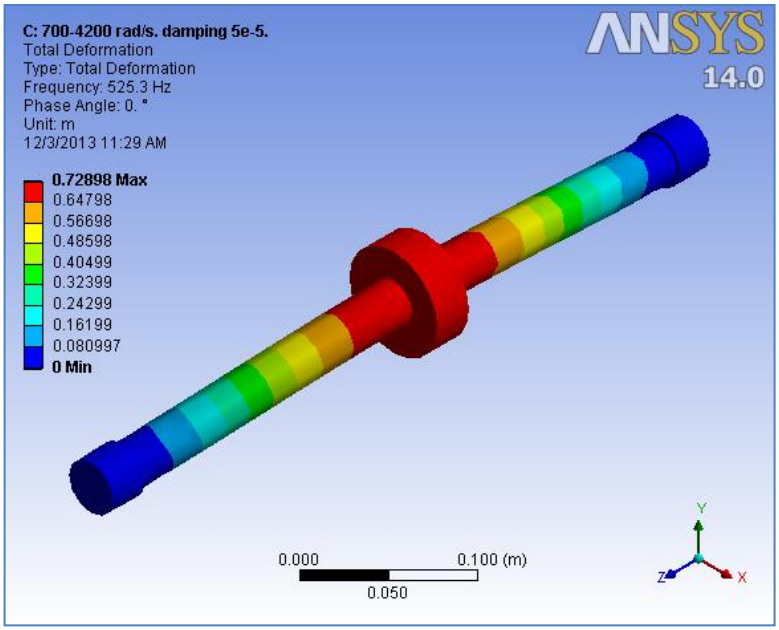


Figure 5.8: Total Deformation Simulation 2

Table 5.4: Damped Frequency and Modal Damping Ratio Simulation 2

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	525.3	-21.78	4.14E-02
2		2	525.94	-21.81	4.14E-02
3		3	1637.4	-143.03	8.70E-02
4		4	1675.8	-150.05	8.92E-02
5		5	1689.5	-151.27	8.92E-02
6		6	3000.5	-469.11	0.15447
7	2	1	525.13	-21.774	4.14E-02
8		2	526.11	-21.817	4.14E-02
9		3	1637.4	-143.03	8.70E-02
10		4	1669	-149.43	8.92E-02
11		5	1696.4	-151.88	
12		6	3000.5	-469.11	0.15447

13	3	1	524.94	-21.766	4.14E-02
14		2	526.3	-21.824	4.14E-02
15		3	1637.4	-143.03	8.70E-02
16		4	1662.3	-148.82	8.92E-02
17		5	1703.3	-152.5	
18		6	3000.5	-469.11	0.15447
19	4	1	524.74	-21.758	4.14E-02
20		2	526.5	-21.833	4.14E-02
21		3	1637.4	-143.03	8.70E-02
22		4	1655.5	-148.21	8.92E-02
23		5	1710.2	-153.11	
24		6	3000.5	-469.11	0.15447
25	5	1	524.53	-21.75	4.14E-02
26		2	526.71	-21.841	4.14E-02
27		3	1637.4	-143.03	8.70E-02
28		4	1648.8	-147.6	8.92E-02
29		5	1717.2	-153.72	8.92E-02
30		6	3000.5	-469.11	0.15447
31	6	1	524.32	-21.741	4.14E-02
32		2	526.92	-21.85	4.14E-02
33		3	1637.3	-143.04	8.70E-02
34		4	1642.2	-146.98	8.91E-02
35		5	1724.2	-154.33	8.92E-02
36		6	3000.5	-469.11	0.15447

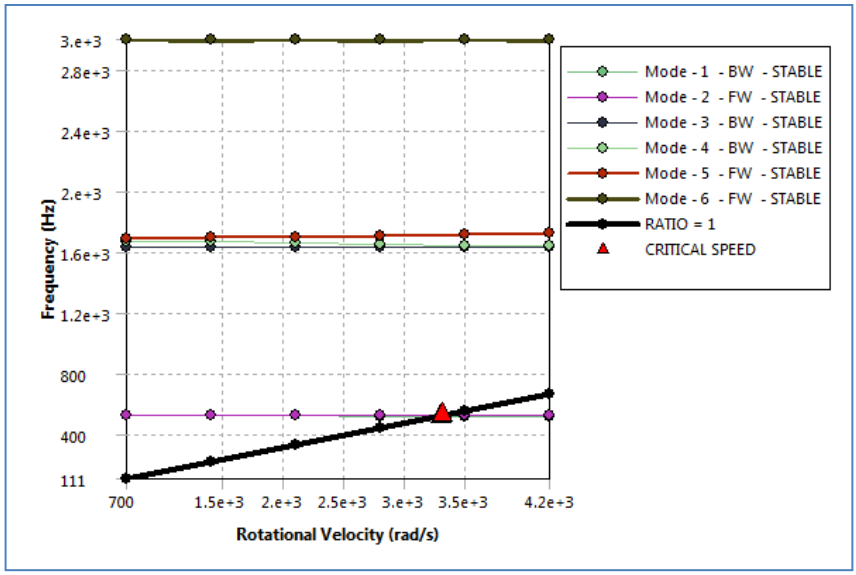


Figure 5.9: Campbell Diagram (Frequency) Simulation 2

Table 5.5: Mode Frequency Simulation 2

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	3296.1 rad/s	525.3 Hz	525.13 Hz	524.94 Hz	524.74 Hz	524.53 Hz	524.32 Hz
2	FW	STABLE	3309.1 rad/s	525.94 Hz	526.11 Hz	526.3 Hz	526.5 Hz	526.71 Hz	526.92 Hz
3	BW	STABLE	NONE	1637.4 Hz	1637.4 Hz	1637.4 Hz	1637.4 Hz	1637.4 Hz	1637.3 Hz
4	BW	STABLE	NONE	1675.8 Hz	1669. Hz	1662.3 Hz	1655.5 Hz	1648.8 Hz	1642.2 Hz
5	FW	STABLE	NONE	1689.5 Hz	1696.4 Hz	1703.3 Hz	1710.2 Hz	1717.2 Hz	1724.2 Hz
6	FW	STABLE	NONE	3000.5 Hz	3000.5 Hz	3000.5 Hz	3000.5 Hz	3000.5 Hz	3000.5 Hz

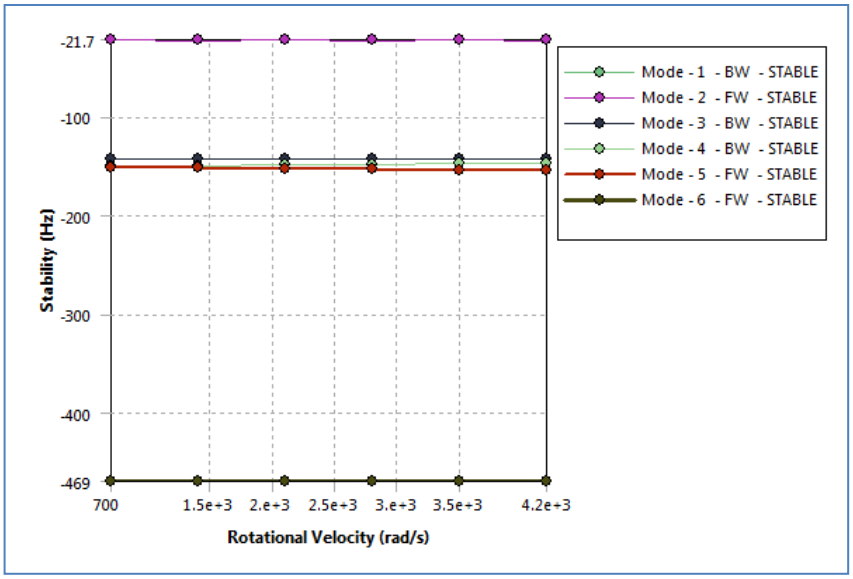


Figure 5.10: Campbell Diagram (Stability) Simulation 2

Table 5.6: Mode Stability Simulation 2

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	NONE	-21.78 Hz	-21.774 Hz	-21.766 Hz	-21.758 Hz	-21.75 Hz	-21.741 Hz
2	FW	STABLE	NONE	-21.81 Hz	-21.817 Hz	-21.824 Hz	-21.833 Hz	-21.841 Hz	-21.85 Hz
3	BW	STABLE	NONE	-143.03 Hz	-143.03 Hz	-143.03 Hz	-143.03 Hz	-143.03 Hz	-143.04 Hz
4	BW	STABLE	NONE	-150.05 Hz	-149.43 Hz	-148.82 Hz	-148.21 Hz	-147.6 Hz	-146.98 Hz
5	FW	STABLE	NONE	-151.27 Hz	-151.88 Hz	-152.5 Hz	-153.11 Hz	-153.72 Hz	-154.33 Hz
6	FW	STABLE	NONE	-469.11 Hz	-469.11 Hz	-469.11 Hz	-469.11 Hz	-469.11 Hz	-469.11 Hz

Discussion

Setting the damping ratio to $10e^{-5}$, Frequency Campbell diagram and a Stability Campbell Diagram are generated.

From Frequency Campbell Diagram two *critical speeds* are obtained.

iii. 3296.1 rad/s
iv. 3309.1 rad/s

Critical speed is that will produce resonance to the system and thus multiply the vibration that occurs. These two speeds are to be avoided from operating at.

From the Stability Campbell Diagram, it is evaluated that all *modes* are stable.

Simulation 3: Damping Ratio $5e^{-6}$

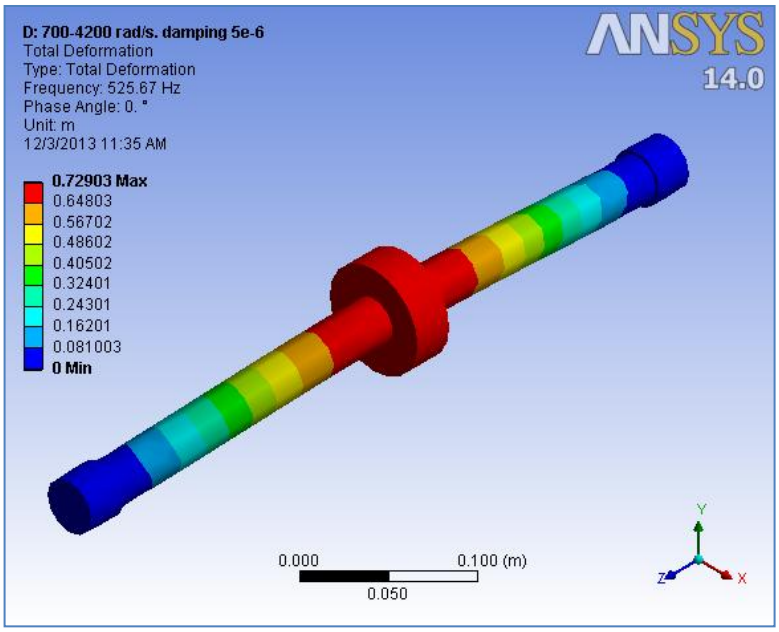


Figure 5.11: Total Deformation Simulation 3

Table 5.7: Damped Frequency and Modal Damping Ratio Simulation 3

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	525.67	-9.3381	1.78E-02
2		2	526.31	-9.345	1.78E-02
3		3	1643.5	-21.465	1.31E-02
4		4	1682.4	-22.138	1.32E-02
5		5	1696.1	-22.318	
6		6	3036.5	-54.073	1.78E-02
7	2	1	525.5	-9.3343	1.78E-02
8		2	526.48	-9.3487	1.78E-02
9		3	1643.5	-21.465	1.31E-02
10		4	1675.6	-22.048	1.32E-02
11		5	1703	-22.408	

12		6	3036.5	-54.073	1.78E-02
13	3	1	525.31	-9.3304	1.78E-02
14		2	526.67	-9.3526	1.78E-02
15		3	1643.5	-21.465	1.31E-02
16		4	1668.9	-21.958	1.32E-02
17		5	1709.9	-22.498	
18		6	3036.5	-54.073	1.78E-02
19	4	1	525.11	-9.3266	1.78E-02
20		2	526.87	-9.3564	1.78E-02
21		3	1643.5	-21.465	1.31E-02
22		4	1662.1	-21.868	1.32E-02
23		5	1716.8	-22.588	
24		6	3036.5	-54.073	1.78E-02
25	5	1	524.9	-9.3228	1.78E-02
26		2	527.08	-9.3602	1.78E-02
27		3	1643.4	-21.465	1.31E-02
28		4	1655.4	-21.778	1.32E-02
29		5	1723.8	-22.678	
30		6	3036.5	-54.073	1.78E-02
31	6	1	524.69	-9.319	1.78E-02
32		2	527.29	-9.364	1.78E-02
33		3	1643.4	-21.465	1.31E-02
34		4	1648.7	-21.688	1.32E-02
35		5	1730.8	-22.768	
36		6	3036.5	-54.073	1.78E-02

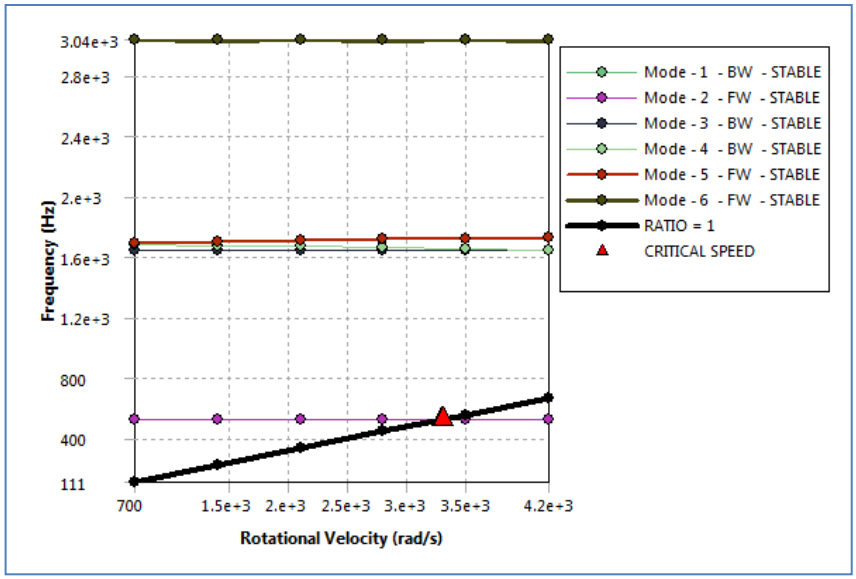


Figure 2.12: Campbell Diagram (Frequency) Simulation 3

Table 5.8: Mode Frequency Simulation 3

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	3298.4 rad/s	525.67 Hz	525.5 Hz	525.31 Hz	525.11 Hz	524.9 Hz	524.69 Hz
2	FW	STABLE	3311.4 rad/s	526.31 Hz	526.48 Hz	526.67 Hz	526.87 Hz	527.08 Hz	527.29 Hz
3	BW	STABLE	NONE	1643.5 Hz	1643.5 Hz	1643.5 Hz	1643.5 Hz	1643.4 Hz	1643.4 Hz
4	BW	STABLE	NONE	1682.4 Hz	1675.6 Hz	1668.9 Hz	1662.1 Hz	1655.4 Hz	1648.7 Hz
5	FW	STABLE	NONE	1696.1 Hz	1703. Hz	1709.9 Hz	1716.8 Hz	1723.8 Hz	1730.8 Hz
6	FW	STABLE	NONE	3036.5 Hz	3036.5 Hz	3036.5 Hz	3036.5 Hz	3036.5 Hz	3036.5 Hz

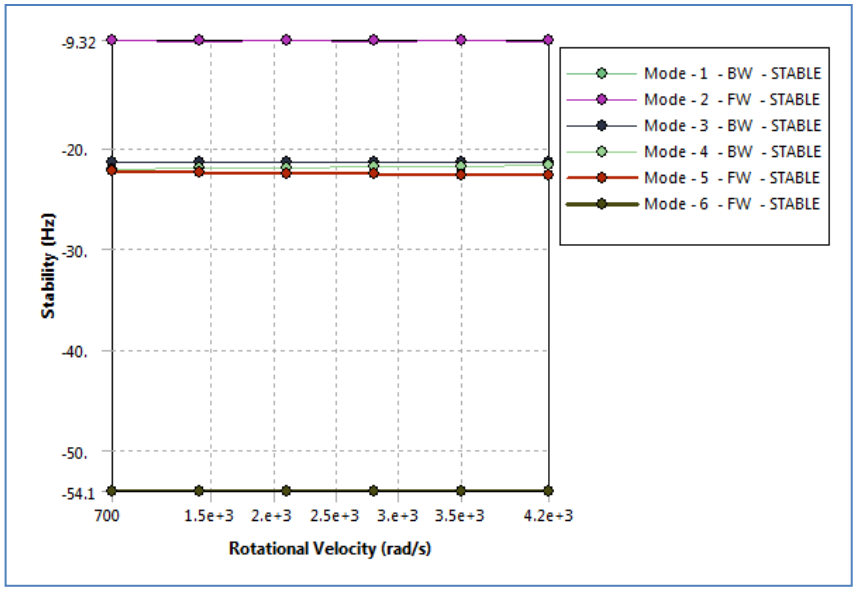


Figure 5.13: Campbell Diagram (Stability) Simulation 3

Table 5.9: Mode Stability Simulation 3

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	NONE	-9.3381 Hz	-9.3343 Hz	-9.3304 Hz	-9.3266 Hz	-9.3228 Hz	-9.319 Hz
2	FW	STABLE	NONE	-9.345 Hz	-9.3487 Hz	-9.3526 Hz	-9.3564 Hz	-9.3602 Hz	-9.364 Hz
3	BW	STABLE	NONE	-21.465 Hz	-21.465 Hz	-21.465 Hz	-21.465 Hz	-21.465 Hz	-21.465 Hz
4	BW	STABLE	NONE	-22.138 Hz	-22.048 Hz	-21.958 Hz	-21.868 Hz	-21.778 Hz	-21.688 Hz
5	FW	STABLE	NONE	-22.318 Hz	-22.408 Hz	-22.498 Hz	-22.588 Hz	-22.678 Hz	-22.768 Hz
6	FW	STABLE	NONE	-54.073 Hz	-54.073 Hz	-54.073 Hz	-54.073 Hz	-54.073 Hz	-54.073 Hz

Discussion

Setting the damping ratio to $10e^{-6}$, Frequency Campbell diagram and a Stability Campbell Diagram are generated.

From Frequency Campbell Diagram two *critical speeds* are obtained.

v. 3298.4 rad/s
vi. 3311.4 rad/s

Critical speed is that will produce resonance to the system and thus multiply the vibration that occurs. These two speeds are to be avoided from operating at.

From the Stability Campbell Diagram, it is evaluated that all *modes* are stable.

Simulation 4: Damping Ratio $5e^{-7}$

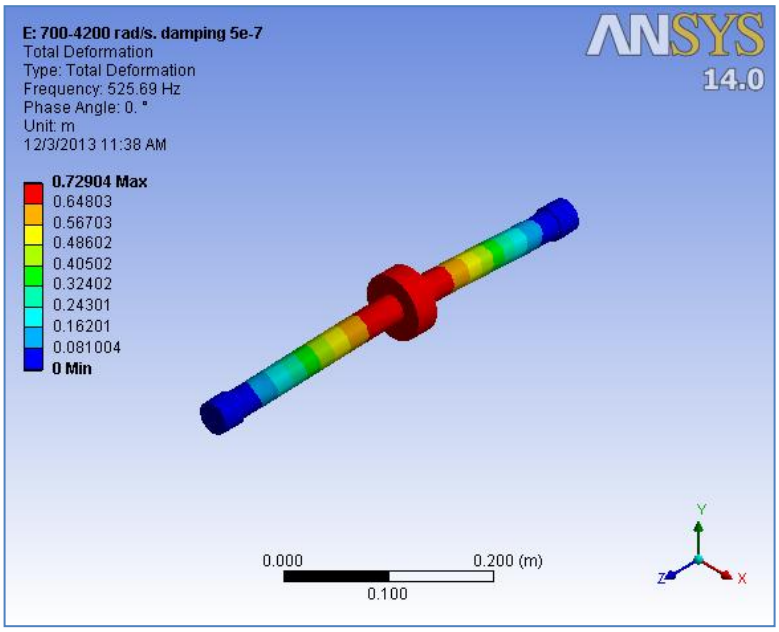


Figure 5.14: Total Deformation Simulation 3

Table 5.10: Damped Frequency and Modal Damping Ratio Simulation 3

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio
1	1	1	525.69	-8.0938	1.54E-02
2		2	526.33	-8.0984	1.54E-02
3		3	1643.6	-9.3085	5.66E-03
4		4	1682.5	-9.3468	5.56E-03
5		5	1696.2	-9.4227	
6		6	3036.9	-12.569	4.14E-03
7	2	1	525.52	-8.0903	1.54E-02
8		2	526.5	-8.1019	1.54E-02
9		3	1643.6	-9.3084	5.66E-03
10		4	1675.7	-9.3088	5.56E-03

11		5	1703.1	-9.4607	5.55E-03
12		6	3036.9	-12.569	4.14E-03
13	3	1	525.33	-8.0869	1.54E-02
14		2	526.69	-8.1054	1.54E-02
15		3	1643.6	-9.3084	5.66E-03
16		4	1669	-9.2709	5.55E-03
17		5	1710	-9.4987	
18		6	3036.9	-12.569	4.14E-03
19	4	1	525.13	-8.0835	1.54E-02
20		2	526.89	-8.1088	1.54E-02
21		3	1643.6	-9.3084	5.66E-03
22		4	1662.2	-9.2329	5.55E-03
23		5	1716.9	-9.5367	
24		6	3036.9	-12.569	4.14E-03
25	5	1	524.92	-8.0801	1.54E-02
26		2	527.1	-8.1121	1.54E-02
27		3	1643.6	-9.3083	5.66E-03
28		4	1655.5	-9.1951	5.55E-03
29		5	1723.9	-9.5746	5.55E-03
30		6	3036.9	-12.569	4.14E-03
31	6	1	524.71	-8.0768	1.54E-02
32		2	527.31	-8.1155	1.54E-02
33		3	1643.5	-9.3076	5.66E-03
34		4	1648.9	-9.1578	5.55E-03
35		5	1730.9	-9.6126	5.55E-03
36		6	3036.9	-12.569	4.14E-03

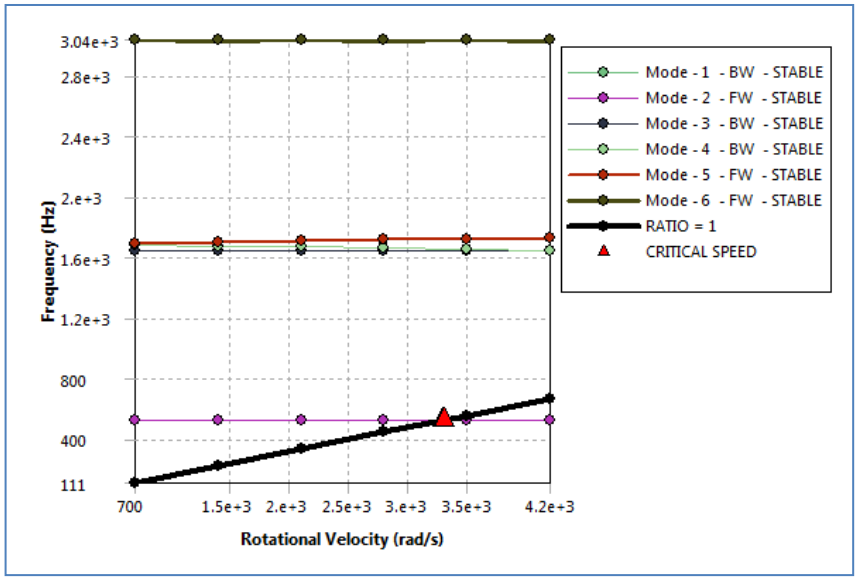


Figure 5.15: Campbell Diagram (Frequency) Simulation 4

Table 5.11: Mode Frequency Simulation 3

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	3298.5 rad/s	525.69 Hz	525.52 Hz	525.33 Hz	525.13 Hz	524.92 Hz	524.71 Hz
2	FW	STABLE	3311.5 rad/s	526.33 Hz	526.5 Hz	526.69 Hz	526.89 Hz	527.1 Hz	527.31 Hz
3	BW	STABLE	NONE	1643.6 Hz	1643.6 Hz	1643.6 Hz	1643.6 Hz	1643.6 Hz	1643.5 Hz
4	BW	STABLE	NONE	1682.5 Hz	1675.7 Hz	1669. Hz	1662.2 Hz	1655.5 Hz	1648.9 Hz
5	FW	STABLE	NONE	1696.2 Hz	1703.1 Hz	1710. Hz	1716.9 Hz	1723.9 Hz	1730.9 Hz
6	FW	STABLE	NONE	3036.9 Hz	3036.9 Hz	3036.9 Hz	3036.9 Hz	3036.9 Hz	3036.9 Hz

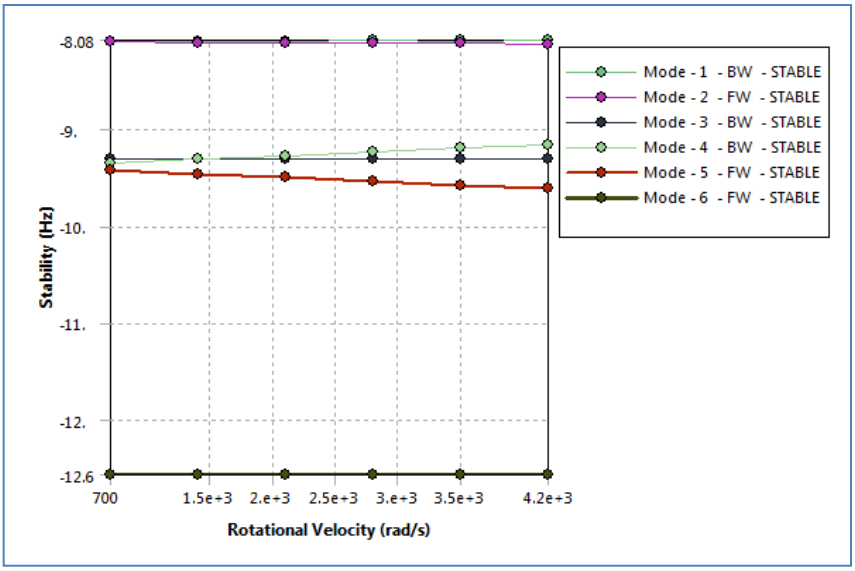


Figure 5.16: Campbell Diagram (Stability) Simulation 4

Table 5.12: Mode Stability Simulation 4

Mode	Whirl Direction	Mode Stability	Critical Speed	700. rad/s	1400. rad/s	2100. rad/s	2800. rad/s	3500. rad/s	4200. rad/s
1	BW	STABLE	NONE	-8.0938 Hz	-8.0903 Hz	-8.0869 Hz	-8.0835 Hz	-8.0801 Hz	-8.0768 Hz
2	FW	STABLE	NONE	-8.0984 Hz	-8.1019 Hz	-8.1054 Hz	-8.1088 Hz	-8.1121 Hz	-8.1155 Hz
3	BW	STABLE	NONE	-9.3085 Hz	-9.3084 Hz	-9.3084 Hz	-9.3084 Hz	-9.3083 Hz	-9.3076 Hz
4	BW	STABLE	NONE	-9.3468 Hz	-9.3088 Hz	-9.2709 Hz	-9.2329 Hz	-9.1951 Hz	-9.1578 Hz
5	FW	STABLE	NONE	-9.4227 Hz	-9.4607 Hz	-9.4987 Hz	-9.5367 Hz	-9.5746 Hz	-9.6126 Hz
6	FW	STABLE	NONE	-12.569 Hz	-12.569 Hz	-12.569 Hz	-12.569 Hz	-12.569 Hz	-12.569 Hz

Discussion

Setting the damping ratio to $10e^{-5}$, Frequency Campbell diagram and a Stability Campbell Diagram are generated.

From Frequency Campbell Diagram two *critical speeds* are obtained.

vii. 3298.5 rad/s
viii. 3311.5 rad/s

Critical speed is that will produce resonance to the system and thus multiply the vibration that occurs. These two speeds are to be avoided from operating at.

From the Stability Campbell Diagram, it is evaluated that all *modes* are stable.

Summary of Results

Table 5.13: Summary of Results

Damping ratio	Critical Speed	Mode Stability
$5e^{-4}$	i. 3168.8 rad/s ii. 3181.2 rad/s	*Mode* 3 unstable
$5e^{-5}$	i. 3296.1 rad/s ii. 3309.1 rad/s	All *modes* are stable
$5e^{-6}$	i. 3298.4 rad/s ii. 3311.4 rad/s	All *modes* are stable
$5e^{-7}$	i. 3298.5 rad/s ii. 3311.5 rad/s	All *modes* are stable

CHAPTER 6

CONCLUSION

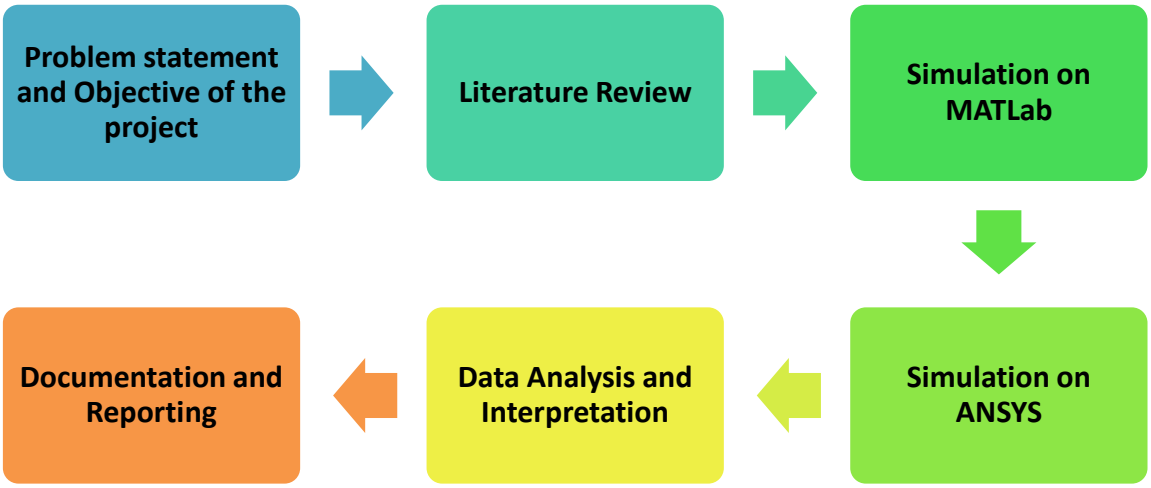
ANSYS software is capable of executing simulation to evaluate vibration of a system. The parameter which is manipulated is the damping ratio which becomes the factor to the vibration and stability of the shaft operation.

Critical speed is the speed where resonance will occur and multiply the vibration. The shaft should be avoided from operating at those speeds. Critical speed is detected on every simulation. The highest critical speed is in simulation 4 where damping ratio is $5e^{-7}$.

The next result of interest is the stability of operation. The negative values indicate that the system is stable. As far, all operation is stable except mode 3 in Simulation 1 with damping ratio of $5e^{-4}$.

CHAPTER 7

KEY MILESTONE



CHAPTER 8

GANTT CHART

DETAIL/WEEK	1	2	3	4	5	6	7		8	9	10	11	12	13	14
First meeting with coordinator and supervisor								Mid-Semester break							
Preliminary research work															
Submission of Extended Proposal Defence															
Proposal Defence															
Project work continues															
Submission of Interim Draft Report															
Submission of Interim Report															

Figure 8.17: Gantt Chart of Final Year Project I

DETAIL/WEEK	1	2	3	4	5	6	7		8	9	10	11	12	13	14	15
Project Work Continue								Mid-Semester break								
Submission of Progress Report																
Project Work Continue																
Pre-SEDEX																
Submission of Draft Report																
Submission of Dissertation (soft bound)																
Submission of Technical Paper																
Oral Presentation																
Submission of Project Dissertation (Hard Bound)																

Figure 8.18: Gantt Chart of Final Year Project I

CHAPTER 9

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