

ROTOR DYNAMIC ANALYSIS OF CENTRIFUGAL COMPRESSOR DUE TO LIQUID CARRY OVER

By

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CERTIFICATION OF APPROVAL

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

Approved by,

.....

AP. Dr. Tadimalla V.V.L.N. Rao

CERITIFICATION

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 the original work contained here have not been undertaken or done by unspecified sources or persons.

.....
AIDA KHALEEDA MOHD PAUZI

ABSTRACT

This paper shows a study of rotor dynamic response of centrifugal compressor due to liquid carry over. The rotor system is supported by supporting structure bearings with different stiffness. This case study shows the rotor dynamic analysis of a Jeffcott rotor. From the study of rotor dynamic on Jeffcott rotor, the results are as follows; Campbell Diagram, critical speed, mode stability and whirl orbits. By using different value stiffness bearings, the results will be further discussing in results and discussion. The simulation of rotating rotor will be executed in ANSYS Mechanical APDL. With the simulation, the research can achieve the objectives.

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CHAPTER 1

INTRODUCTION

1.1 Background of the project

In an oil and gas industry development is very much related to financial, economical, management or technical challenges. This will effect in many ways especially in financial. When it involved environmental legislation as the extreme working conditions such as high in viscosity reservoirs, it needs immediate technical solution. This is important especially for current and future turbomachinery in the industry. Other than extreme working conditions as mentioned above, compressor operability range can cause the machines to rotor dynamic problems. Compressed gas such as Hydrogen Sulphide (H_2S) is one aggressive nature gas that is related to the issues. The effect of wet gas on rotor dynamics characteristic of a high-speed centrifugal compressor has been researched [1]. It is defined that ‘wet gas’ is as the inclusion of liquid in gas was both are in two phase medium and in different ratio in several factors. Moreover, it can conclude that the presence of ‘wet gas’ had a straight effect on thermodynamic performance. [2]

The main sources of rotor vibration within are forced response and self-induced vibrations. The cost of downtime of gas lift/re-injection is very high and can cause downtime of high rotor dynamic vibration levels.

Fouling is a function of gas cleanness that plays an important role in oil and gas operations that depends on the formation of reservoir. It can also effect the performance of the vessels and scrubbers that is used to clean the processed gas that entered to the compressor. While destabilizing cross coupling forces that is generated by the impellers and seals are one of the function of a rotor speed, different kind of density and pressure of gas as the impellers across the seals span. [3][4]

1.2 Problem Statement

The rotor dynamic analysis of the compressor unit shows the presence of significant unbalance response as a result of liquid carry over. The critical vibration can effect the operations. Rotating machinery produces vibrations depending upon the structure of the mechanism involved in the process. This paper is to study an experienced significant level of vibration that made its continued operation impossible. The rotor dynamic analysis unit shows the expected result and investigation of a rotor dynamic analysis

1.3 Objectives and Scope of Study

Objectives

The main objectives of this project are:

- i) To study Rotor Dynamic Analysis of a Jeffcott Rotor
- ii) To determine the critical speed and vibration of Jeffcott Rotor

Scope of study

This study focuses on the how to calculate the results of the rotor dynamic analysis using different solution and methods. Throughout the study, the author will be exposing to ANSYS which will be used to find the difference in rotor vibrations synchronous response. The author need to learn and expert himself on ANSYS software in order to carry out simulation and analysis of the study. By using ANSYS, the author will gain the rotor vibration in different molecular weight of gasses and will be recorded.

CHAPTER 2

LITERATURE REVIEW

2.1 Rotor dynamics

Rotating machines are the key equipment that is used diverse engineering fields. The stability of bearing-rotor system will result a strong vibration and even disastrous accident of machinery. [5] The meaning of rotor dynamics is the branch of engineering that studies the lateral and torsional vibrations of rotating shafts [6]. The main objective of rotor dynamics are estimating the rotor vibrations and containing the vibration level under an acceptable limit. The shaft or rotor with disk, bearings and the seals are the main and principal components of a rotor dynamic system. [7] The bearings are to support the rotating components and provide the additional damping needed to stabilize the system in rotor dynamics. The seals are to prevent and overcome undesired leakage flows inside the machines of the processing or lubricate fluids. A stator is to support the structure.

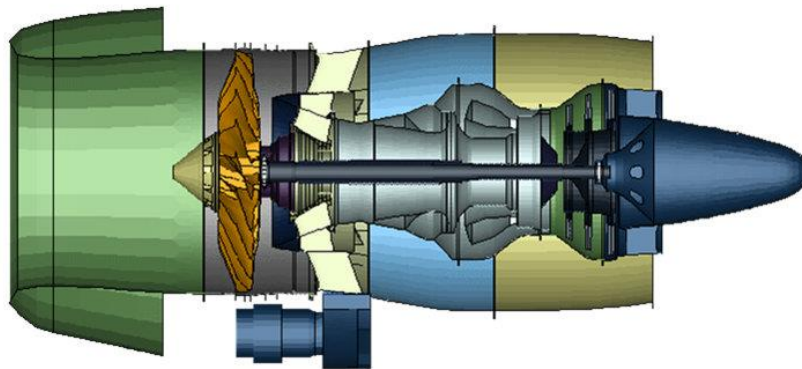


Figure 1: Rotor dynamics in simulation

When the speed increase of rotation increases, the amplitude of vibration often passes through a maximum. On the other hand, it also contains the rotor vibrations. That is when the

critical speed occurred. This is because the amplitude is to excite by unbalance of rotating equipment or structure such as engine balance. A catastrophic failure can occur when the amplitude of vibration at critical speed is excess. Moreover, the rotating machinery can produce vibrations depends on the structure mechanism that are involved in the whole process. If there is any mistake or faults during the process, the vibration signatures will excite or increase. When designing rotating machinery, the main aspects that should be considered is the vibration behaviour of the machine due to imbalance. When a rotational speeds matches with natural frequency, the critical speed of rotating machine can be occurred. First critical speed happened when the lowest speed which is natural frequency is first considered. However, when the speed increases, it can be an additional critical speed to the structure. Therefore, by using reducing rotational unbalance and the unwanted external forces plays an important role to reduce the all the forces which can initiate resonance. Other than considering vibration behaviour when designing rotational machine, we should concern when the vibration is in resonance. It can create destructive energy and this should be very well considered. This is to avoid the operations that are nearly to critical and pass safely through the rotating when in acceleration or deceleration. Is this factor is ignored and not encountered, it will result in excessive wear, tear on the machine or human injury or even worse loss of lives.

The dynamics of the machines is very complex and difficult to model theoretically. For modelling and analysis of the machine for natural frequencies, there are equations and method that should be considered. For example, structural components such as parameters models, Rayleigh-Ritz method and finite element method (FEM)

2.2 Simple Models of Rotors

Equilibrium position, O is the instantaneous position of the disturbed rotor centreline; S. G is the position of mass center of the rotor. The instantaneous angle between SG and O_x axis is ϕ . The instantaneous angle between line OS and O_x axis is $(\phi - \alpha)$. The distance of O and S is calculated during analysis, is the amplitude of whirl of the rotor. The center of mass of the rotor moves in u_G and v_G in x and y directions respectively.

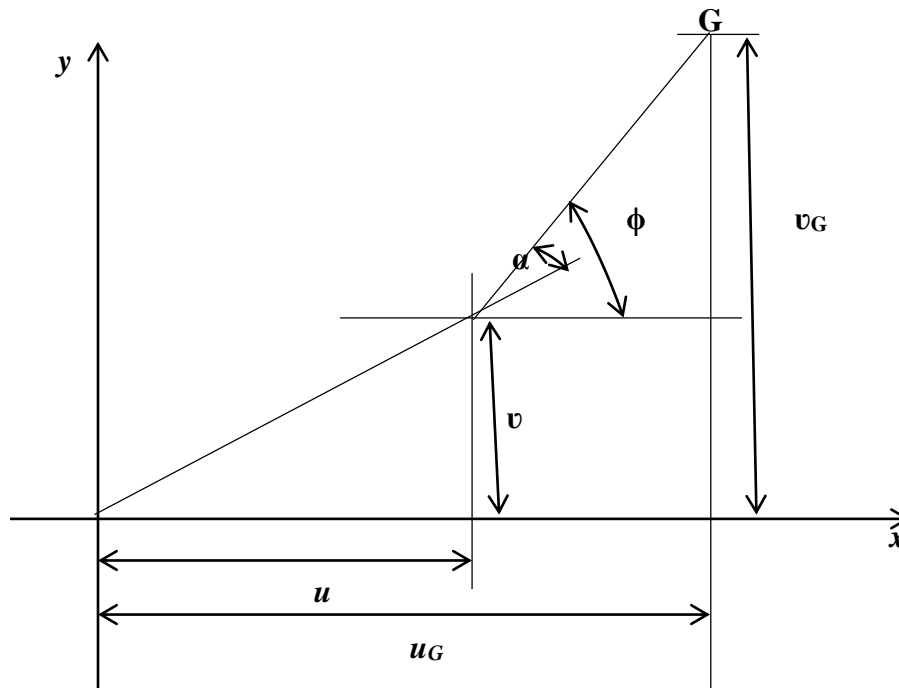


Figure 2: Instantaneous position of bearing centreline S and rotor mass center G

v - Velocity in y direction

u - Velocity in x direction

$$u_G = u + \epsilon \cos \phi$$

Equation 2.2.1

$$v_G = v + \epsilon \sin \phi$$

Equation 2.2.2

From equation 3.1 and 3.2, by differentiating these equations twice with respect to time and noting that ε is constant gives the equation 3.3 and 3.4 below

$$\ddot{u}_G = \ddot{u} + \varepsilon (-\dot{\phi}^2 \cos \phi - \phi \sin \dot{\phi}) \quad \text{Equation 2.2.3}$$

$$\ddot{v}_G = \ddot{v} + \varepsilon (-\dot{\phi}^2 \sin \phi + \phi \cos \dot{\phi}) \quad \text{Equation 2.2.4}$$

By deriving Equation 3.3 and 3.4, it has not restricted the analysis to the case of rotor spinning with a constant angular velocity. With a constant speed of rotation, Ω , $\dot{\phi} = \Omega$ and $\ddot{\phi} = 0$. Therefore,

$$\ddot{u}_G = \ddot{u} - \varepsilon \Omega^2 \cos \Omega t \quad \text{Equation 2.2.5}$$

$$\ddot{v}_G = \ddot{v} - \varepsilon \Omega^2 \sin \Omega t \quad \text{Equation 2.2.6}$$

For the rotor to be considered, the center of mass is offset from the shaft centreline at equilibrium by a small quantity ε , and the displacement of the center of mass is given by u_G and v_G . However, the displacement of the springs and dampers (at the bearings) are still in terms of u and v . Thus, by replacing \ddot{u} by \ddot{u}_G and \ddot{v} by \ddot{v}_G .

$$m\ddot{u}_G + k_{xT}u + k_{xC}\psi = 0 \quad \text{Equation 2.2.7}$$

$$m\ddot{v}_G + k_{xT}v + k_{xC}\theta = 0 \quad \text{Equation 2.2.8}$$

By substituting for \ddot{u}_G and \ddot{v}_G from Equation (3) and rearranging gives

$$m\ddot{u}_G + k_{xT}u + k_{xC}\psi = m\varepsilon\Omega^2 \cos \Omega t \quad \text{Equation 2.2.9}$$

$$m\ddot{v}_G + k_{xT}v + k_{xC}\theta = m\varepsilon\Omega^2 \sin \Omega t \quad \text{Equation 2.2.10}$$

Based on Equation 3.9 and 4.0, it shows that the lateral offset of the mass center from the equilibrium position caused out-of-balance forces to act on the system. Therefore, we can get equations of motion of system with disk or rotor. [12]

2.3 Jeffcott Rotor

Jeffcott rotor model also known as LaVal rotor consist of rigid disc of mass in the center of shaft. It supported by static supporting structure or bearings. Figure 2.3.1 shows a Jeffcott Rotor Model.

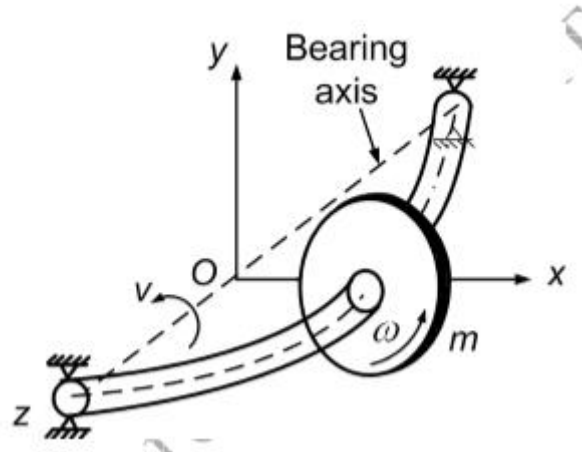


Figure 3: Jeffcott Rotor

C = Disc at center of rotation

G = Center of Gravity

ω = Speed of the shaft

v = Shaft whirls (Bearings axis with whirl frequency)

There are two types of whirled motion which are asynchronous and anti-synchronous. When a rub occur between rotor and stator is called anti-synchronous whirl. While asynchronous is when the speed increase. This also can happen because when the rotor is asymmetric or dynamic properties of bearings are directionally dependent [17]. Figure 2.3.2 and Figure 2.3.3 shows asynchronous and anti-synchronous whirl.

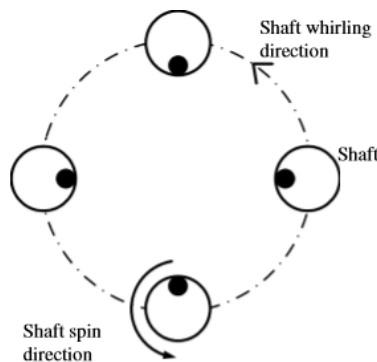


Figure 4: Asynchronous Whirl (R.Tiwari, 2012)

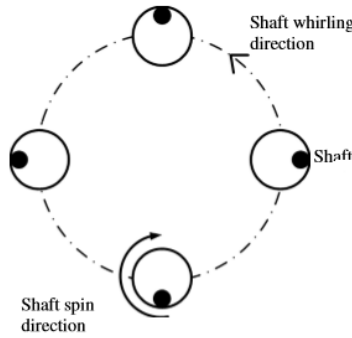


Figure 5: Anti-synchronous Whirl (R.Tiwari, 2012)

2.4 Campbell Diagram

In understanding the dynamic behaviour of the rotating machines, the Campbell diagram is one of the most important tools in rotor dynamic analysis. It is basically consisting of a plot of the natural frequencies of the system as functions of the spin speed. Even though based on complete linearity, the Campbell diagram of the linearized model can yield many important information concerning a nonlinear rotating system. [18]

[19] There are two types of waves in Campbell diagram. For example, forward waves and backward waves. Forward waves occur when the rotation speed is added to the propagation speed of the travelling wave. Backward waves are 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. Figure 2.4.1 is one the example of Campbell Diagram.

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

$$f_f = f_s + i \frac{\Omega}{60} \quad \text{Equation 2.41}$$

$$f_b = f_s - i \frac{\Omega}{60} \quad \text{Equation 2.42}$$

$$f_s^2 = f^2 + K \left(\frac{\Omega}{60} \right)^2 \quad \text{Equation 2.43}$$

f_f = Frequencies forward

f_b = Frequency backward

i = Number of nodal diameter

Ω = Speed (rpm)

K = Dimensionless value

f = Natural frequency

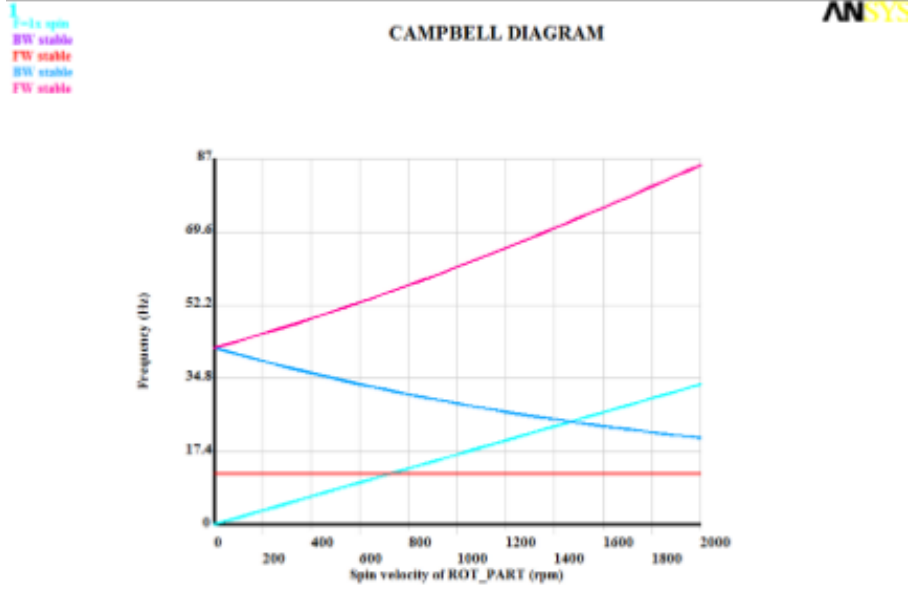
The critical speed is mainly to generate the whirl speed map. This include excitation frequency lines of interest, and graphically note the intersections to obtain the critical speeds associated with each excitation. Example of critical speed can be seen in table 2.4.1. The whirl frequency and damping ratio versus the rotor speed are commonly used to plot both whirl speed. The diagrams are plot for a finite of rotations to get the accurate Campbell diagram.

$$\xi_i = -\frac{\alpha}{\sqrt{\alpha + \omega}} = -\frac{\alpha}{\omega} \quad \text{Equation 2.44}$$

ξ_i = Damping ratio

α = Damping Constant

ω = Damped natural frequency of whirl speed



Laval-Jeffcott: Critical speed and campbell analysis

Figure 6: Campbell Diagram (Srikrishnanivas,2012)

Critical Speed, rpm (Slope of excitation line = 1)				
Mode No	Ansys	Dyrobes	Ratio	Whirl
1	727.29	728.00	0.9990	BW
2	727.33	728.00	0.9991	FW
3	1467.23	1469.00	0.9988	BW

Table 1: Critical Speed (Srikrishnanivas,2012)

2.5 Whirling

When a rotor is in a motion or movement, the rotor will eventually curve or bend. It will follow in a circular or elliptical motion. This happened because a whirling has occurred. Whirling is a rotor undergoes rotation because of a centrifugal force acting on it [16]. There are two types of whirling which are backward whirling and forward whirling. [14] The Figure 2.51 and Figure 2.52 shown below shows both types of whirling.

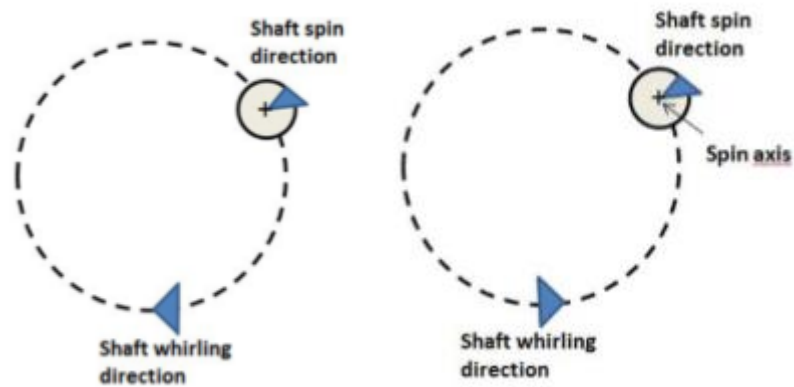


Figure 7: Forward Whirling

Figure 8: Backward Whirling

A forward whirl is when the direction of the whirl undergoes the same direction of the shaft. Whereas a backward whirl is the goes the opposite of the direction of the shaft. One of the reason for backward whirl are because of directionally dependent of the bearings [15]. The natural frequency of whirling both whirling is called natural whirling frequency.

Whirl Orbit

A whirl orbit occurs when nodes or points located at the shaft axis and spin in a curve path during rotation. Figure 2.5.3 shows an example of whirl orbit in ANSYS Mechanical APDL. Moreover, it can be in circular or elliptical form.

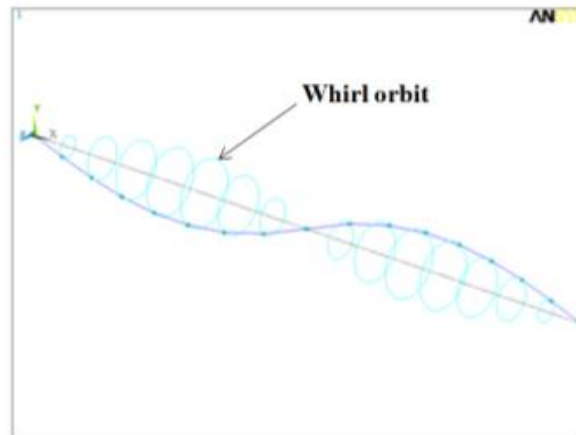


Figure 9: Whirl Orbit (Srikrishnanivas, 2012)

The circular whirl orbit can be in horizontal or vertical direction. This only happens when both bearings or supporting structure have the same stiffness. Whereas ellipse whirl in horizontal and vertical direction occurs when bearings or supporting structure have different values of stiffness. [16].

CHAPTER 3

METHODOLOGY

3.1 Project Flow

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

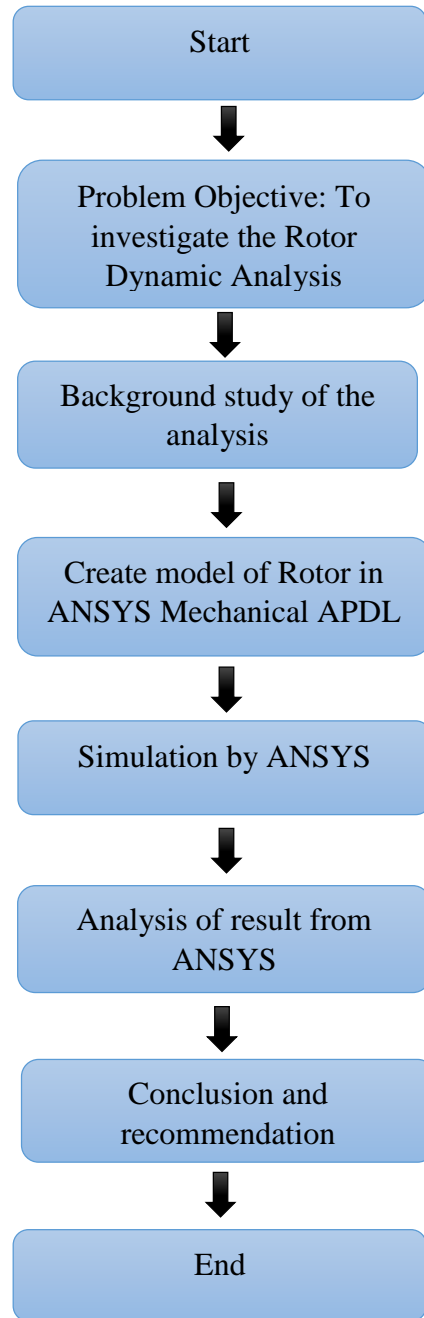
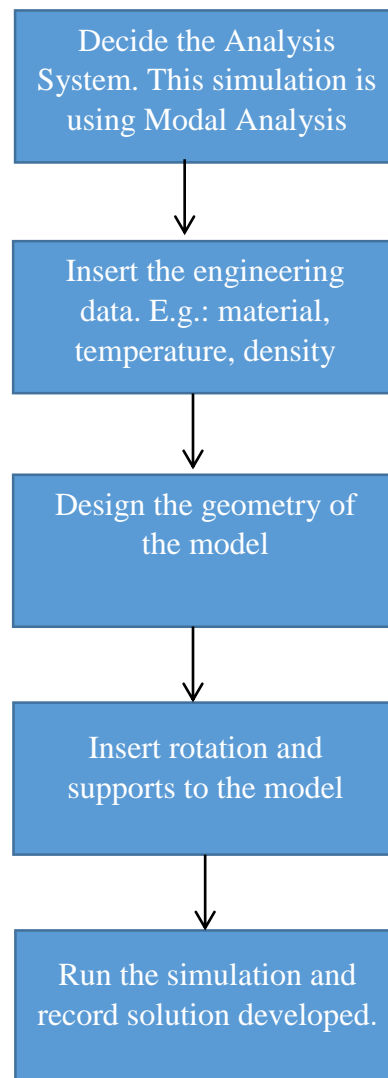


Figure 10: Project Flow

3.2 Simulation using ANSYS Software

ANSYS workbench will be used in this project for simulation. ANSYS Workbench is the latest version of the software. It combines the strength of the core product solvers with the project management tools necessary to manage the project workflow. It also can analyses is built as systems, which can be a project. There are 5 steps of basic procedure of ANSYS Software. The figure below showed the basic procedure of ANSYS

Figure 11: Basic Procedure of ANSYS



3.3 Jeffcott Rotor Model Geometry

The Jeffcott rotor was design with one disk and two bearings installed at the end of the shaft. The model was based on the study of Srikrishnanivas (2012).

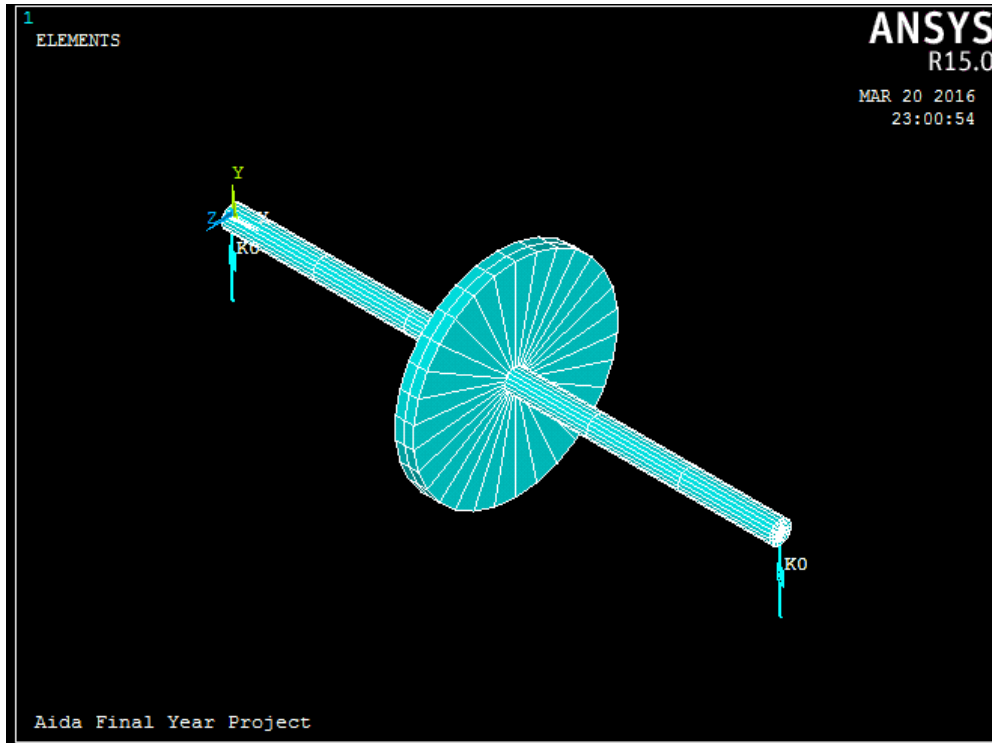


Figure 12: Jeffcott Rotor Model

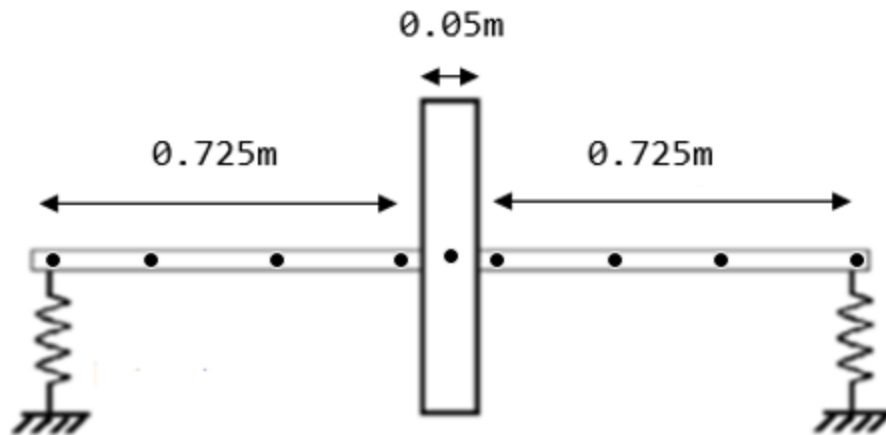


Figure 13: Side view of Jeffcott Rotor

The properties of the Jeffcott Rotor are as below.

Components	Parameter	Value
Disk	Thickness	0.05m
	Outer Radius	0.28m
	Inner Radius	0.03m
Shaft	Length	1.50m
	Outer Radius	0.03m
	Inner Radius	0.00m
	Poisson's ratio, ν	0.03m
	Density of the shaft, ρ	7800 Kg/m ³
	Young's Modulus, E	2e+11
Bearings	Spring Constant, cxx	12.81e+06
	Spring Constant, cyy	8.81e+06

Table 2: Properties of Jeffcott Rotor Model

The Jeffcott are design using variables of commands. The commands as below are used in designing a Jeffcott Rotor model. N commands are used to determine the nodes at the rotor. The design is used by performing commands below:

Nodes

N,1,0

N,2,0.25

N,3,0.5

N,8,0.725

N,4,0.75

N,9,0.775

N,5,1

N,6,1.25

N,7,1.5

N,10,0,-0.2

N,20,1.5,-0.2

Element Properties

```
ET,1,BEAM188,,2
SECTYPE,1,BEAM,CSOLID
SECDATA,0.03,20
SECTYPE,2,BEAM,CTUBE
SECDATA,0.03,0.28,30
TYPE,1
MAT,1
SECNUM,1
*DO,I,1,6
E,I,I+1
*ENDDO
TYPE,1
SECNUM,2
FLST,2,2,1
FITEM,2,8
FITEM,2,4
FLST,2,2,1
FITEM,2,9
FITEM,2,4
E,P51X
ET,2,MASS21
R,2,1.401,1.401,1.401,0.002,0.00136,0.00136
TYPE,2
REAL,2
E,4
ET,3,COMBI214
KEYOPT,3,2,1
KEYOPT,3,3,1
R,3,12.81e6,8.815e6,16.39e6,-25.06e6,232.9e3,294.9e3,
,kxx,kyy,kxy,kyx,cxx,cyy,
```

```

RMORE, -81.92e3, -81.92e3, ! rmore, cxy,cyx,
TYPE,3
REAL,3
FLST,2,2,1
FITEM,2,1
FITEM,2,10
E,P51X
FLST,2,2,1
FITEM,2,7
FITEM,2,20
E,P51X
ALLSEL,ALL

```

3.4 Campbell Diagram and Critical Speeds

The Campbell diagram are plot by using the Command that are used in Mechanical APDL. To determine the frequency of the rotating system, it its carried out in Campbell diagram using different value of speed. When the model undergoes rotational speed, the analysis using CORIOLIS command takes place. To solve Jeffcott modal analysis with gyroscopic effect, QRDAMP are used in this software. In this case, the speed that are used is 100, 2000, 4000, 4500 and 9000 (RPM). The analysis are used by performing using commands below.

```

/Solu
RATIO = 4*ATAN(1)/30
ANTYPE,MODAL
NBF = 11 ! NUMBER OF MODES
CORIOLIS,ON,,ON ! CORIOLIS ACTIVE FOR STATIONARY FRAME
MODOPT,QRDAMP,NBF,,ON
*DO,I,1,5
OMEGA,SPEED(I)*RATIO ! ROTATIONAL VELOCITY ABOUT X-DIRECTION
MXPAND,NBF
SOLVE
*ENDDO

```

```
FINI
!antype,harmic
/POST1
PRCAMP,,1,RPM
*GET,F1,CAMP,1,FREQ,5
*GET,F2,CAMP,2,FREQ,5
*GET,F3,CAMP,3,FREQ,5
*GET,F4,CAMP,4,FREQ,5
*DIM,LABEL,CHAR,1,4
*DIM,VALUE,ARRAY,4,3
LABEL(1,1) = 'WHIRL BW'
LABEL(1,2) = 'WHIRL FW'
LABEL(1,3) = 'WHIRL BW'
LABEL(1,4) = 'WHIRL FW'
FINISH
```

3.5 Key Milestone and Gantt Chart

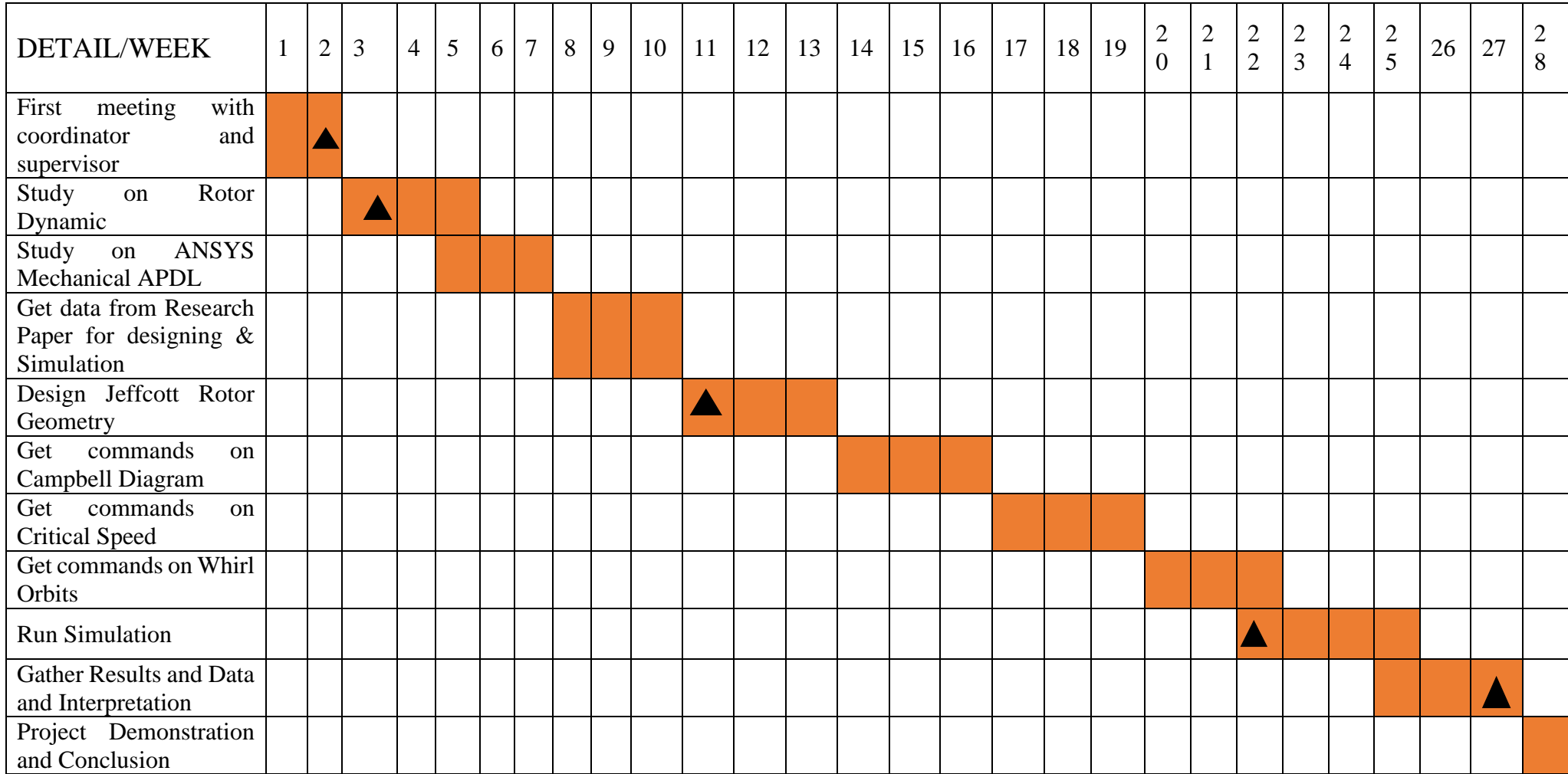


Figure 14: Gantt Chart of Final Year Project

Key Milestone

Below are the suggested Key Milestone for Final Year Project

- ▲ Problem Statement and Objective of the project
- ▲ Literature Review
- ▲ Simulation on ANSYS
- ▲ Data Analysis and Interpretation
- ▲ Documentation and Reporting

CHAPTER 4

RESULT AND DISCUSSION

4.1 Campbell Diagram and Critical Speeds

In this analysis, the speed that performed for the Jeffcott Rotor are from 0 to 9000 (RPM) using multiple load steps. The analysis undergoes rotation along the axis. Figure 4.1.1 shows the result of Campbell diagram in rotor dynamic analysis. Along the x-axis shows the rotational speed (RPM) whereas the frequency (Hz) plotted along y-axis. The frequency varies along the rotational speed. It can be seen that the forward and backward whirl branch out from the speed point. This is because inherent dependence on the shaft rotational speed. The critical speed can be identify in Campbell diagram.

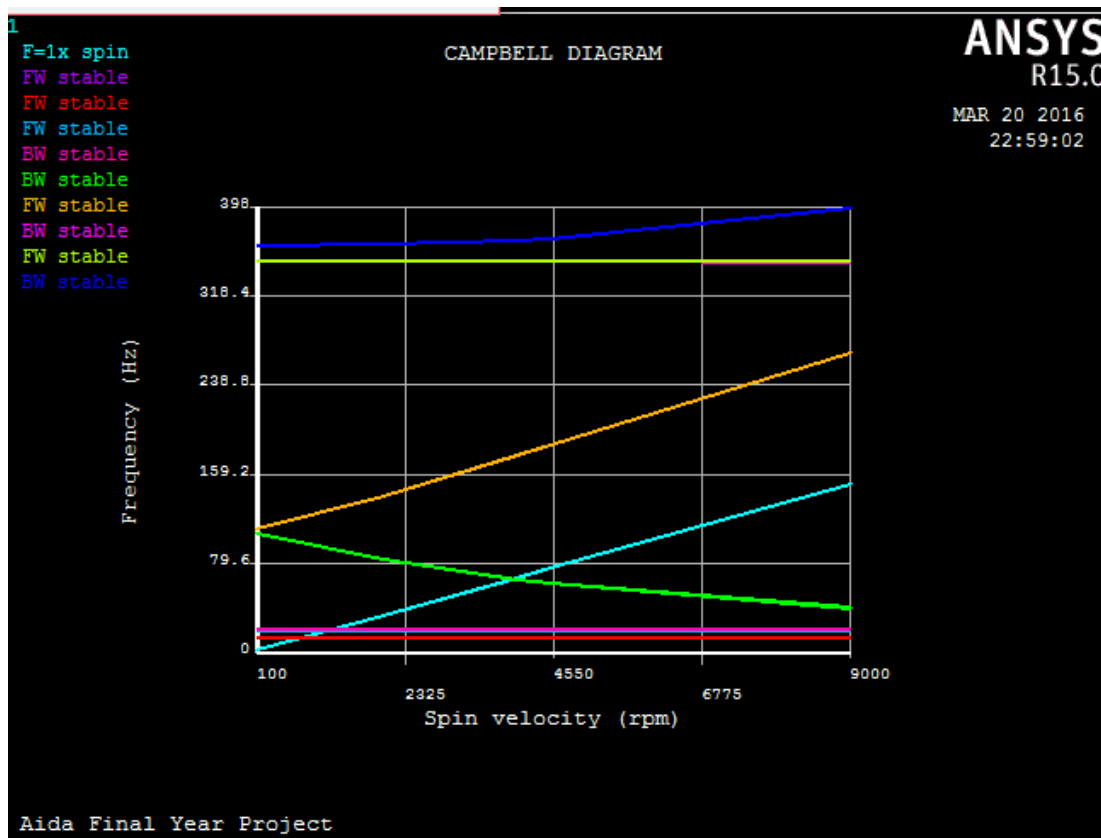


Figure 15: Campbell Diagram

When a 45° line cutting the natural frequency, critical speeds of the rotor can be identified. The aqua blue line is known as critical speed line. Its starts from the origin and crosses the frequency line and rotational speed line. Figure 4.1.2 shows the results of critical speed in the rotor. From the table, the critical speed line across line mode 2,3,4, and 5. At critical speed, the vibration of the rotor increase drastically. Therefore, at mode 2,3,4, and 5 vibrates vigorously along the Jeffcott rotor. The critical speed line does intersect at line mode 1,6,7,8, and 9. Hence, no critical occurs on those lines.

Mode	Whirl Direction	Critical Speed
1	FW	NONE
2	FW	862.34
3	FW	1412.58
4	BW	1216.57
5	BW	3225.44
6	FW	NONE
7	BW	NONE
8	FW	NONE
9	BW	NONE

Table 3: Critical Speed of Analysis

4.2 Mode Stability

From the Campbell diagram in Figure 4.1.1, we can also determine the threshold stability of each mode. Table 4.2.1 shows the stability of each mode at the Jeffcott rotor. The stability of a system can be determined with the help of a damping. In other cases, the internal damping may decrease the stability of a rotor. Hence the rotor will be undesirable.

Mode	Whirl Direction	Mode Stability
1	FW	STABLE
2	FW	STABLE
3	FW	STABLE
4	BW	STABLE
5	BW	STABLE
6	FW	STABLE
7	BW	STABLE
8	FW	STABLE
9	BW	STABLE

Table 4: Mode Stability of Jeffcott Rotor

4.3 Whirl Orbits

The whirl orbits from the ANSYS Mechanical APDL are shown in Figure 4.3.1. Each orbit in the figure represents each mode at the shaft and the disk. The figure shows the rotation system in horizontal and vertical direction. Ellipse whirls are shown because the supporting structure which is the bearings have different value of stiffness.

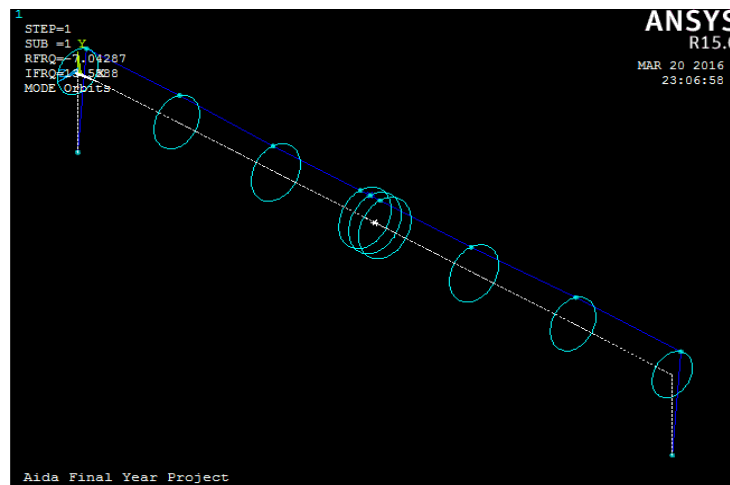


Figure 16: Whirl Orbit of Jeffcott Rotor

CHAPTER 5

CONCLUSION

It can be concluding that the analysis shows the critical speed at mode 2,3,4 and 5 only. No critical speed detected at the rest of the mode. Thus, no vibration occurs. ANSYS software is capable of executing simulation to evaluate vibration of a system. Based on the result and discussion; the author will conduct a simulation using the same parameters but different value. The project is considered to be feasible by taking into account the time constraint and the capability of final year student with the assist from the supervisor and coordinator.

RECOMMENDATION

There are many type of cases that can improve this case study. More parameters and variables that can be consider for future case study. For example, run the simulation the rotor with casing and without casing. Moreover, there would be cases where the supporting structure(bearings) have different stiffness. With this case study, the whirl orbit will have different, pattern of Campbell diagram and different critical speed. With further study, this case study will enhance the knowledge of rotor dynamic analysis with different stiffness bearings.

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APPENDICES

```
/COM,Structural
/TITLE, Aida Final Year Project
/OUT,SCRATCH
/PREP7
*DIM,SPEED,,5           ! SPIN VELOCITY (RPM)
SPEED(1) = 100.
SPEED(2) = 2000.
SPEED(3) = 4000.
SPEED(4) = 4500.
SPEED(5) = 9000.
! *****
! MATERIAL PROPERTIES
! *****
MPTEMP,1,0
MPDATA,EX,1,,2e11
MPDATA,PRXY,1,,0.3
MPDATA,DENS,1,,7800
! *****
! NODES
! *****
N,1,0
N,2,0.25
N,3,0.5
N,8,0.725
N,4,0.75
N,9,0.775
N,5,1
N,6,1.25
```

```

N,7,1.5
N,10,0,-0.2
N,20,1.5,-0.2
! *****
! ELEMENT PROPERTIES
! *****
ET,1,BEAM188,,2
SECTYPE,1,BEAM,CSOLID
SECDATA,0.03,20
SECTYPE,2,BEAM,CTUBE
SECDATA,0.03,0.28,30
TYPE,1
MAT,1
SECNUM,1
*DO,I,1,6
  E,I,I+1
*ENDDO
TYPE,1
SECNUM,2
FLST,2,2,1
FITEM,2,8
FITEM,2,4
E,P51X
FLST,2,2,1
FITEM,2,9
FITEM,2,4
E,P51X
ET,2,MASS21
R,2,1.401,1.401,1.401,0.002,0.00136,0.00136
TYPE,2

```

```

REAL,2
E,4
ET,3,COMBI214
KEYOPT,3,2,1
KEYOPT,3,3,1
R,3,12.81e6,8.815e6,16.39e6,-25.06e6,232.9e3,294.9e3,      ! COMBI214:  r,
,kxx,kyy,kxy,kyx,cxx,cyy,
RMORE,-81.92e3,-81.92e3,      ! rmore, cxy,cyx,
TYPE,3
REAL,3
FLST,2,2,1
FITEM,2,1
FITEM,2,10
E,P51X
FLST,2,2,1
FITEM,2,7
FITEM,2,20
E,P51X
ALLSEL,ALL
! *****
! DISPLACEMENT
! *****
D,ALL,UX,
D,ALL,ROTX
D,10,ALL
D,20,ALL
ALLSEL,ALL,ALL
! *****
! TRANSIENT DIAGRAM
! *****

```

```

/ESHAPE,1.0
EPlot
/VIEW,1,1,1,1
/ANG,1
/REPlot,FAST
FINI
! *****
! ANALYSIS TYPE
! *****
/SOLU
RATIO = 4*ATAN(1)/30
ANTYPE,MODAL
NBF = 11                ! NUMBER OF MODES
CORIOLIS,ON,,,ON      ! CORIOLIS ACTIVE FOR STATIONARY FRAME
MODOPT,QRDAMP,NBF,,,ON
*DO,I,1,5
    OMEGA,SPEED(I)*RATIO    ! ROTATIONAL VELOCITY ABOUT X-DIRECTION
    MXPAND,NBF
    SOLVE
*ENDDO
FINI
! antype,harmic
/POST1
! *****
! CAMPBELL DIAGRAM
! *****
PRCAMP,,1,RPM
*GET,F1,CAMP,1,FREQ,5
*GET,F2,CAMP,2,FREQ,5
*GET,F3,CAMP,3,FREQ,5

```



```
*GET, F4, CAMP, 4, FREQ, 5
*DIM, LABEL, CHAR, 1, 4
*DIM, VALUE, ARRAY, 4, 3
LABEL(1,1) = 'WHIRL BW'
LABEL(1,2) = 'WHIRL FW'
LABEL(1,3) = 'WHIRL BW'
LABEL(1,4) = 'WHIRL FW'
FINISH
```

