B. TECH. PROJECT REPORT

On

Effect of Coupling Types on Rotor Vibration

BY

Ronak Kanubhai Prajapati (130003030)

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Effect of Coupling Types on Rotor Vibration

A PROJECT REPORT

Submitted in partial fulfillment of the requirements for the award of the degrees

of
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Submitted by:
RONAK KANUBHAI PRAJAPATI

Guided by:
Dr. Anand Parey, Associate Professor, IIT Indore

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<u>CANDIDATE'S DECLARATION</u>

I hereby declare that the project entitled **"Effect of Coupling Types on Rotor Vibration"** submitted in partial fulfillment for the award of the degree of Bachelor of Technology in 'Mechanical Engineering' completed under the supervision of **Dr. Anand Parey, Associate Professor, Mechanical Engineering, IIT Indore** is an authentic work.

Further, I declare that I have not submitted this work for the award of any other degree elsewhere.

Signature and name of the student with date

<u>CERTIFICATE by BTP Guide(s)</u>

It is certified that the above statement made by the student is correct to the best of my/our knowledge.

Signature of BTP Guide(s) with dates and their designation

Preface

This report on "**Effect of Coupling Types on Rotor Vibration**" is prepared under the guidance of Dr. **Anand Parey**.

In this report, I have tried to introduce the reader to effect of coupling types and increase in shaft speed on vibration amplitudes of a rotor dynamic system, while at the same time collecting our experiences in operating the required system. Most importantly, the report contains the findings of our study which complement as well as add on the knowledge base acquired by our literature review.

I have tried my best in explaining the content in a lucid manner constrained by our abilities and knowledge. I have also added graphs and diagrams where possible/applicable to make it more illuminating.

Ronak Kanubhai Prajapati
B.Tech. IV Year
Discipline of Mechanical Engineering
IIT Indore

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Last but not the least; I would like to thank my parents for their unwavering support and encouragement.

Ronak Kanubhai Prajapati

B.Tech. IV Year

Discipline of Mechanical Engineering

IIT Indore

Abstract

Couplings are widely used in compressors, gas turbines, and aerospace applications because of their ability to transmit torque and their consequent ability to compensate misalignment in almost all directions. The basic function of a coupling is to transmit torque from the driver to the driven piece of rotating equipment. The coupling type and increase in shaft speed has a great influence on rotor vibration. Investigation of the factors influencing the rotor vibrations due to coupling type and increase in shaft speed will help to improve the design and control of a rotor dynamic system. In the following pages, a 3D finite element (FE) model of a rotor dynamic system is built with different coupling types i.e. Rigid, Jaw, Flexible Couplings. Based on this model, the effect of coupling type on natural frequencies of the system was observed. An experimental setup was established with the help of Machine Fault Simulator to observe the change in vibration amplitude with coupling types and increase in shaft speed. The experimental results showed that coupling type greatly influenced the vibration amplitude and a trend was observed while increasing the shaft speed. Finally, conclusions were delivered based on these results and further research scope was discussed.

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CHAPTER 1

INTRODUCTION

Due to the high speed of some rotating machinery, the need for a better understanding of the vibration phenomena is becoming a necessity for practical engineers for the purpose of troubleshooting. Most rotating equipment consists of a driver and driven machine coupled through a mechanical coupling. The mechanical coupling is used mainly to transmit torque from the driver to the driven machine [1]. Due to current trends in the design of rotating machinery towards higher speeds and lower vibration manufacturers are tending to produce machines which operate closer to lateral critical speeds than has previously been necessary. Consequently, the effect of coupling upon the higher speeds and misalignment on vibration amplitudes of such machines is becoming an increasingly important consideration for rotor-bearing systems [2].

The vibration in rotating machinery is mostly caused by unbalance, misalignment, mechanical looseness and other malfunctions. However, the perfect alignment between the driving and driven machines cannot be attained in real world [3-5].

To ensure that the rotor of a piece of rotating machinery is sufficiently designed to withstand the stresses and strains of the operating environment, the coupling chosen to join the driving and driven mechanisms must be properly selected. In addition to transmitting torque from the driving to the driven pieces of machinery, the coupling must also compensate for all possible unintentional vibrations apart from misalignment of the rotating devices [1].

CHAPTER 2

COUPLING

Introduction

Couplings are mechanical elements that ‘couples’ two drive elements which enables motion to be transferred from one element to another. The drive elements are normally shafts. We tend to see lot of applications of the couplings mainly in the automobiles, for example the drive shaft which connects the engine and the rear axle in a bus or any automobile is connected by means of a universal joint.

As with all mechanical devices, a coupling must match its’ intended purpose and application parameters, including many different performance, environmental, use and service factors. All must be satisfied for the coupling to work properly. When selected with these design parameters in mind, and when installed and operating correctly, a coupling should have no failure issues over its’ lifetime. However, when one or more of these is not met a coupling can prematurely fail, resulting in either a small inconvenience or possibly serious financial loss or personal injury. [6]

Types

In order to transmit torque between two shafts that either tend to lie in the same line or slightly misaligned, a coupling is used. Based on the area of applications there are various types of coupling available. But they are generally categorised in the following varieties

1. Rigid Couplings

2. Flexible or Compensating Couplings

3. Miscellaneous Couplings

Rigid Couplings

Rigid Couplings are mainly used in areas where the two shafts are coaxial to each other. By virtue of their simple rugged design rigid couplings are generally able to transmit more power but do not have the ability to compensate for shaft misalignment.

Types of couplings that fall under the rigid couplings category are: -

- Rigid Sleeve or Muff Couplings

This is the basic type of coupling. This consists of a pipe whose bore is finished to the required tolerance based on the shaft size. Based on the applications of the coupling a keyway in made in the bore in order to transmit the torque by means of the key. Four threaded holes are provided in order to lock the coupling in position. The photo shows a type of the rigid sleeve or muff coupling.

Fig 2.1: Rigid Sleeve Coupling

- Flanged Coupling

The coupling basically consists of two flanged end pieces as shown in the figure. A spigot and recess is provided in the flanges to provide location between them. The flanges are connected firmly by means of fitted bolts which are tightened accordingly to the torque to be transmitted.

Fig 2.2: Flanged Coupling

Flexible Couplings

Flexible couplings are normally used in areas where the coaxiallity between the connecting shafts is not always assured and in areas where there is a possibility of occurrence of shocks in the transmission is applicable. They are also called as Elastic Couplings.

The different types of flexible couplings are

1. Flanged Pin Bush Couplings
2. Beam Coupling
3. Gear Tooth Coupling
4. Tyre Coupling
5. Oldham Coupling
6. Universal Coupling or Hooke’s Coupling
7. Bellows Coupling

- Beam Coupling

A beam coupling, also known as helical coupling, is a flexible coupling for transmitting torque between two shafts while allowing for angular misalignment, parallel offset and even axial motion, of one shaft relative to the other. This design utilizes a single piece of material and becomes flexible by removal of material along a spiral path resulting in a curved flexible beam of helical shape. Since it is made from a single piece of material, the Beam Style coupling does not exhibit the backlash found in some multi-piece couplings.

Fig 2.3: Beam Coupling

Miscellaneous Couplings

This group of couplings incorporate design features which are frequently unique, approximations or combinations of universal, Oldham and flexible shaft couplings. Two widely used couplings in this category are the Jaw and Sleeve types.

- Jaw Coupling

Jaw type couplings Consist of two metal hubs which are fastened to the input and output shafts. Trapped between the hubs is an elastomer element, known as a spider or cushion whose legs are confined between alternating metal projections from the adjacent hubs. The spider is the wearing member and can be readily replaced without dismantling adjacent equipment. The coupling is capable of operating without lubrication and is unaffected by oil, grease, dirt or moisture.

Fig 2.4: Jaw Coupling

Coupling Selection Criteria's

No one type of coupling can provide the universal solution to all coupling problems; hence many designs are available, each possessing construction features to accommodate one or more types of application requirements. The coupling selection is based on operating factors. The coupling has to be dimensioned in a way that the permissible coupling load is not exceeded with any operating condition. For this purpose, the actual loads have to be compared to the permissible parameters of the coupling. The shaft-hub-connection has to be investigated by the customer. [11]

Step 1. <u>Nominal Torque</u>

Determine the Nominal Torque of your application by using the following formula:

$$Nominal\ Torque\ \ Nm = \frac{(kW \times 9550)}{RPM}$$

(kW = HP x 0.7457)

Step 2. <u>Service Factor</u>

Using the Application Service Factors select the service factor which best corresponds to your application.

Load	Driving Equipment	
	Motor or Turbine	Reciprocating Engine
Uniform	1.0	1.5
Light Shock	1.5	2.0
Medium Shock	2.0	2.5
Heavy Shock	2.5	3.0

Table 2.1: Service Factor Selector

The service factors listed are intended only as a general guide. For typical service factors used in various applications refer to manufacturers catalogue.

Step 3. <u>Design Torque</u>

Calculate the Design Torque of your application by multiplying the Nominal Torque calculated in Step 1 by the Application Service Factor determined in Step 2.

Design Torque = Nominal Torque × Application Service Factor.

Step 4. <u>Performance Data</u>

Using the Performance Data Chart, locate the Nominal Torque column. Scan down this column to the first entry where the Torque Value is greater than or equal to the Design Torque calculated in Step 3.

Step 5. <u>Maximum RPM</u>

Refer to the maximum RPM value for the coupling to ensure that the application requirements are met (some couplings have different maximum RPM for different elements). If the requirement is not satisfied at this point, another type of coupling may be required for the application.

Step 6. <u>Comparison</u>

Compare the application driver/driven shaft sizes to the maximum bore size available on the coupling selected. If coupling bore size is not large enough for the shaft diameter, select the next largest coupling that will accommodate the driver/driven shaft diameters. Then recheck the maximum RPM value for the new coupling, as the maximum RPM value will drop off as the coupling size increases.

Selection parameters for shaft couplings

SKF designation/name selection criteria	General type/family	Coupling type (→ note 1)	Shaft capacity range (→ note 2)	Maximum torque capacity (→ note 3)	Power capacity (per 100 r/min)	Maximum r/min (for smallest coupling)	Maximum parallel misalignment (β)	Maximum angular misalignment (α°)
–	–	–	mm	Nm	kW	r/min	mm	°
Flex	Tyre	E	9–190	14 675	525	4 500	1,1–6,6	<= 4°
Gear	Gear	M	13–425	555 000	5,810	8 000	1,2–12,7	<= 1,5°
Grid	Taper grid	M	12–420	336 000	3,523	4 500	0,3–0,76	<= 1,4°
Disc	Laminate disc	M	10–190	40 000	4,200	24 000	(→ note 10)	<= 0,67°
Chain	Chain	M	10–155	17 100	136	5 000	0,038	<= 2°
FRC	Jaw	E	9–100	3 150	33	3 600	0,5	<= 1°
Jaw (L)	Straight jaw	E	9–60	280	2,7	3 600	0,038	<= 1°
Rigid	Rigid	M	32–125	4 000	118	4 500	Use only with very good alignment	
Universal joint	Universal joint	M	6–55	5 300	– (→ note 9)	1 800	Ref Cat.	<= 25°
ES-Flex (USA only) (→ note 7)	Elastomer-in-shear	M	12–152	8 135	85	9 200	1,6	<= 1°

Table 2.2: SKF Coupling Selector Guide

CHAPTER 3

LITERATURE REVIEW

Effect of Coupling Type on Bearing Forces

As a coupling flexes, it generates forces at the shaft support bearings. But designers seldom consider these forces (loads) when selecting couplings and bearings. The reaction forces may seem surprisingly high to engineers who never considered them before. In some equipment, especially delicate instruments that have slender shafts running in fragile bearings, such forces can shorten the life of these bearings and shafts. Radial misalignment of parallel shafts causes larger reaction forces than angular misalignment. [7]

- Flexible Couplings

The reaction, or resistance, of flexible couplings to radial (and angular) misalignment increases proportionally with shaft deflection. This reaction can be defined as a spring rate and expressed as force per unit deflection. Because flexible couplings accommodate misalignment in a bending mode, reaction is proportional to the flexing member thickness. In some types, such as the membrane and bellows couplings, the flexing members can be thin because torque is transmitted with the members in shear. Thus, high torsional stiffness is attained with relatively low radial forces.

On the other hand, couplings such as the beam and flexible leaf, transmit torque with the members in bending, the same mode used to accommodate misalignment. Here, relatively thick members are needed to obtain a reasonable level of torsional stiffness. But, making the coupling torsionally stiff causes proportionally higher radial reaction forces.

The magnitude of radial forces depends on the severity of the bend. It follows that keeping bending angles to a minimum does the same for the bearing forces, but inevitably reduces misalignment capability. The longer distance between flexure points enables them to operate with shallower bends for a given radial shaft offset, and so reduces the radial forces. Couplings with multiple flexing elements (beam and bellows) can also use this approach, but generally require extensive modifications to divide the flexing elements into two separate clusters. Some short models are available that can be connected by an intermediate shaft [7].

- Elastomeric couplings

Certain coupling types incorporate flexible elastomeric elements and offer a range of torsional damping properties. But their associated characteristics — relative rotation between hubs, and backlash — can be counter-productive in precision motion control applications. Depending on the coupling type, elastomers are loaded in shear, compression, bending, or combinations of these modes.

One example of an elastomeric device, the jaw coupling is probably the most widely used general purpose coupling. This device transmits torque and accommodates misalignment by compressing legs (or lobes) of an elastomeric insert between its jaws.

As with flexural couplings, the reaction forces of elastomeric jaw couplings are proportional to radial shaft deflection. However, the ability of these couplings to handle radial misalignment and minimize bearing forces is generally less than for other types. Therefore, the jaw coupling is best for applications where the connected shafts can be manipulated into near alignment so as to keep bearing forces within an acceptable range.

Softer elastomer inserts can be used to accommodate radial misalignment and minimize bearing forces. However, the trade-off is a commensurate reduction in torsional stiffness. Two jaw couplings mounted back-to-back will accommodate radial misalignment, but are rarely used this way, usually because of added cost. [7]

- Rigid Couplings

The two elementary rigid coupling styles are machined set-screw, for smaller shafts, and ribbed, two-piece cast, for larger and higher horsepower applications. The ribbed style, all but unchanged, remains the coupling of choice for large shafts, but it is uneconomical for shafts two inches or less in diameter. The machined, set-screw coupling is basically a cylinder with a keyway and set screws. From a simplistic viewpoint, this is a sound design, but in practice its shortcomings become evident. It can loosen under vibration, and set screws can leave dents or dimples in the shaft or keyway, deforming the surface so that adjusting or removing the coupling becomes difficult. While hardened, shafts may resist dents, they also can prevent set screws from locking and properly holding in place. And with set-screw couplings, keyways are necessary for torque transmission, thus the phase relationship between coupled shafts cannot be changed. Clamping couplings don’t damage the shaft surface, they serve on hardened shafting, and they maintain hold under vibration and cyclic or reversing loads. Also, assembly and adjustment are not generally a problem. The coupling will clamp down around the shaft circumference when the cap screws are tightened, and static friction between coupling and shaft keeps them turning together.

This holding friction is influenced by the number of cap screws, the applied axial force along the each of these screws, and the coefficient of friction between contacting coupling and shaft surfaces. Axial screw force is a function of screw thread lubrication and tightening torque, while the frictional coefficient is affected by the hardness, lubricity, and finish of contacting surfaces.

However, the ability of these couplings to handle radial misalignment and minimize bearing forces is very less compared to Elastomeric and Flexible Couplings.

Faults in Rotor-Bearing System

Rotor Unbalance

In any rotating machinery, the rotor unbalance is always present and it is one of the most common sources of severe vibration. The unbalance is defined as the product of the rotor mass and its eccentricity (the eccentricity is a distance of the center of gravity of the rotor from its center of rotation). When a severely unbalanced rotor is rotated freely on frictionless bearings, it stops at nearly fixed orientation. It indicates that the gravity force acting at center of gravity pulls the rotor to a fixed orientation due to its eccentricity.

The unbalance vibration occurs at machine rotational frequency, in general, but sometimes higher-harmonics of rotational speed are excited. Machine vibration caused by unbalance can mostly be detected by monitoring shaft displacement amplitude and phase as the machine is run through its critical speed, filtering out non-rotational speed frequencies by using tracking filter.

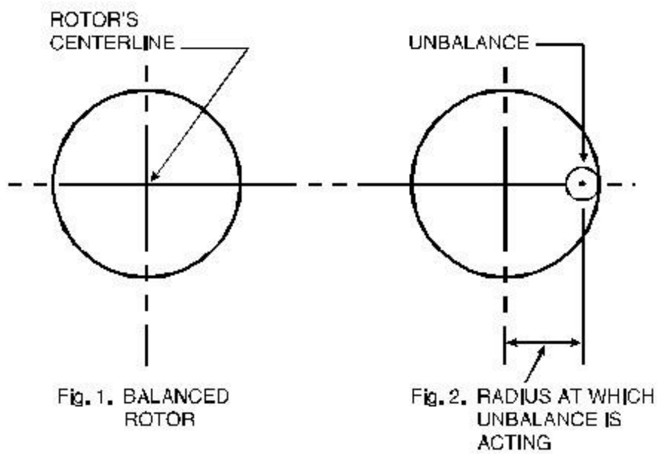


Fig 3.1: Rotor Unbalance

Misalignment

The objective of alignment is to have two coupled shafts perfectly collinear under operating conditions or between the bearing and the shaft their axis should be co-linear or two bearings carrying a common shaft should have their axis collinear. Accordingly, misalignments can be classified as (i) parallel (ii) angular, and (iii) combination of parallel and angular misalignments (Fig. 3.2). Like the unbalance, the misalignment is an installation and subsequent maintenance problems, since it can be corrected and prevented by using the proper installation and maintenance procedures. Shafts with a heavy pre-load carried by the bearings (e.g., in angular contact ball bearings in tandem, the preloading must be applied to keep the bearing in assembled position), as distinct from the out-of-balance load, can show variation characteristics similar to those caused by bearing misalignment. This category of fault is probably the second most common cause of machine vibration, after the unbalance. A pre-load might also be applied at a bearing as a consequence of gear-mesh forces, aerodynamic forces and hydrodynamic forces. Misalignment may be present because of improper machine assembly or as a consequence of thermal distortion, and it results in additional loads being applied to the bearing.

The misalignment of adjoining shafts, or of the bearings of one shafts, causes abnormal loads to be transmitted through the bearings, and imposes additional bending stress on the shaft thereby reducing its fatigue life. In some cases misalignment, may cause one bearing to be unloaded (if the pre-load so applied is in the opposite direction to the normal gravity loads, this can result in lowering the machine critical speed or even instability in the system. These symptoms are sometimes present in addition to that of excessive vibration. The vibration associated with misalignment occurs at 1x machine running speed but, unlike unbalance case, there is usually a substantial component in the axial direction, which may be greater than radial direction.

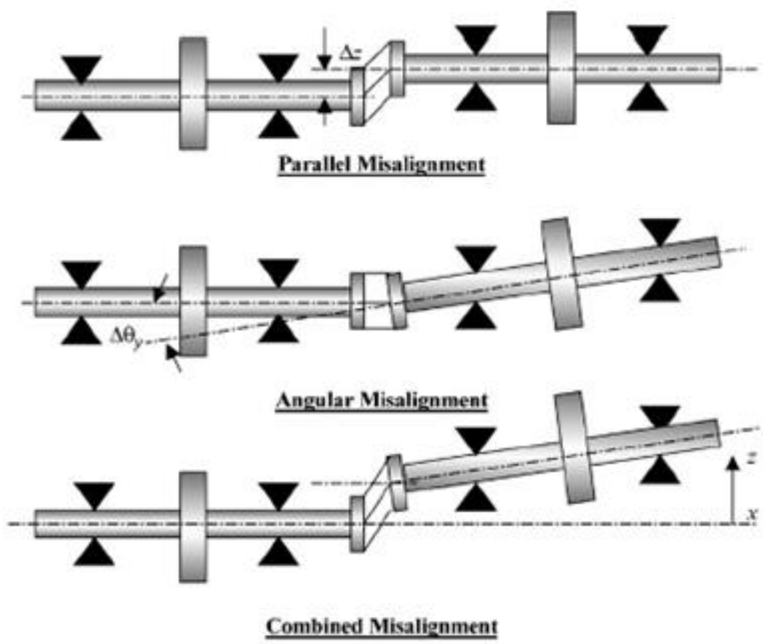


Fig 3.2: Schematic of Rotor with Misalignment at A Coupling

Mechanical Rub

Mechanical Rubs are produced when the rotating shaft comes into contact with the stationary components of the machine. Mechanical Rubs are said to be accompanied by a great deal of high-frequency spectral activities. A Mechanical Rub is generally a transitory phenomenon. The Mechanical Rub may typically be caused by the mass unbalance, turbine or compressor blade failure, defective bearings and/or seals, or by rotor misalignment, either thermal or mechanical. Several different physical events may occur during a period of contact between the rotor and the stator: initial impacting stage, frictional behavior between the two contacting parts and an increase in the stiffness of the rotating system whilst contact is maintained, to name just three. The behavior of the system during this period is highly non-linear and may be chaotic.

The main component of Mechanical Rub is at 1 x rotational speed; harmonics and sub-harmonics may also be present. Alternatively, the tangential friction force between shaft and stationary component may be so large that it results in a backward precession of the shaft around the inside of the stationary component together with substantial slipping.

Mechanical Looseness of Components

Mechanical looseness, the improper fittings between component parts, is generally characterized by a long string of harmonics of running frequency with abnormally high amplitudes. In some machines vibration levels, may be excessive as a consequence of components being assembled too loosely, for example in the case of a bearing, which is not properly secured.

Pedestal looseness is one of the common faults that occur in rotating machinery. It is usually caused by the poor quality of installation or long period of vibration of the machine. Under the action of the imbalance force, the rotor system with pedestal looseness will have a periodic beating. This will generally lead to a change in stiffness of the system and the impact effect. Therefore, the system will often show very complicated vibration phenomenon. In the case of rotor/bearing/stator systems, the dynamic phenomena (which include chaotic and orderly periodic motion of system elements), occur usually as secondary effects of a primary cause. This primary cause is most often an action of the rotor unbalance related rotating force, directly exciting rotor lateral vibrations, and/or the unilateral, radial force applied to the rotor.

Shaft Cracks

The presence of various flaws (such as cracks, notches, slits etc.) in any structures and machineries may lead to catastrophic failures. They are particularly likely to occur in instances where shaft stresses are high and where machine has endured many operation cycles throughout its life, so that material failure has occurred as a consequence of fatigue. If the shaft cracks can be detected before catastrophic failure occurs, then the machine can be temporarily taken out of service and repaired before situation gets out of hand. The presence of a transverse shaft crack sometimes is detected by monitoring changes in vibration characteristics of the machine. The shaft stiffness at the location of the crack is reduced, by an amount depending on the crack size. This in turn affects the machine natural frequencies, so that changes in natural frequencies may be symptomatic of a shaft crack.

Modal Analysis

Modal analysis is used to determine the vibration characteristics of a structure, namely the natural frequencies and the mode shapes of the structure. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions which determined by the inherent characteristics and the materials of a structure. They are also required if you want to do a spectrum analysis or a mode superposition harmonic or transient analysis. By modal analysis on the rotor-bearing system we can get the response of the structure withstands the different kinds of dynamic loads. On the one hand, the vibration characteristics for structure can be predicted. On the other hand, we can avoid the resonance phenomenon occurs in the design process, do the modification of the existing structure, and improve the reliability of the machine. [rotary comp]

- Modal Analysis Using FEM

For the most basic problem involving a linear elastic material which obeys Hooke's Law, the matrix equations take the form of a dynamic three-dimensional spring mass system. The generalized equation of motion is given as: [8]

$$[M][\ddot{U}] + [C][\dot{U}] + [K][U] = [F]$$

where [M] is the mass matrix, $[\ddot{U}]$ is the 2nd time derivative of the displacement [U] (i.e., the acceleration), $[\dot{U}]$ is the velocity, [C] is a damping matrix, [K] is the stiffness matrix, and [F] is the force vector. The general problem, with nonzero damping, is a quadratic eigenvalue problem. However, for vibrational modal analysis, the damping is generally ignored, leaving only the 1st and 3rd terms on the left-hand side:

$$[M][\ddot{U}] + [K][U] = [0]$$

This is the general form of the eigensystem encountered in structural engineering using the FEM. To represent the free-vibration solutions of the structure harmonic motion is assumed, [9] so that $[\ddot{U}]$ is taken to equal $\lambda[U]$, where λ is an eigenvalue (with units of reciprocal time squared, e.g., s^{-2}), and the equation reduces to: [10]

$$[M][U]\lambda + [K][U] = [0]$$

In contrast, the equation for static problems is:

$$[K][U] = [F]$$

which is expected when all terms having a time derivative are set to zero.

Vibration Analysis

Vibration is a mechanical phenomenon whereby oscillations occur about an equilibrium point. The oscillations may be periodic, such as the motion of a pendulum—or random, such as the movement of a tire on a gravel road. In many cases, however, vibration is undesirable, wasting energy and creating unwanted sound. For example, the vibrational motions of engines, electric motors, or any mechanical device in operation are typically unwanted. Such vibrations could be caused by imbalances in the rotating parts, uneven friction, or the meshing of gear teeth. Careful designs usually minimize unwanted vibrations.

Vibration Analysis (VA), applied in an industrial or maintenance environment aims to reduce maintenance costs and equipment downtime by detecting equipment faults. Most commonly VA is used to detect faults in rotating equipment (Fans, Motors, Pumps, and Gearboxes etc.) such as Unbalance, Misalignment, rolling element bearing faults and resonance conditions.

VA can use the units of Displacement, Velocity and Acceleration displayed as a Time Waveform (TWF), but most commonly the spectrum is used, derived from a Fast Fourier Transform of the TWF. The vibration spectrum provides important frequency information that can pinpoint the faulty component.

CHAPTER 4

SETUP

Modal Analysis

For Modal analysis in ANSYS Workbench, a 3D CAD Model of the system needs to be developed with the help of a modelling software i.e. Solidworks 2013. Rotor shaft, Rotor, Motor Shaft and Couplings are modeled using Soliworks 2013 with the exact dimensions as used in the experimental setup. The Couplings were designed from data available from manufacturer website. Material Properties of each component of the system needs to be acquired to replicate the stiffness and rigid behavior of components experiencing vibration. Below are the images of the system with acquired measurements. All the dimensions shown are in mm.

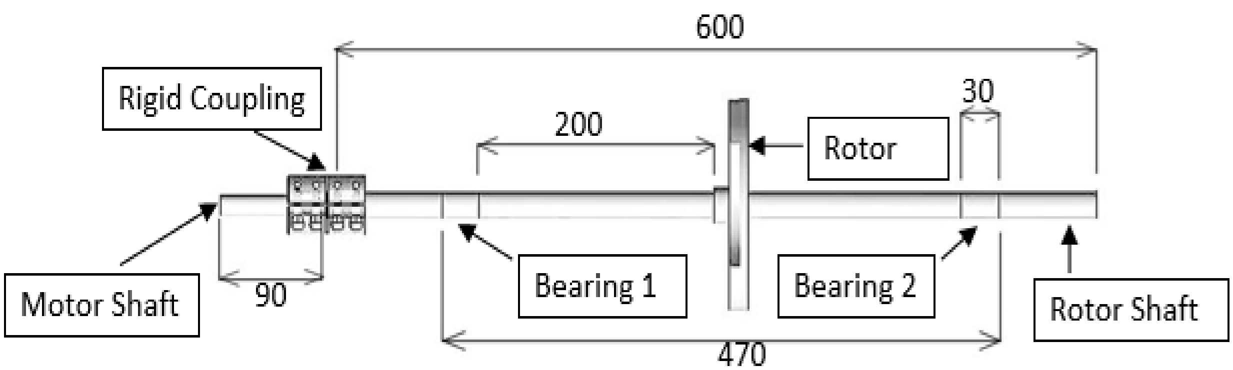


Fig 4.1: 2D Model of Rotor-Coupling System

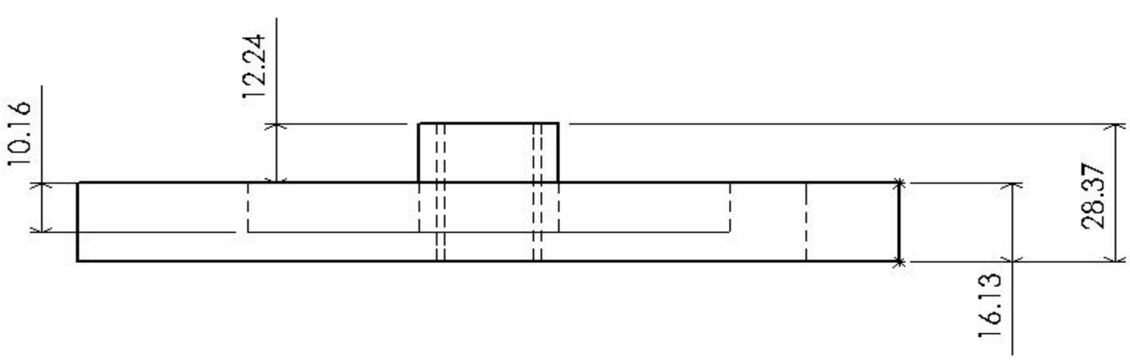


Fig 4.2: 2D Model of Rotor [Side View]

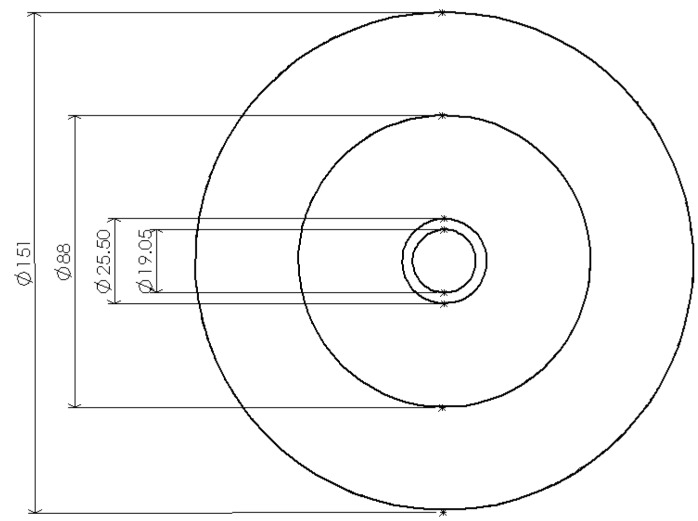


Fig 4.3: 2D Model of Rotor [Front View]

Coupling Type	**Bore(mm)**		**Torque (Nm)**	**Max Speed (RPM)**	**Length (mm)**
	Max	**Min**			
Rigid	20	16	350	4000	65
Flexible	19	16	21.47	10,000	63.5
Jaw	19	-	4.88	14,000	50.3

Table 4.1: Coupling Specifications

Item	Dimension/mm
Rotor Shaft Diameter	19
Motor Shaft Diameter	15.8

Table 4.2: Shaft Dimensions

Item	Material	Density [g /cc]	Poisson Ratio	Young’s Modulus
Rotor Shaft	TGP	7.87	0.29	206 GPa
Motor Shaft	C45 Carbon Steel	7.85	0.29	210 GPa
Rotor	Aluminium	2.6898	0.34	68.3 GPa
Rigid Coupling	1215 Carbon Steel	7.87	0.29	200 GPa
Jaw Coupling	Sintered Iron	6.6	0.25	115 GPa
Spider (Jaw Coupling)	Rubber	1	0.49	0.003 GPa
Flexible Coupling	Aluminium	2.6898	0.34	68.3 GPa

Table 4.3: Material Properties

Vibration Analysis

While analysis of a single machinery fault may be beneficial, there are many occasions when the analysis of the interaction between dynamic stiffness, resonance, and speed is essential in order to gain an understanding of real world vibration spectra. With the MFS, the expertise required to diagnose industrial machinery problems in well controlled experiments can be developed and enhanced. With a plant running at full production, it is virtually impractical to gain an understanding of the kinetics and dynamics of machinery without adversely affecting production and profits: The MFS enables offline training and experimentation which in turn will minimize production downtime.

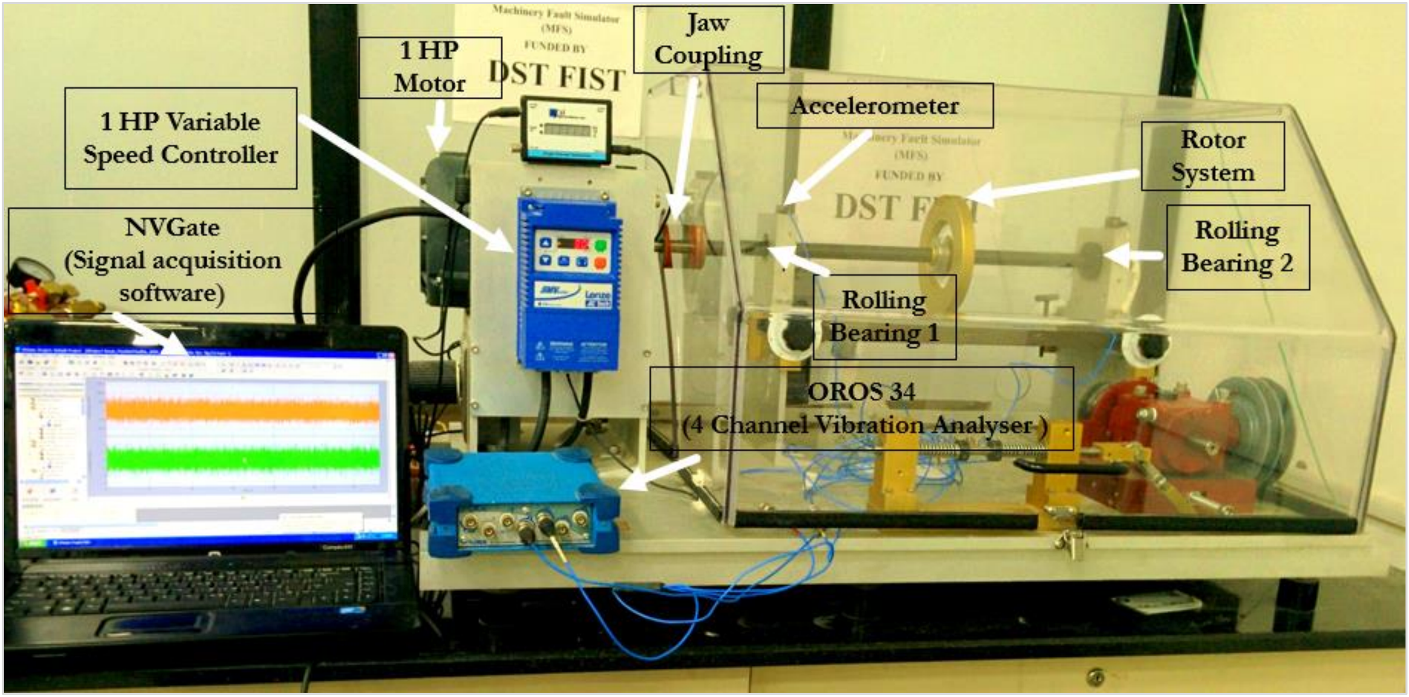


Fig 4.4: Experimental Setup for Vibration Analysis

Fig 4.5 depicts the experimental setup to study the effect of coupling types on rotor vibration. It consists of a DC motor, Jaw coupling and circular rotor on the shaft. The shaft of 19 mm diameter is supported by two identical rolling bearings. The bearing pedestals are provided in such a way as to adjust in vertical direction to align the system or to create misalignment. The shaft is driven by a 1 HP DC motor. A 1 HP Variable Speed controller is used so that the system can be operated at different speeds.

A piezoelectric accelerometer manufactured by PCB PIEZOTRONICS (Type ICP, no 333B32) with 10.57mV/(m/s^2) sensitivity is used along with the 4-Channel Vibration Analyser (OROS 34). The accelerometer is mounted on the bearing point with the help of wax provided with the kit. This accelerometer connects to OROS-34 with the help of 10-32 plug to BNC plug attached to a white 10-ft long FEP jacket coaxial cable. The OROS 34 is connected with LAN to a Software Platform NVGate installed in the Laptop Cable, the vibration signal is recorded in NVGate, it hosts the suite of OROS software modules. The Recorder captures raw, time-domain data during your acquisition and analysis process. The Time Domain Signal obtained is converted to Frequency domain using FFT Analysis.

CHAPTER 5

METHODOLOGY

Modal Analysis

The procedure for a modal analysis consists of four main steps:

- Build the model.
- Apply loads and obtain the solution.
- Expand the modes and Review the results.

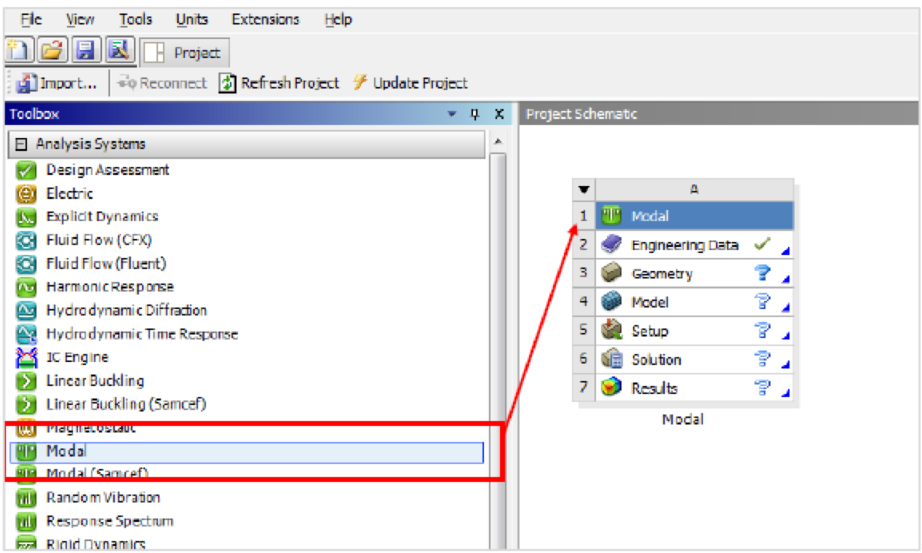


Fig 5.1: Modal Analysis Schematic in ANSYS Workbench

Step 1. Build the model

Building a model in ANSYS means importing 3D CAD model into the Geometry Section of the schematic. Models imported from CAD systems may require extensive repair if they are not of suitable quality for meshing. Due to this, Geometry repair tools are used in the Geometry section for making the model meshing suitable.

Material properties (i.e. Young's Modulus, Poisson's Ratio, and Density) of all components needs to be added in Engineering Data Section. Apply Bearing support and assigning materials to all components.

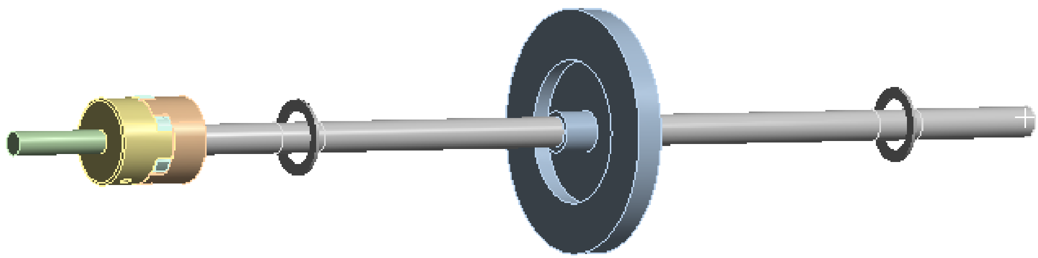


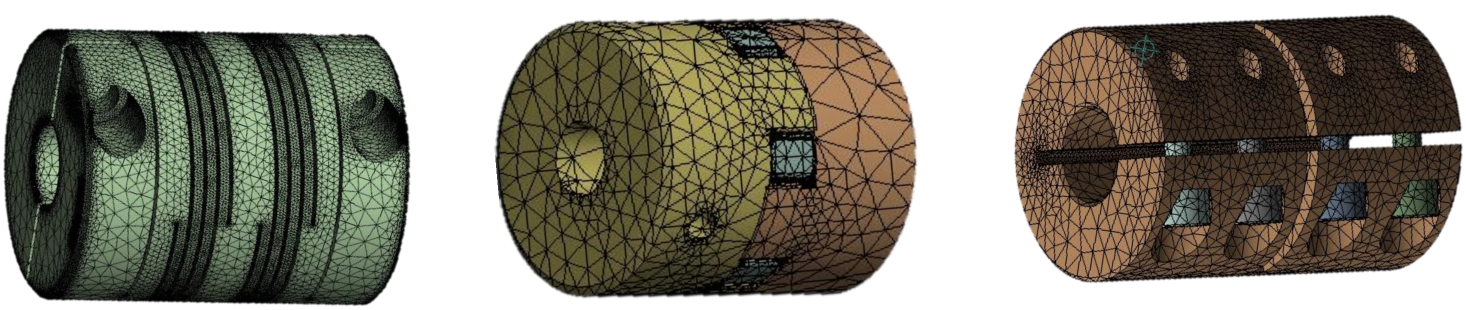
Fig 5.2: System with bearing support

- Meshing

The procedure for generating a mesh of nodes and elements consists of three main steps:

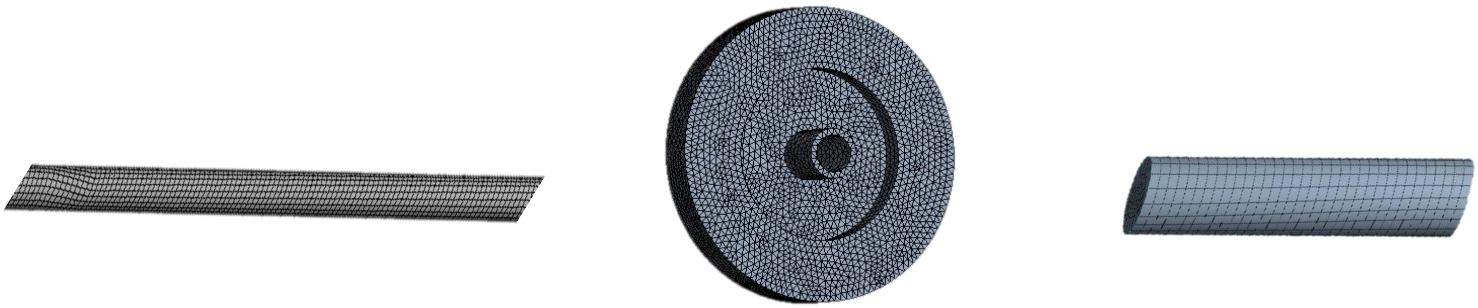
1. Set the element attributes.
2. Set mesh controls. (ANSYS offers a large number of mesh controls, which you can choose from to suit your needs.)
3. Generate the mesh.

The second step, setting mesh controls, is not always necessary because the default mesh controls are appropriate for many models. If no controls are specified, the program will use the default settings on the **DESIZE** command to produce a free mesh. As an alternative, you can use the **SmartSize** feature to produce a better quality free mesh.



Flexible Coupling | Jaw Coupling | Rigid Coupling

Fig 5.3: Meshed Couplings



Rotor Shaft Rotor Motor Shaft

Fig 5.4: Meshed Components

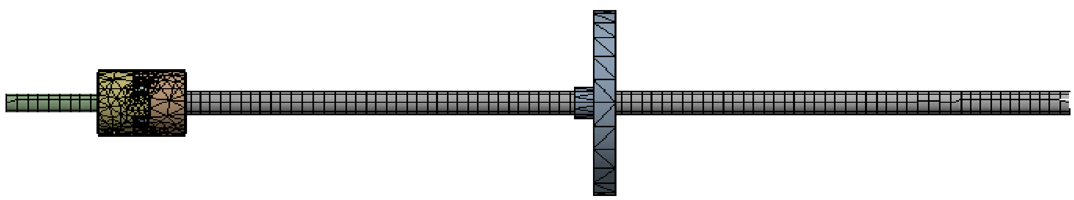


Fig 5.5: Meshed System

Step 2. Apply Load and obtain solution

Apply loads, specify load step options, and Specify the number of modes to find (default is 6). Optionally specify a frequency search range (defaults from 0Hz to 1e+08Hz). Begin the finite element solution for the natural frequencies. After the initial solution, you expand the mode shapes for review.

Step 3. Expand the modes and Review the results

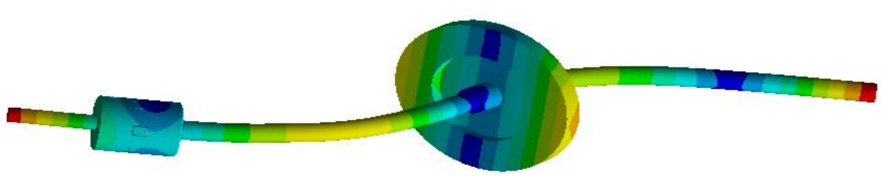


Fig 5.6: System Under Free Vibration Mode

Vibration Analysis

Before recording the signals, the shaft is checked for alignment. Also, the surface level is checked by using spirit level. The Sensitivity (i.e. 10.57mV/(m/s^2)), Accelerometer type (i.e. ICP), Sampling Frequency (2.048 kS/s), Channel no. are defined in the NVGate Software. The accelerometer should be checked before mounting at the Bearing1 by gently tapping the surface and checking the corresponding response in the NVGate software. The Accelerometer should be properly installed otherwise a loose attachment to the bearing housing will record improper signal and may distort the study of the signals.

Next, the vibration data is acquired using OROS 34 and recorded in laptop using NVGate software. A typical vibration spectrum is acquired on Bearing1 for 500 RPM (8.3Hz) to study the behavior of vibration frequency spectrum at balanced condition with minimal misalignment. The speed is then increased to 1000 RPM (16.7Hz),1500 RPM (25Hz),2000 RPM (36.6Hz) using the 1 HP Variable Speed Controller. The vibration signal is then recorded for each speed.

After acquiring spectrum for rigid coupling, the coupling is then changed and same procedure is followed for jaw coupling and flexible coupling. While changing the coupling one must tighten the bearing housings properly and should run the system at 5 RPM to check the alignment of the tighten bearing housings.

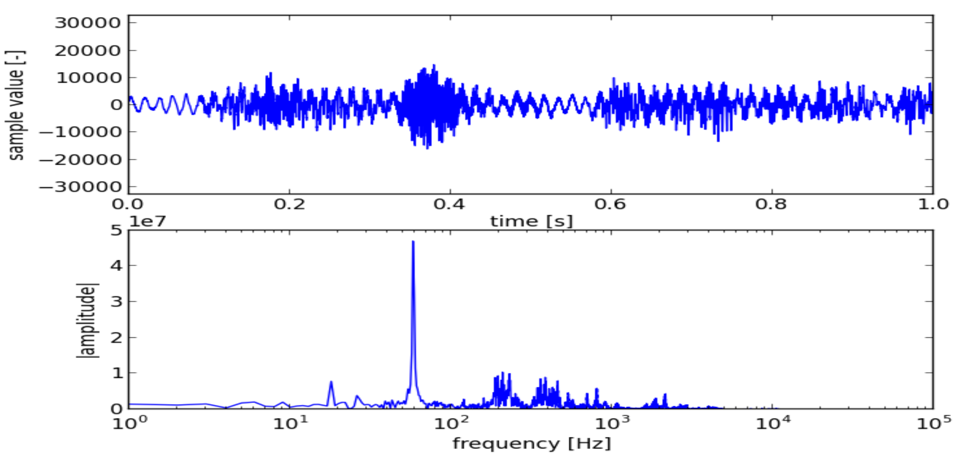


Fig 5.7: Example of a Time Domain Graph and its FFT Graph

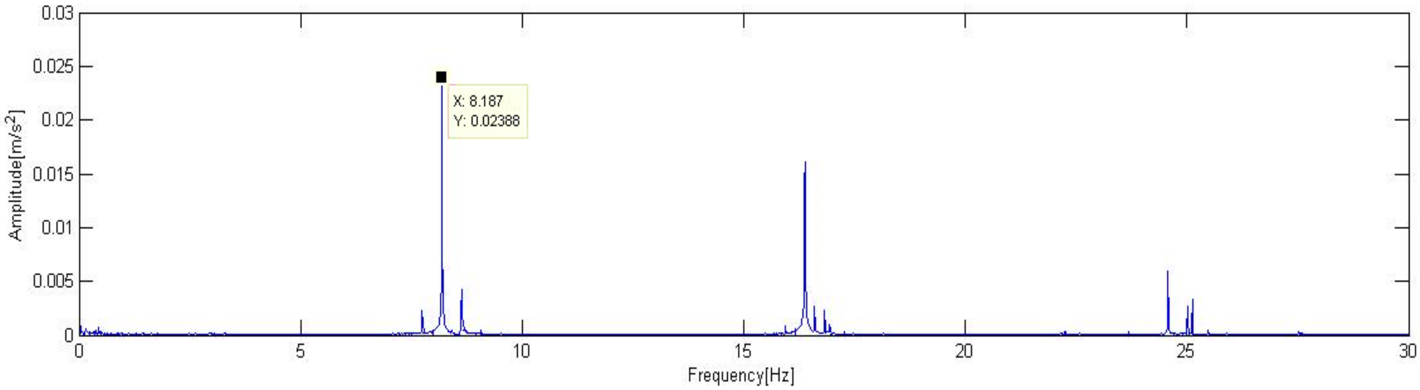
CHAPTER 6

RESULTS AND DISCUSSION

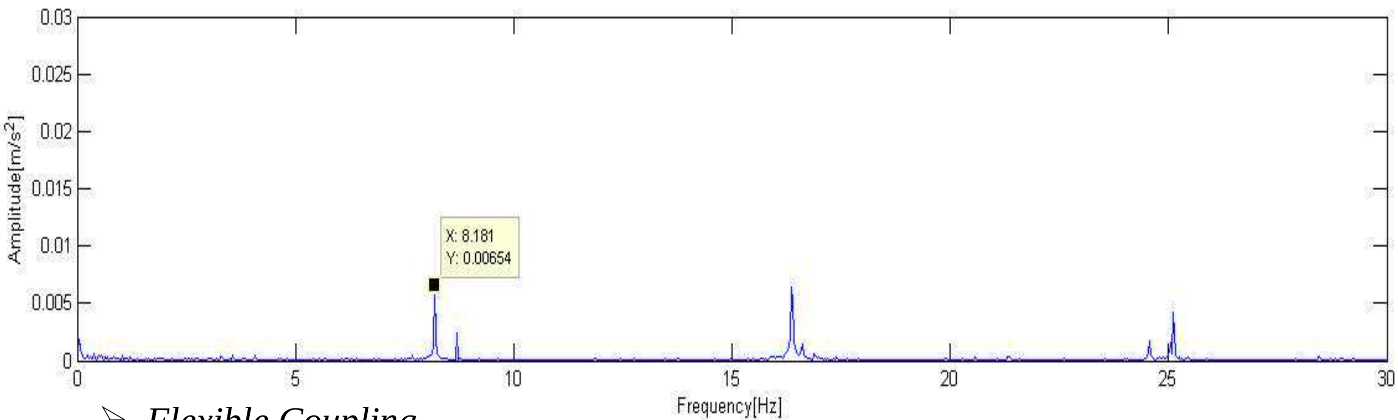
- **Vibration Analysis**

1) At Speed=500 RPM, Sampling Frequency=2.048kS/s.

   ➢ *Rigid Coupling*



   ➢ *Jaw Coupling*



   ➢ *Flexible Coupling*

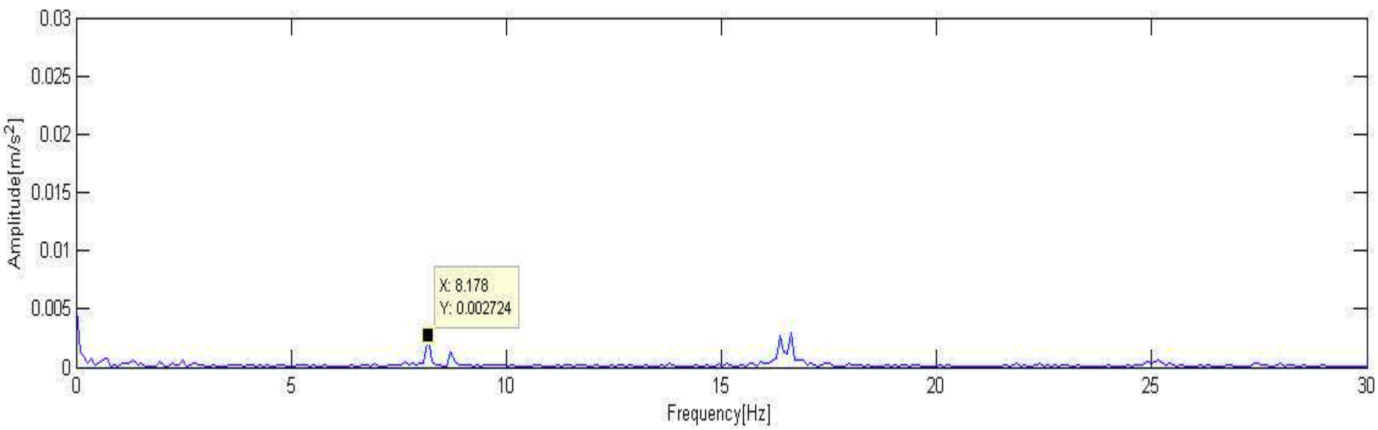
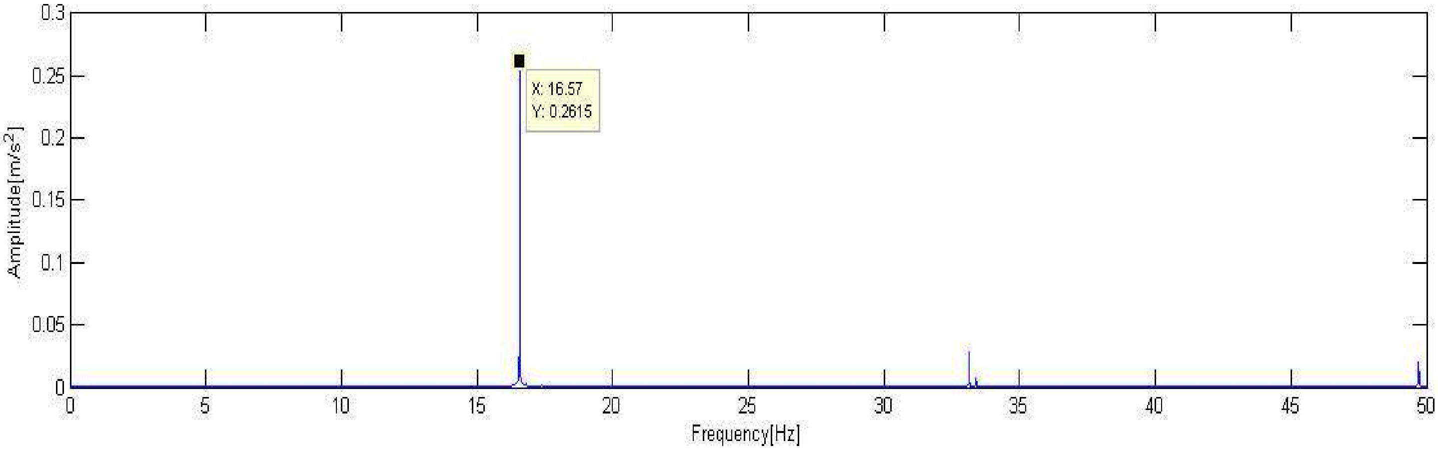


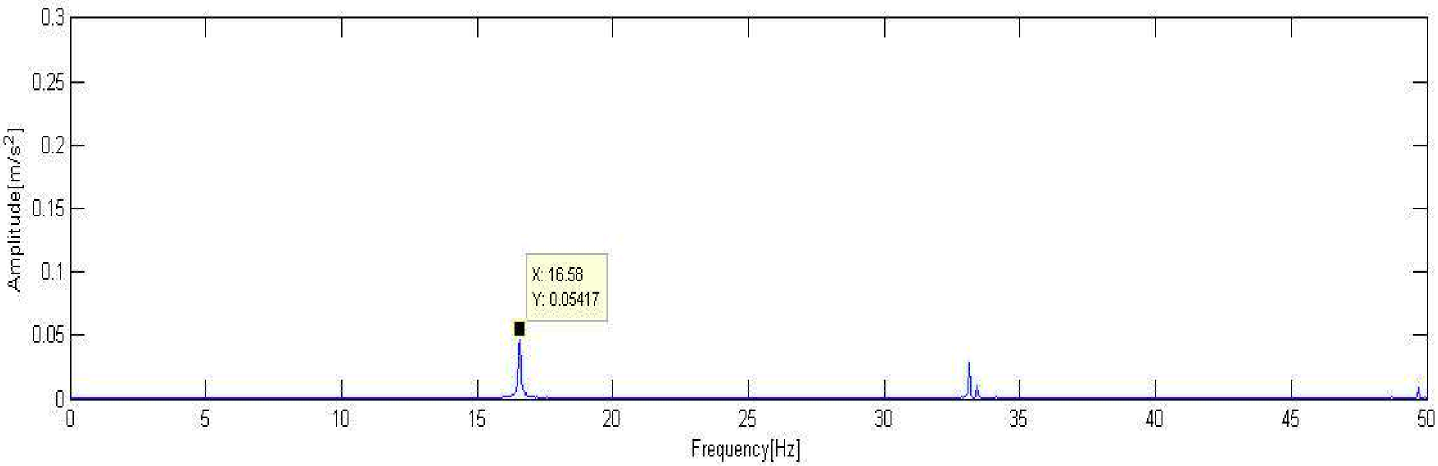
Fig 6.1: FFT Analysis for 500 RPM

2) At Speed=1000 RPM, Sampling Frequency=2.048kS/s.

- *Rigid Coupling*



- *Jaw Coupling*



- *Flexible Coupling*

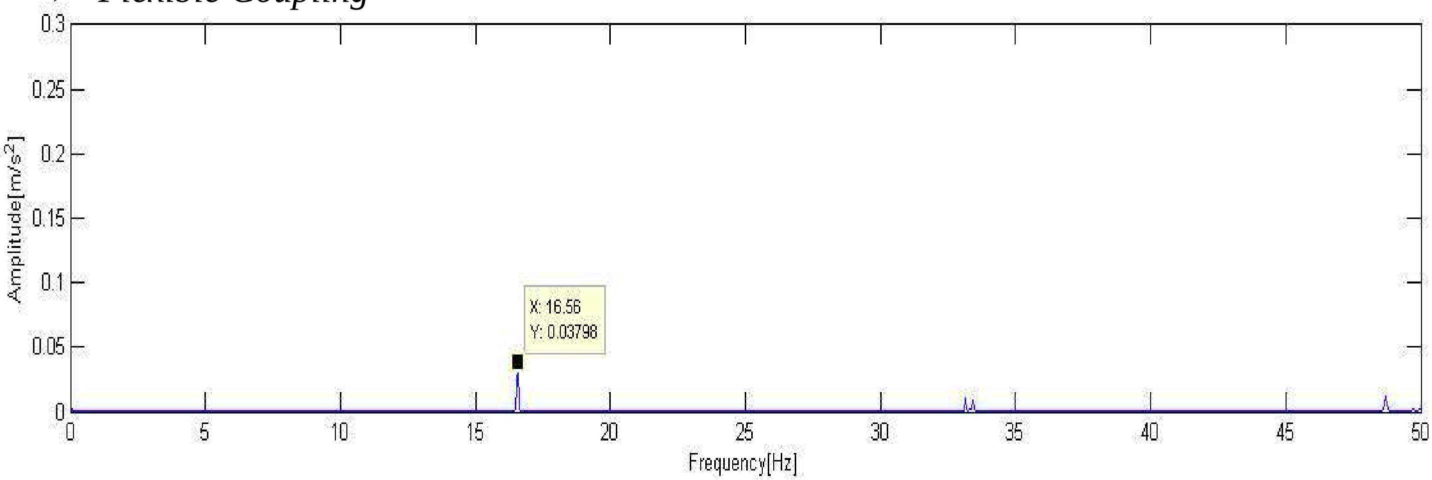
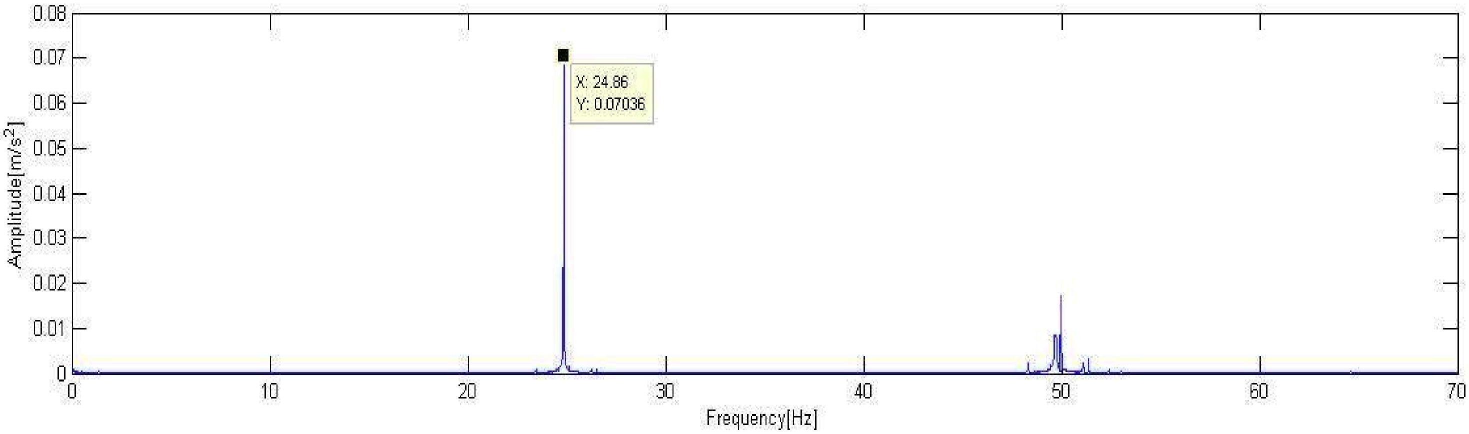


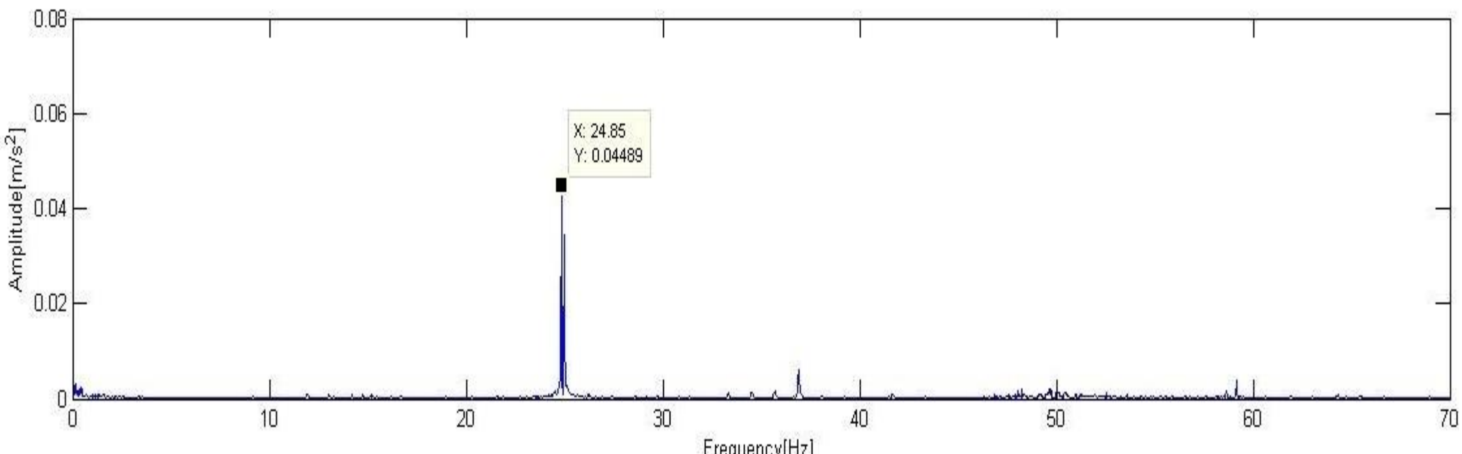
Fig 6.2:FFT Analysis for 1000 RPM

3) At Speed=1500 RPM, Sampling Frequency=2.048kS/s.

- *Rigid Coupling*



- *Jaw Coupling*



- *Flexible Coupling*

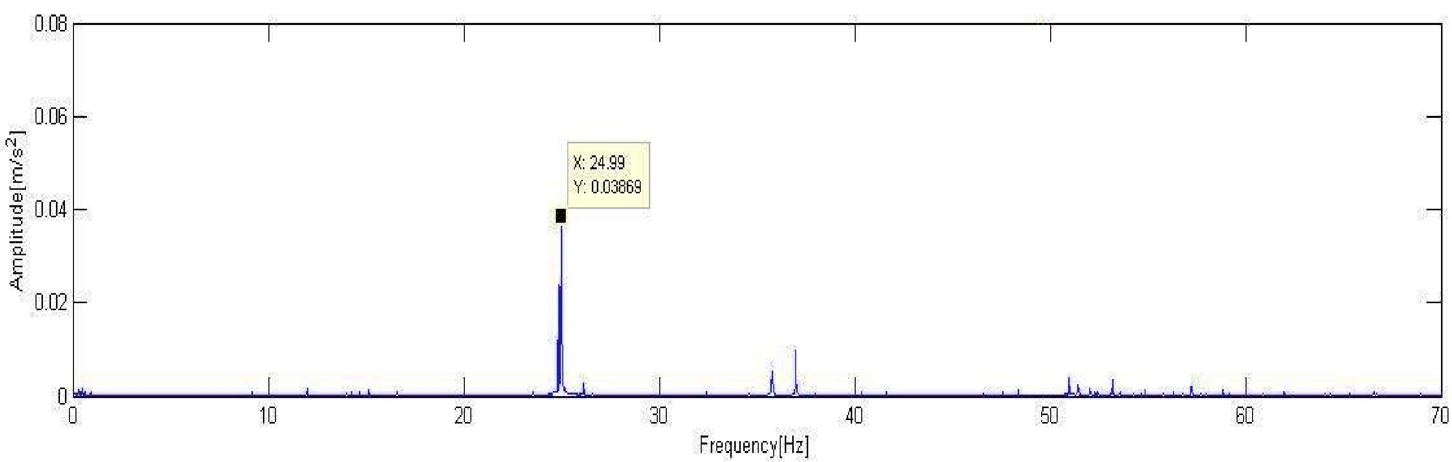
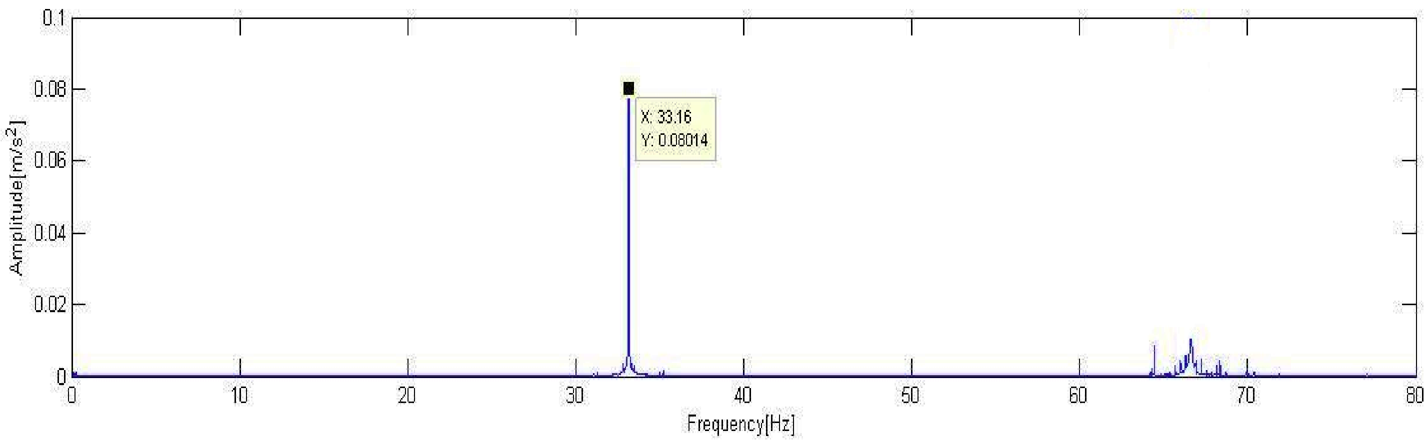


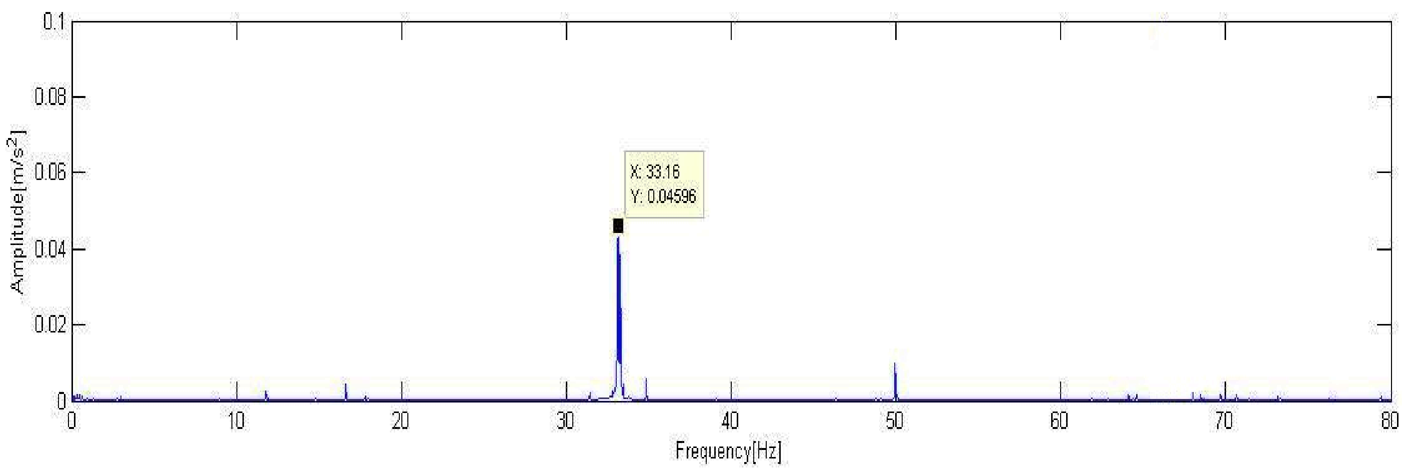
Fig 6.3:FFT Analysis for 1500 RPM

4) At Speed=2000 RPM, Sampling Frequency=2.048kS/s.

➢ *Rigid Coupling*



➢ *Jaw Coupling*



➢ *Flexible Coupling*

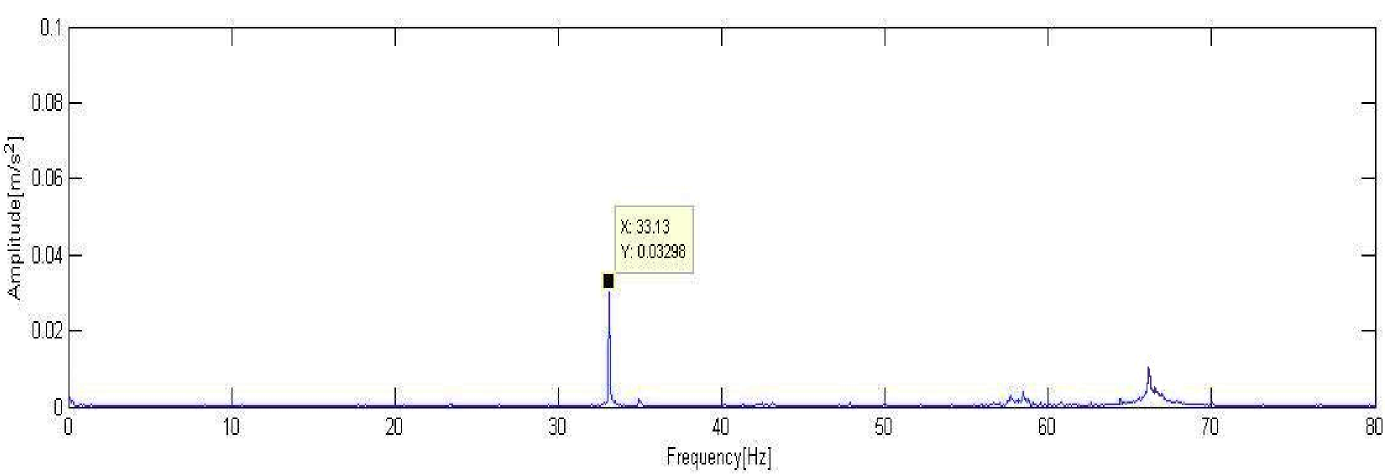


Fig 6.4:FFT Analysis for 2000 RPM

- **Modal Analysis**

System	Natural Frequency(Hz)
Rigid Coupling	20.428
Jaw Coupling	17.381
Flexible Coupling	8.0428

Table 6.1: Modal Analysis Results

Discussion

- The results of modal analysis show that with different coupling type the natural frequency of the system changes due to change in mode shape.
- The vibration amplitude of rigid coupling increases suddenly at 1000 RPM and then decreases at 1500 RPM and follows a slow increasing pattern afterwards.
- The vibration amplitude of jaw coupling increases at 1000 RPM and then decreases gradually and almost becomes constant.
- The vibration amplitude of flexible coupling is the lowest of all which increases at 1000 RPM and starts decreasing after 1500 RPM.
- The Trend Graph obtained is: -

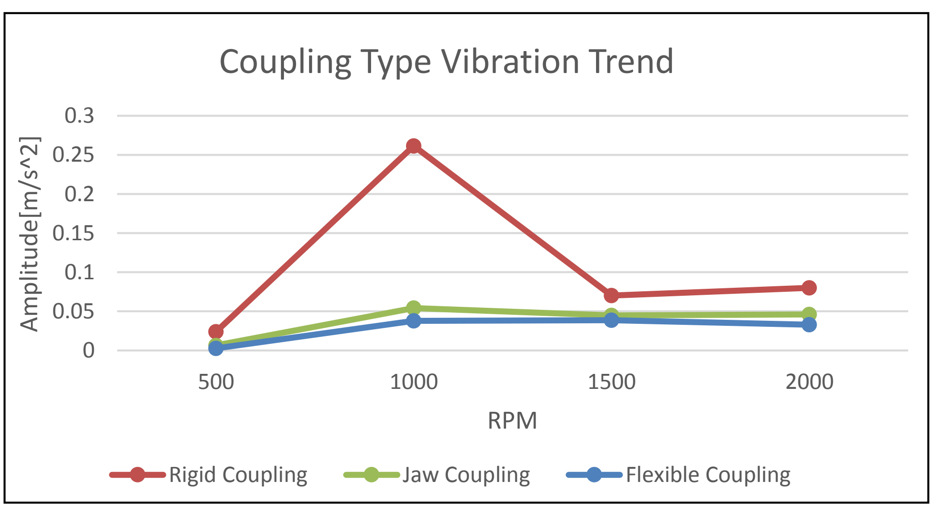


Fig 6.5: Vibration Trend Graph

CHAPTER 7

CONCLUSION AND FUTURE SCOPE

Rigid coupling is more prone to vibrations as compared to jaw and flexible coupling. In Jaw coupling, the vibration level experienced is less compared to rigid coupling but more than flexible coupling. The flexible coupling can resist the vibration due to increase in shaft speed in better way. A trend of vibration amplitude observed as Rigid>Jaw>Flexible Coupling at all 4 Speeds.

By Introducing misalignment in a controlled manner, cases with angular and parallel can be performed to get an estimate of rotor vibration trend in each coupling assembly with increase in speed and to predict which coupling is more reliable in such cases. By Introduction of mass unbalance the vibration amplitudes of all the 3 couplings can also be observed. All 3 couplings can be manufactured with the same material to conclude if the effect of coupling design prevails the effect of different coupling material on the vibration amplitude of the system.

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