

Design and Development of Test Setup to Study the Basic Procedure of Vibration analysis

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Abstract: - In past several years Vibration monitoring and vibration diagnosis of various machines has gained enormous importance which includes smaller and medium-sized machines with increasing machine frequency. For localization of defects and causes of damage to machines, a vibration diagnosis is a fantastic tool, also one can be used as an objective defense against unjustified warranty claims. At low speed, vibration can be ignored but at moderate and high speed this becomes larger and various parts of machines will not work properly as a result failure of machine parts. Fast Fourier transform (FFT) analyzer with 8 channel accelerometer probe are used for measuring the vibration amplitude. Accelerometer is a transducer which is used for converting mechanical vibration into electrical signal. The transducer output is proportionate to frequency and amplitude which describes what is wrong with the machine and relative severity of the problem respectively. By understanding the importance of vibration analysis, to study the different methods of vibration diagnosis we made a small setup, and all the methods are studied experimentally on that test setup.

Keywords – Vibration Diagnosis, FFT, Monitoring, accelerometer,

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1. INTRODUCTION

Today's global need to understand the concept of vibration and the problems arises due to it. In this paper, an attempt is made to explain the development made in an experimental setup, which can simulate the conditions of vibrations to mitigate the severity of vibrations. The said setup can be equipped with FFT analyzer which further gives the edge to vibration analysis. Fourier analysis is a term for transforming time waveform into a spectrum of amplitude vs frequency values. Fourier analysis is sometimes referred as spectrum analysis, from this amplitude and frequency values are compared with standard values of frequency.

2. LITERATURE REVIEW

Jaswinder Singh "Investigation of shaft rotor system using vibration monitoring technique for fault detection, diagnosis, and analysis" in this review he found out the areas of vibration monitoring of rotating machinery, techniques of faults diagnosis and analysis of resulting vibration signatures have been reviewed. Vibration monitoring aims to define the current condition of the machine and compare it with the previously measured condition. Some element of prediction was inferred by noting trends in the observed parameter.

S. R. Algule and D. P. Hujae "Experimental study of unbalance in shaft rotor system using vibration signature analysis" he studied about the setup which is used for measuring vibrations in terms of acceleration. In shaft rotor system rpm at 500, 1000, 1500 and 2000 of different speeds for 1x frequency in balanced and

unbalanced condition readings were taken by FFT. In which unbalance is created by unbalancing mass.

V. Hariharan and Srinivasan "Vibration analysis of misaligned shaft ball bearing system" he investigated about misalignment are the most cause of machine vibration. Understanding and practicing the fundamentals of rotating shaft parameters is the first step in reducing unnecessary vibration.

Y.V. Throat, Satish S. Kadam "Design and development of test setup for vibration analysis with soft foot" he investigated vibration phenomena causes and significance. The different possibilities of analysis are fabricated and the different element is shown.

Some of the most common machinery problems that causes vibration include: Misalignment of the coupling bearing and gears, Unbalance of rotating components, Looseness, Deterioration of rolling element bearings, Gear wear, Rubbing, Aerodynamic hydraulic problems in fan, blowers and pumps, Resonance, Eccentricity of rotating components such as v- belt, pulley or gears.

Dr. S. J. Lacey, [4] "An overview of bearing vibration analysis" he found that vibration produced by rolling bearings can be complex and can result from geometrical imperfections during the manufacturing process, defects on the rolling surfaces or geometrical errors in associated components. Noise and vibration are becoming more critical in all types of equipment since it is often perceived to be synonymous with quality and often used for predictive maintenance. In this article, the different sources of bearing vibration are considered along with some of the characteristic defect frequencies that may be present. Some examples of how vibration analysis can be used to detect deterioration in machine condition are also given.

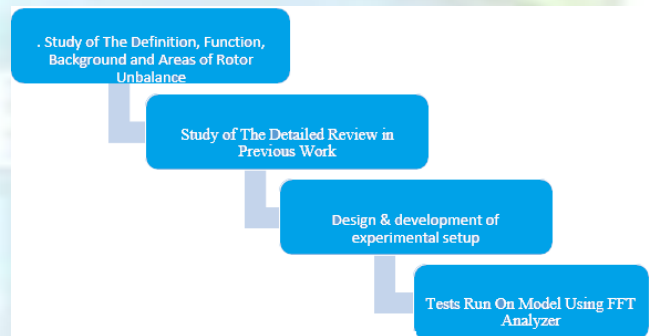
Colin Sanders [6] "Vibration analysis and associated techniques in condition monitoring" he studied that, whole machine or overall vibration occurring in the 10hz-10khz band is considered the best parameter for monitoring structural problems like imbalance, looseness, etc., and many such problems will cause excessive whole machine vibration. Measurements can either be tended to produce an ongoing evaluation of condition or the values obtained compared to the machine's normal value (ISO 2372).

Ali Waziri and Prof. M. J. Patil investigate shaft crack is a detection of a very serious matter. A crack could lead to a catastrophic failure of the rotor if not detected in a timely manner. When the system has a working crack in

shaft could be made failures. Vibration analysis is widely used to monitor the health of the rotating machinery. In this stage of the project, particular piezoelectric element, the accelerometer is used along with a device called as Fast Fourier Transform Analyzer. The cracked shaft will be vibrated using actuator and then the vibratory motion of the shaft is sensed by the accelerometer. The accelerometer will send the sensed vibration data to F.F.T Analyzer which can change the sensed data by the accelerometer to meaningful data shown on the PC, such as; frequency, Amplitude, displacement and so on.

V. P. Singh [B1], "Mechanical Vibration" R.S. Khurmi. "Theory of machine" this books give basic fundamentals of vibration and effect of stiffness and mass on the motion.

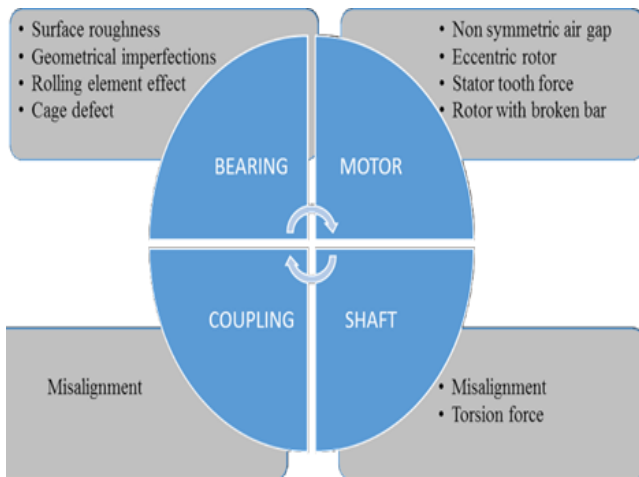
3. METHODOLOGY



1. Problem definition
2. Possible solutions to the problem
3. Action plan
4. Specify functions of the element
5. Design of machine parts
6. Manufacturing of the machine parts
7. Assembly of test set up
8. Testing and monitoring
9. Result
10. Conclusion
11. Remedial measure

For this setup, the above steps are carried out. And done all the analysis procedure.

SOURCES OF VIBRATION



In a machine there are main 4 causes of vibration, these are explained in the above chart.

TEST SETUP FOR VIBRATION ANALYSIS

To visualize the hidden details and making necessary changes before manufacturing, solid model of the experimental setup is created which gives a good insight into the project and saves the resources in the form of material and man hours.

This setup consists of rigid parts, rotating parts and transmitting part. As shown in diagram following parts are assembled together.

1. Base Plate (C-channel)-The major part of the test setup is a base plate which is made up from a c-channel. Have been taken for the base of demonstration setup. Also facilitated with no. of holes to change the bearing support position according to the requirement at various locations"

2. Bearing Base Plate- The main objective of this component is to give rigid and firm support to the bearing of the test setup.

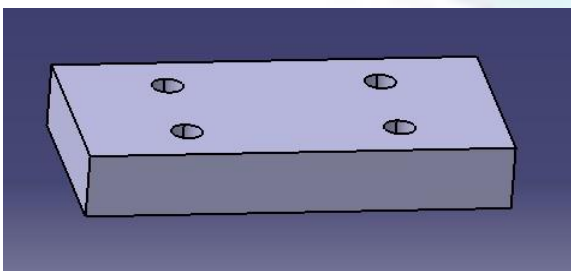


Figure [1] Bearing base plate

3. Bearing support Plate- It gives proper rest position to the bearing.

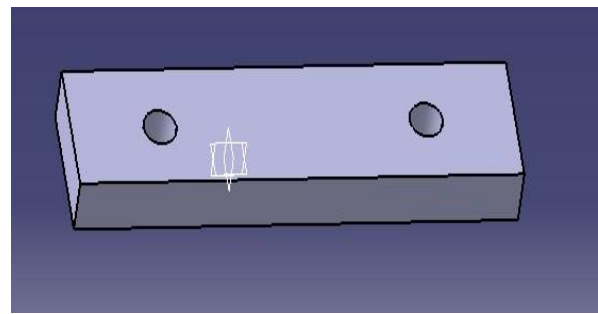


Figure [2] bearing support plate

4. Bearing- Self-aligning bearing is used to support the shaft and transmit the motion.

5. Center Disc and End Disc-

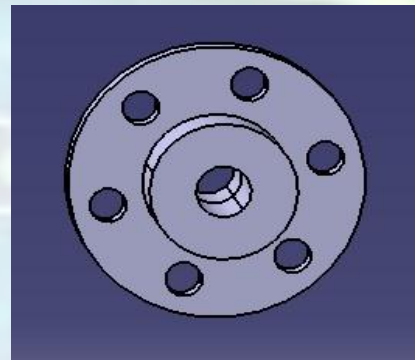


Figure [3] Center disc and End disc

6. Shaft- Shaft is coupled to the motor through the bearing at one end other to the end disc.

7. Plumber block- The bearing is mounted in plumber block.

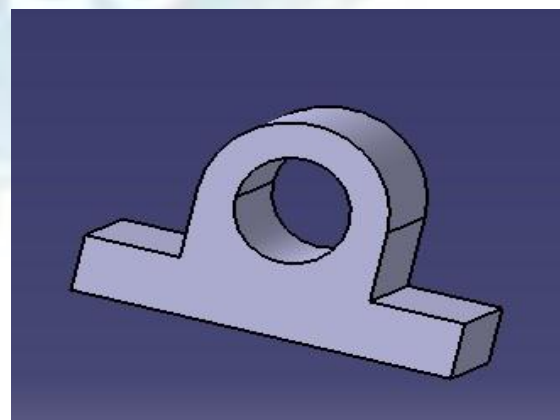


Figure [4] Plumber Block

8. Coupling- It couples shaft and motor.

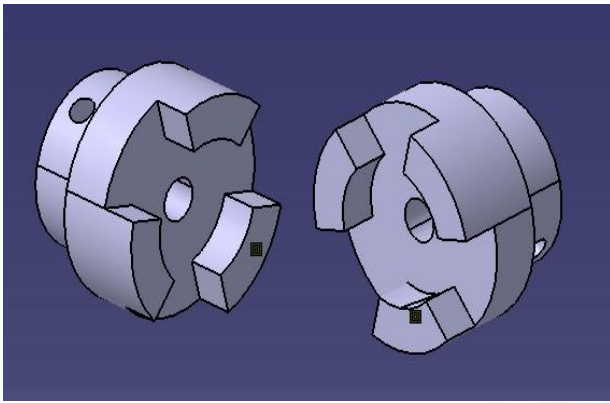


Figure [5] Coupling

4.WORKING DIAGRAM

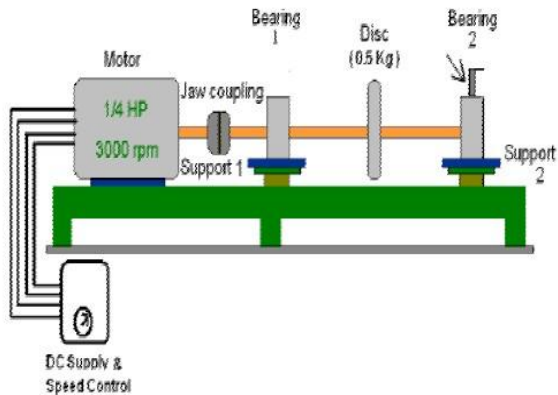


FIGURE [6] EXPERIMENTAL TEST SETUP

Working: -

The experimental setup is as shown in above figure. It consists of a 1/6 H.P. 3 phase A.C. induction motor used to drive the MS rotor mounted in the middle of the shaft having 15 mm diameter with the help of jaw coupling. VFD (Variable Frequency Drive) is used to control the speed of the motor from 0 to 2000 rpm for variable speed. Experimental set up of line diagram and photographs are shown in above figures. This set up is used for measuring vibrations in terms of acceleration. Vibration signals at Plummer block no. 1 & 2 were measured by FFT with the help of uni accelerometer at the vertical location. In shaft rotor system rpm at 500, 1000, 1500 and 2000 of different speeds for 1X frequency in balanced and unbalanced condition readings were taken by FFT.

DESIGN

Design of Shaft:

Material of the Shaft: Plain Carbon Steel 30c8

Yield Tensile Strength (Set): 400 M pa

Factor of Safety: 3

Maximum Shear Stress: $(0.5 \times S_{yt}) / FOS$

$$= (0.5 \times 400) / 3$$

$$= 67 \text{ M pa}$$

Assuming shaft transmit

1. Power = 7.33kw

2. Speed = 500 Rpm

Calculation:

$$T = 60p/2\pi N$$

$$= 60 \times 7.33 \times 1000 / 2 \pi \times 500$$

$$= 139.9926 \text{ N-M}$$

$$= 139992.6879 \text{ N-mm}$$

$$T = \pi \times \tau \times d^3 / 16$$

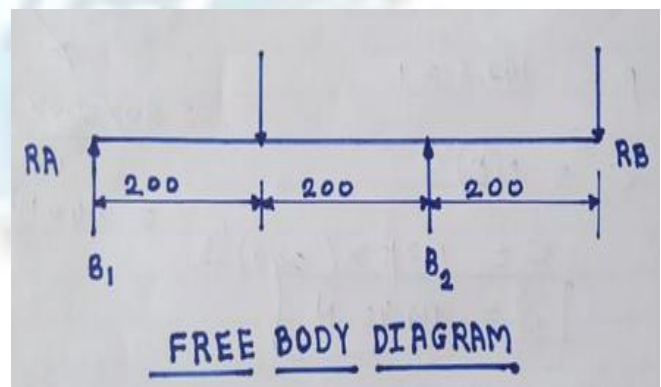
$$D^3 = 139992.6879 \times 16 / 67 \times \pi$$

$$D = 22 \text{ mm}$$

Take standard Diameter of the Shaft 25mm

$$D = 25 \text{ m}$$

5.BEARING SELECTION



1. Weight of the first disc = $2 \text{ Kg} \times 9.81 = 19.62 \text{ N}$

2. Weight of the second disc = $0.5 \text{ Kg} \times 9.81 = 4.905$

3. **FORCE RESOLUTION** = $(19.62 \times 200) - (RB \times 400) + (4.095 \times 600)$

RB = 171.675N

RA = 147.15N

4. FOR FIRST BEARING =

5. Consider as bearing life $L_{10} = 80000$ hrs.
= 240 mill/rev

6. Consider load factor = 1

7. load $P = RA = 147.15 \times 1 = 147.15$ N

8. Dynamic load carrying capacity of bearing C =

$$C = P \cdot (L_{10})^{1/3}$$

$$C = 147.15 \cdot (240)^{1/3}$$

$$C = 914.5 \text{ N}$$

From stranded table, the value of for single deep grew boll bearing $C = 3120$ N Select the bearing from stranded catalog = 61805

9. FOR SECOND BEARING =

$$C = P \cdot (L_{10})^{1/3}$$

$$C = 171.62 \cdot (L_{10})^{1/3}$$

$$C = 1063.66 \text{ N}$$

From stranded table, the value of for single deep grew boll bearing $C = 3120$ N Select the bearing from stranded catalog = 61805

DIMENSIONS

For Test Setup following parts are used and the dimensions of the assembled parts are given bellow,

1. Base Plate: 750x145x70mm
2. Bearing Base Plate: 70x140x42mm
3. Bearing:

Inner Diameter: 25mm

Center Height: 14.5mm

4. Center Disk & End Disk:

Center Disk: Flange- Ø150mm, Thickness- 7mm.

End Disk: Flange- Ø140mm, Thickness- 14mm. 4 holes of Ø15mm

5. Shaft: Ø25mm, Length-762mm.

6. Jaw Coupling with Spider

7. Coupling Cover & Protection Cage

8. Motor support: 160 x128mm

MANUFACTURING PROCESS:

After completion of design and final dimension, we started the actual manufacturing of Test setup as explained below:

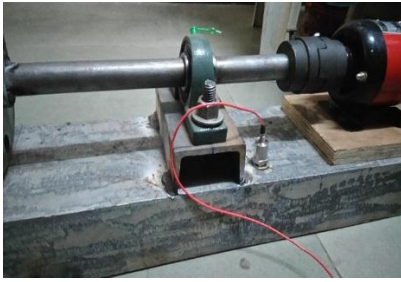
1. By using hydraulic press bending of the C-channel is removed for proper alignment purpose.
2. Grinding of the top surface of C-channel.
3. Rough dimension on the top surface of C-channel is taken for positioning of each component (Bearing, Motor, Central Disc, and End Disc, Shaft length).
4. Baseplate for a bearing and motor is made.
5. As per the dimension and position of the components 8 holes are drilled in the base plate for motor and bearing.
6. All the component is fixed as per the dimension.
7. As per procedure, final assembly of Test set-up is ready.



Fig: TEST SETUP FOR VIBRATION ANALYSIS

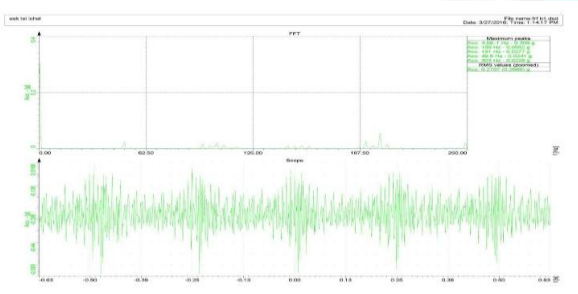
6. EXPERIMENTAL DATA

1] POSITION OF ACCELEROMETER



MOUNTING POSITION – AT BASE

Experimental Results at Base for FFT

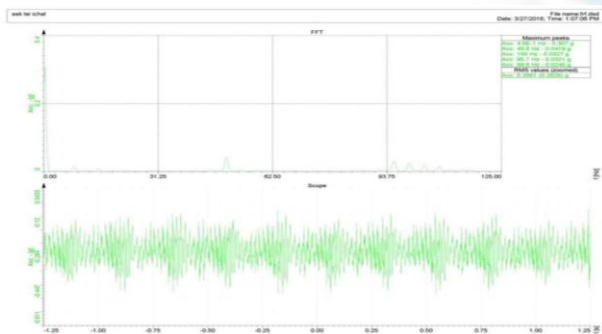


2] POSITION OF ACCELEROMETER AT BEARING 1

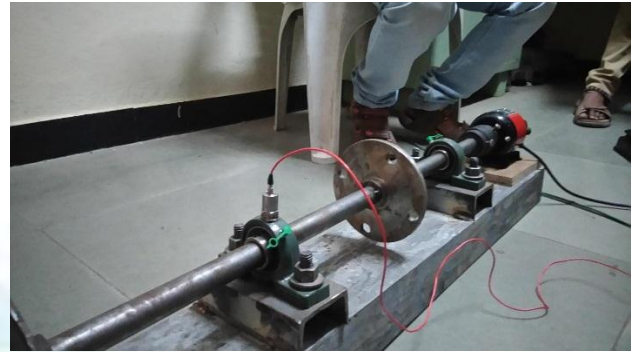


MOUNTING POSITION AT BEARING 1

Experimental Results at BEARING NO. 1 for FFT

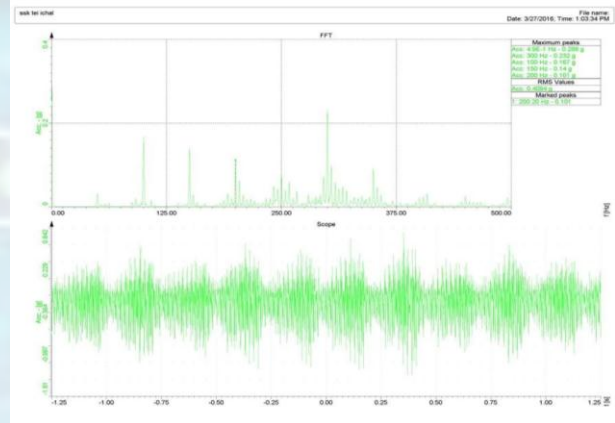


3] POSITION OF ACCELEROMETER AT BEARING 2



MOUNTING POSITION AT BEARING 2

Experimental Results of BEARING NO.2 for FFT.

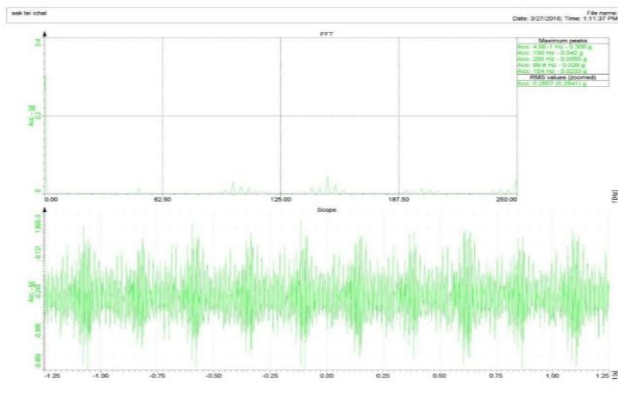


4] POSITION OF ACCELEROMETER AT MOTOR



MOUNTING POSITION AT MOTOR

Experimental Results of MOTOR for FFT



NATURAL FREQUENCY

The following Table gives the information of Natural Frequency for different Bearings

RPM	Rotational frequency (Hz)			Defect frequency (Hz)		
	ω_i	ω_c	ω_r	ω_{bp}	ω_{sp}	ω_{rp}
2298	38.3	0	14.7	306.4	191.5	78.5
1054	17.5	0	6.75	140.4	87.8	36.0
1474	24.5	0	9.44	196.4	122.8	50.3
1000	16.6	0	6.4	133.2	83.3	34.1
1078	17.9	0	6.9	143.6	89.8	36.8
1724	28.7	0	11.0	229.8	143.6	58.9

**NATURAL FREQUENCY FOR MOTOR: 50 Hz
NATURAL FREQUENCY FOR SHAFT**

mode	shaft	shaft with discs
	frequency [Hz]	frequency [Hz]
1	105.51	45.59 (T)
2	105.51	62.07
3	165.29 (T)	62.08
4	178.11	95.96
5	178.11	95.97
6	287.58	122.82 (T)
7	287.58	224.34 (A)
8	328.19 (A)	248.89
9	486.95	248.89
10	486.95	343.31
11	554.62 (T)	343.32
12	564.47	403.34
13	564.47	403.34
14	786.66 (T)	591.36 (T)
15	1004.49	645.28 (A)
16	1004.49	897.54
17	1048.25 (A)	897.54
18	1138.52	931.71 (T)
19	1138.55	1028.40
20	1448.25 (T)	1028.50
21	1536.35	1166.00
22	1536.35	1166.0
23	1594.65 (A)	1253.10
24	1653.56 (T)	1253.10
25	1699.67	1325.40 (A)
26	1699.67	1582.10 (A)

COST ESTIMATION

The total cost required for the manufacturing of test set is as bellow in BOM

Sr. No.	Part Name	No. of Parts	Price
1	C- CHANNEL	1	800
2	BEARING BASE PLATE	2	300
3	BEARING BLOCK	2	350
4	MOTOR	1	950
5	SHAFT	1	300
6	CENTER DISC AND END DISC	1	200
7	JAW COUPLING	1	700
8	MACHINING AND MANUFACTURING COST	-	400
	Total		4000/-

7.RESULTS AND DISCUSSIONS:

The Test setup is installed in the vibration lab, and for all the mounting positions of the accelerometer, the results are taken in FFT analyzer. The peak values of amplitude are seen. And from the natural frequency coming from FFT is compared with the standard natural frequencies of the components.

Which Frequency is above the natural frequency is seen, the components having greater frequency are more responsible for the vibration of the machines these are found out. And the remedial actions are taken on these. In this experiment, the motor frequency is far greater than the natural frequency of the motor, that's why soft it is suggested that the mounting of the motor should be proper, the soft foot is to be used to reduce the vibrational level. Or different mounting technics are to be used.

Sr.No	Component	Experimental Frequency(Hz)	Natural Frequency(Hz)
1	BASE PLATE	303	350
2	BEARING 1	150	133
3	BEARING 2	140	133
4	MOTOR	250	50

8.CONCLUSIONS

The following conclusions can be drawn by conducting the experiment and analyzing the recorded

data: To minimize the vibration level, Isolation is one of the best methods to optimize vibration level.

In today's industrial environment, mitigation of vibration and the subsequent noise created by it is the utmost priority. Knowing the vibration phenomena, its causes and remedial measure is the key requirement for any professional engineer. To understand this fact, the developed experimental setup will help for engineering students and professional engineers and helps improve their proficiency in the said subject.

FUTURE SCOPE:

From above result, we can see that vibrating frequency base plate and bearing is near about its natural frequency. And for the motor it's too high with respect its natural frequency. To minimize the problem of vibration we have to use the soft foot. And some future study is required.

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