Effect of Coupling Types on Rotor Vibration

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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.

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. 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. 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. 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].
M.Xu and Marangoni[2][3] have developed a theoretical model of a complete motor-flexible-coupling-rotor system using the component mode synthesis. General system of equation of motion derived for a system under misalignment and unbalance condition. The derived equations indicate that forcing frequency due to shaft misalignment are even multiple frequencies of the motor rotational speed. They also validated their theoretical results with experimental study. A.Askarian,S.M.R. Hashemi[4] studied effect of axial force, unbalance and coupling misalignment on vibration of rotor gas turbine. It is reported that for unbalance highest amplitude is at 1X of the shaft speed and for misalignment highest amplitude is at 2X of the shaft speed. V.Hariran ,P.S.S. Srinivasan[5] studied the effect of unbalance on a rotor system with flexible flange coupling. The experimental and numerical spectra were obtained for unbalance under different unbalance forces. They concluded that both experimental and numerical results show the dominant peak at 1X and experimental results are in good agreement with numerical results

1.1 Couplings

Couplings are mechanical elements that couples two drive components which enables motion to be transferred from one component to another. The drive components are generally shafts. We tend to see lot of applications of the couplings mainly in the automobiles sector, 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 it's lifetime. However, when one or more of these is not met a coupling can prematurely fail, resulting in either mahcine failure or injury to the operator of the coupled machine. 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 categorized as:-
1. Rigid Couplings
2. Flexible or Compensating Couplings
3. Miscellaneous Couplings

1.2 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. [6]

1.2.1 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. 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 .

1.2.2 Elastomeric couplings

Certain coupling types incorporate flexible elastomer elements and offer a range of torsional damping properties. But their associated characteristics - relative rotation between hubs, and backlash - can be counterproductive in precision motion control applications. Depending on the coupling type, elastomer's 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.

1.2.3 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 key-way 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 key-way, 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, key-ways are necessary for torque transmission, thus the phase relationship between coupled shafts cannot be changed. Clamping couplings dont 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.

1.3 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.

1.3.1 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: [7]

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

where [M] is the mass matrix,[Ü] 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, [8] so that [Ü] is taken to equal $\lambda[U]$,where $\lambda$is an eigenvalue (with units of reciprocal time squared, e.g., s-2), and the equation reduces to: [9]

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

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.

1.4 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.

2 Setup

2.1 Modal Analysis

For Modal analysis in ANSYS Workbench, a 3D CAD Model(Fig. 1) 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(Fig. 2 and 3) as used in the experimental setup. The Couplings were designed from data available(Table 2) from manufacturer website. Material Properties of each component of the system(Table 3) needs to be acquired to replicate the stiffness and rigid behavior of components experiencing vibration.All the dimensions shown are in mm.

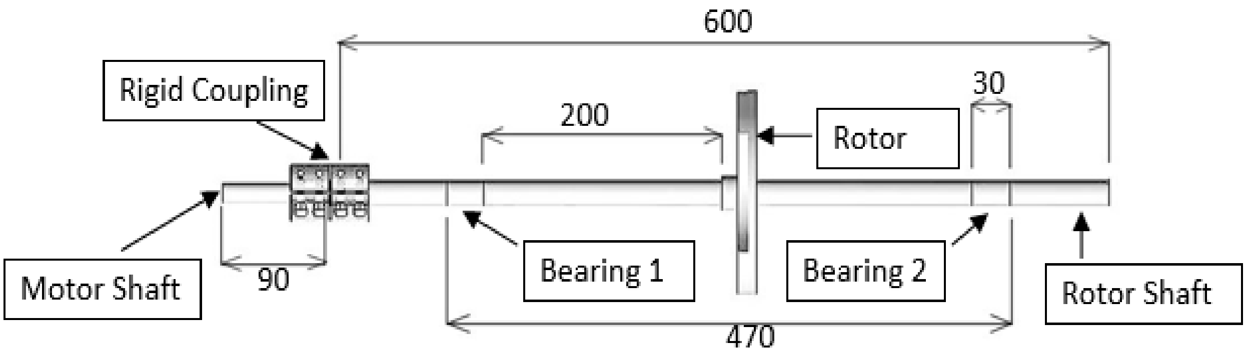


Figure 1: 2D Model of Rotor-Coupling System

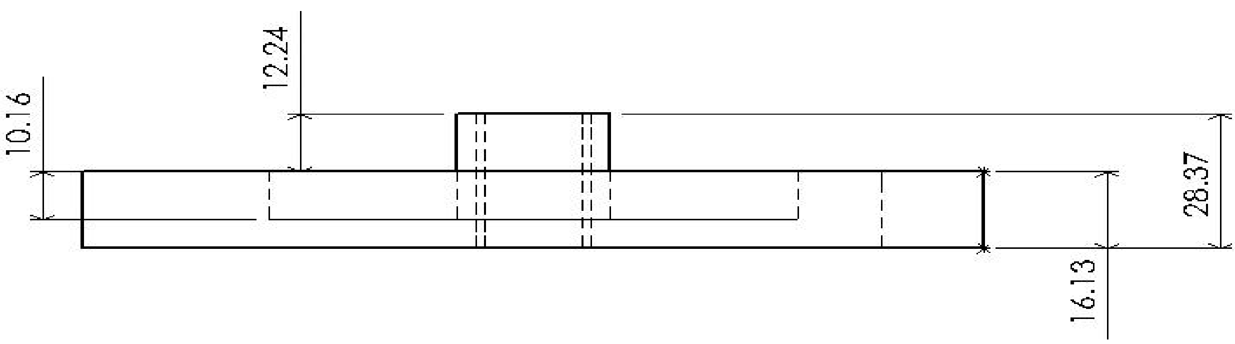


Figure 2: 2D Model of Rotor [Side View]

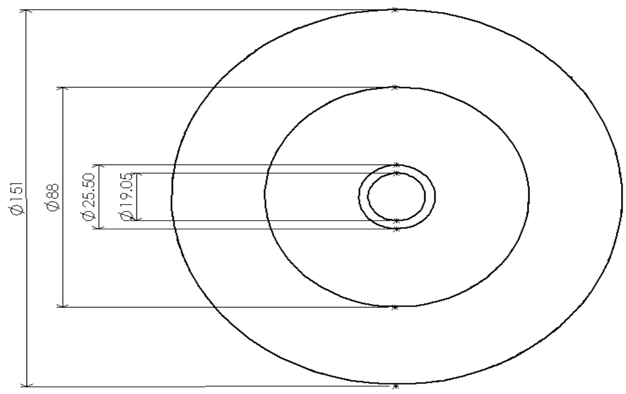


Figure 3: 2D Model of Rotor [Front View]

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

Table 1: Shaft Dimensions

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 2: Coupling Specifications

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 3: Material Properties

2.2 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.

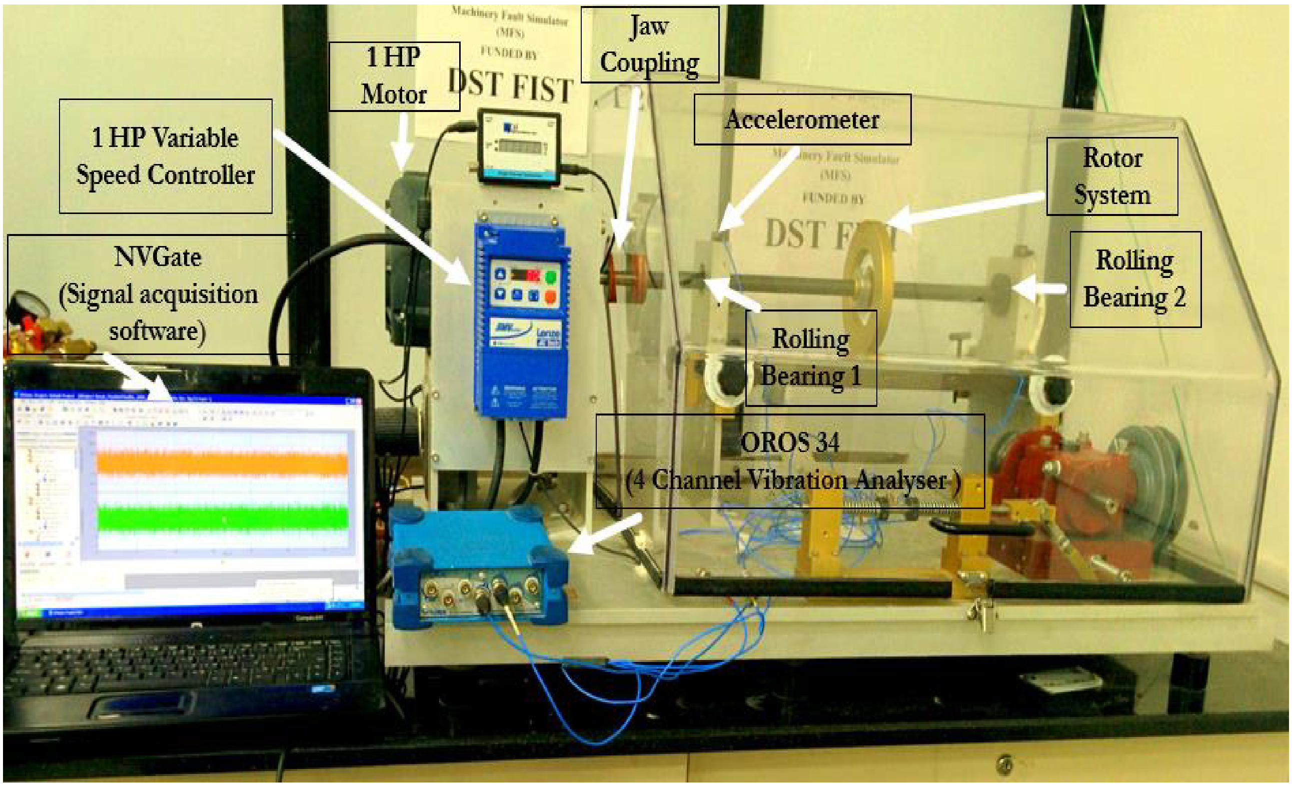


Figure 4: Experimental Setup

Fig 4 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 10.57 mV/(m/s2) 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.

3 Methodology

3.1 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.

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.

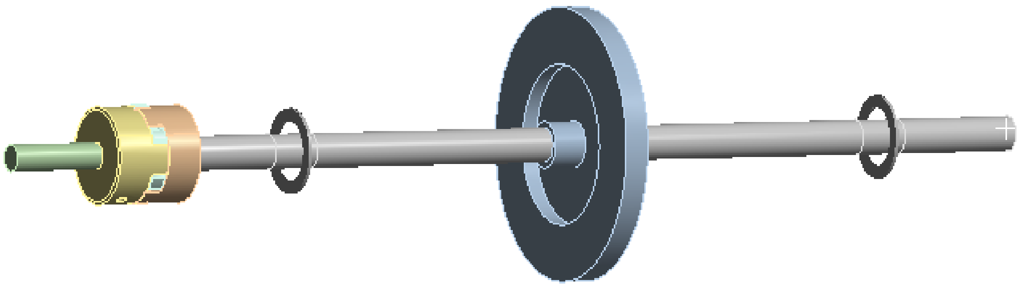


Figure 5: System with bearing support

- 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.

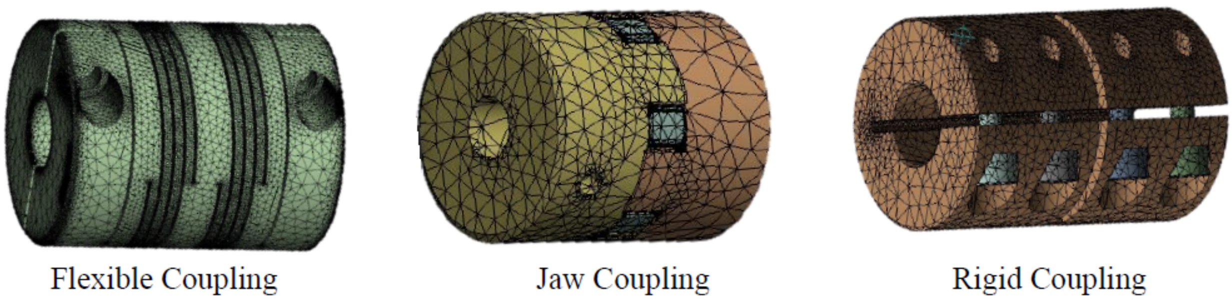


Figure 6: Meshed Couplings

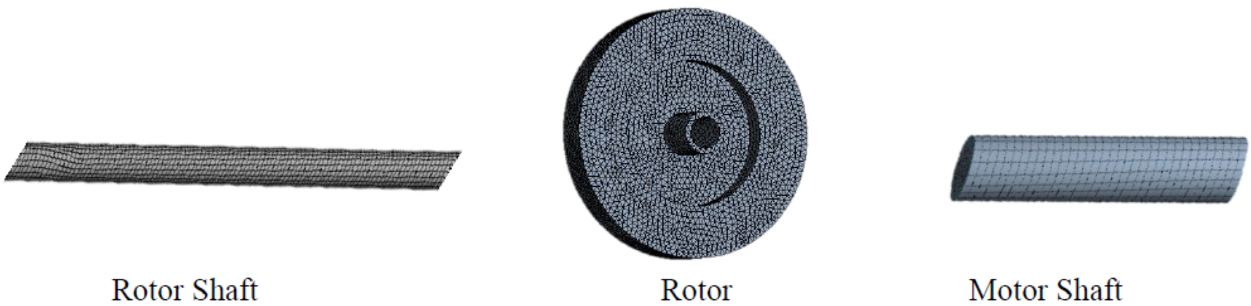


Figure 7: Meshed Components

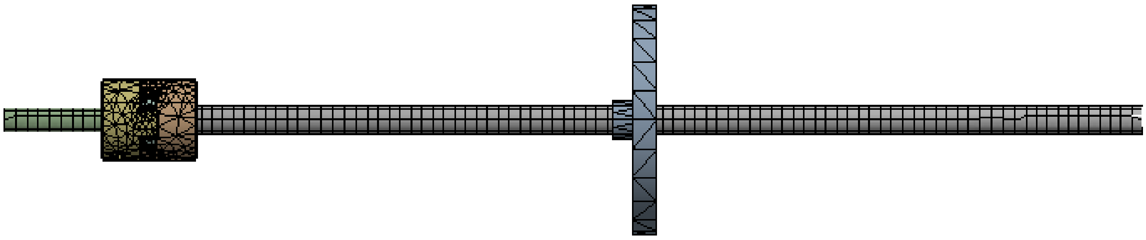


Figure 8: Meshed System

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.

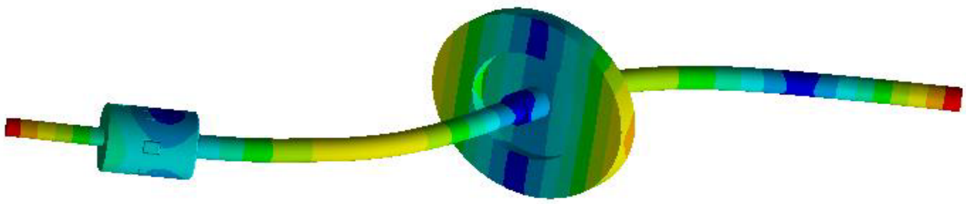


Figure 9: System Under Free Vibration Mode

3.2 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/s2)), 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.

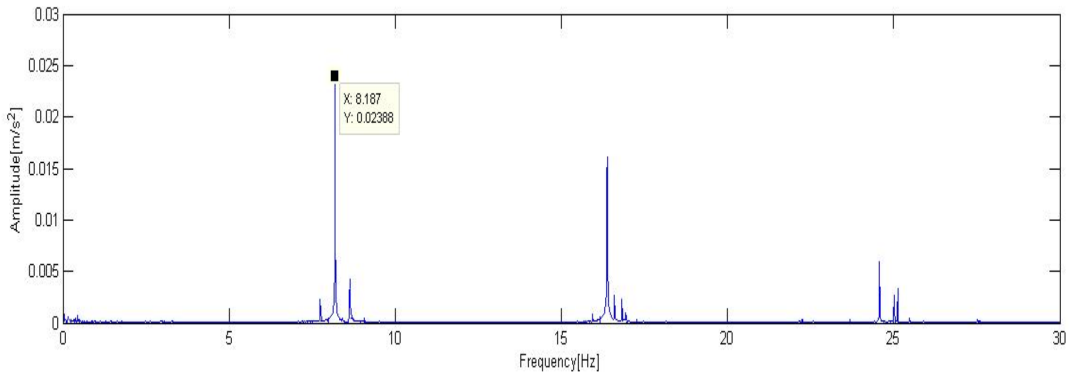
After acquiring spectrum for rigid coupling, the coupling is then changed and same procedure is followed for jaw coupling and flexible coupling. While changing the coupling one must tighten the bearing housings properly and should run the system at 5 RPM to check the alignment of the tighten bearing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

4 Results

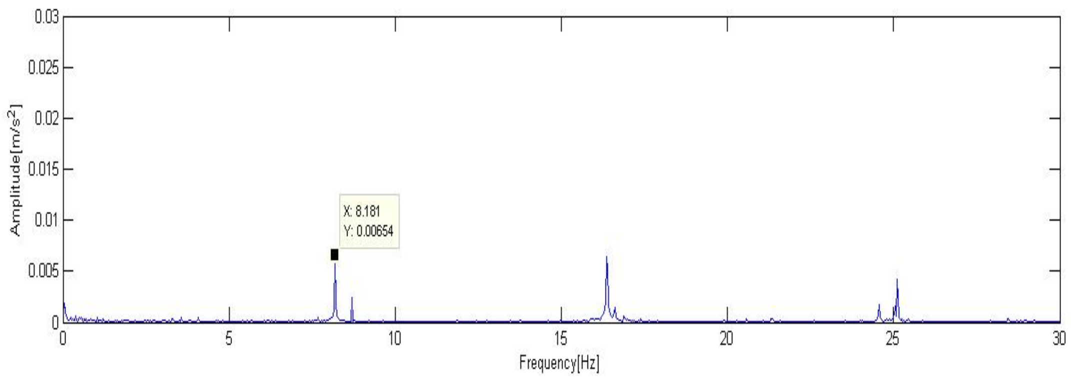
4.1 FFT Analysis

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

- Rigid Coupling



- Jaw Coupling



- Flexible Coupling

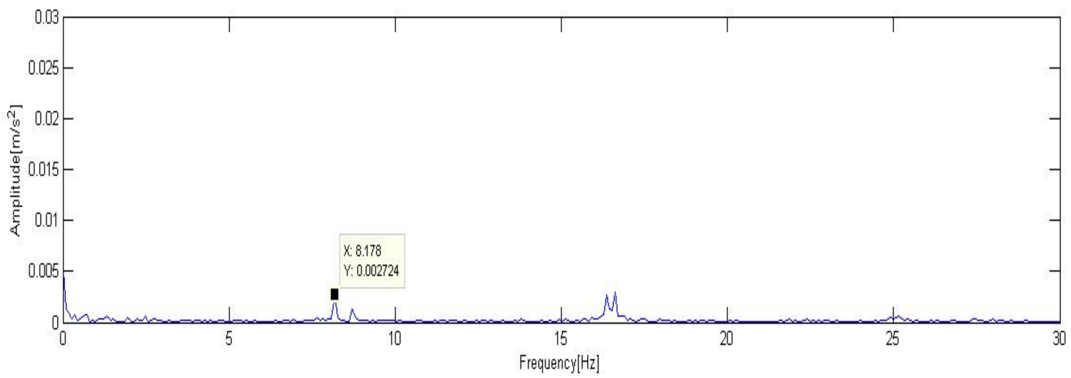
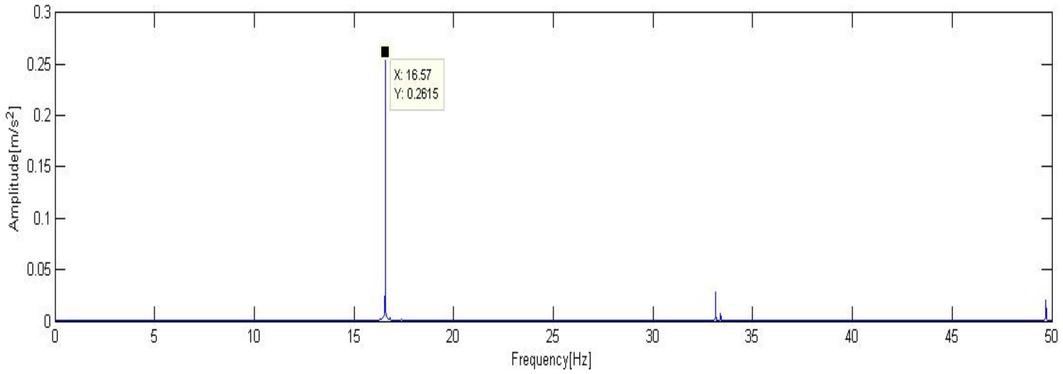


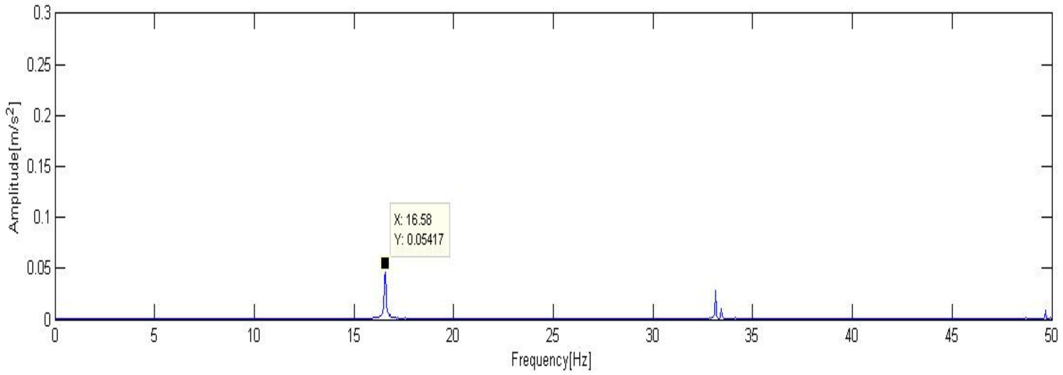
Figure 10: FFT Analysis for 500 RPM

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

- Rigid Coupling



- Jaw Coupling



- Flexible Coupling

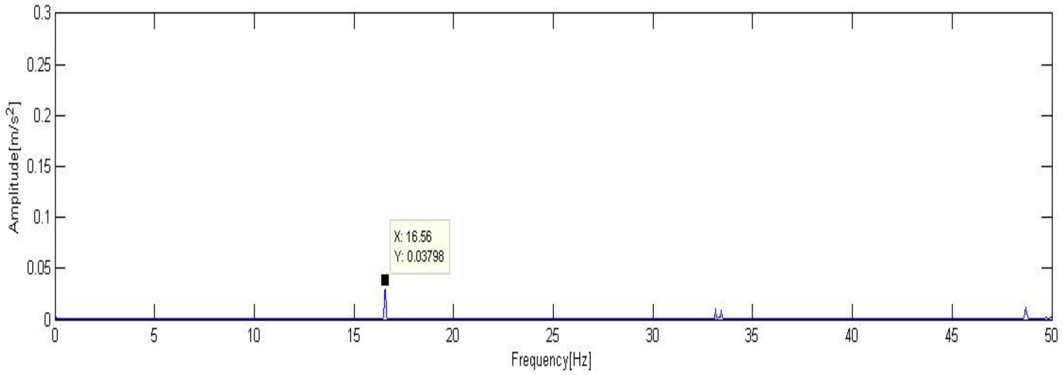
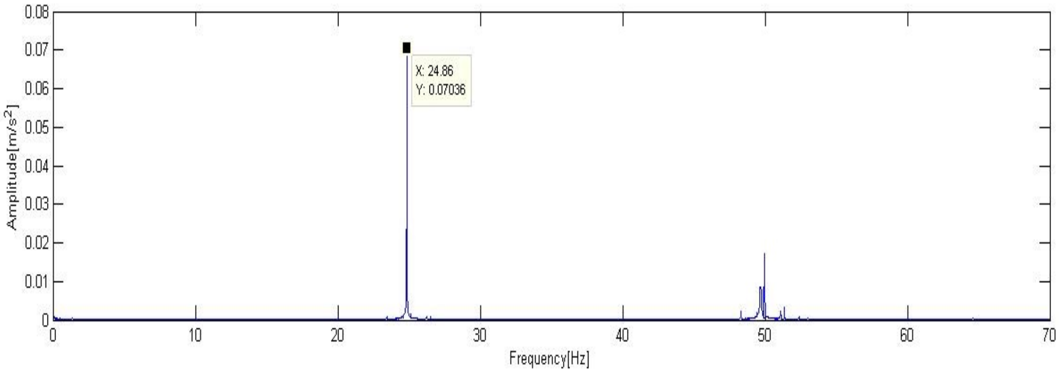


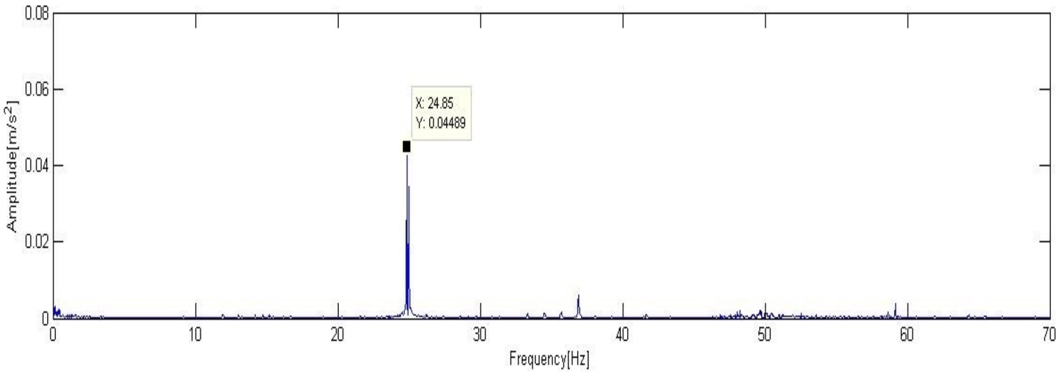
Figure 11: FFT Analysis for 1000 RPM

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

- Rigid Coupling



- Jaw Coupling



- Flexible Coupling

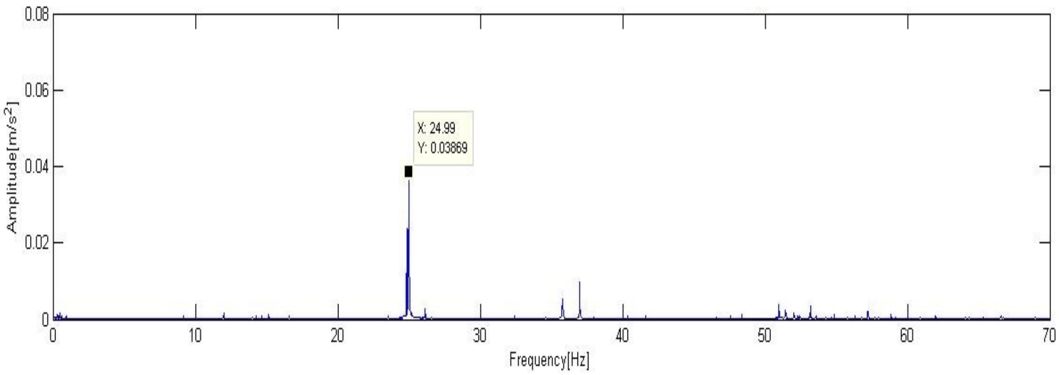
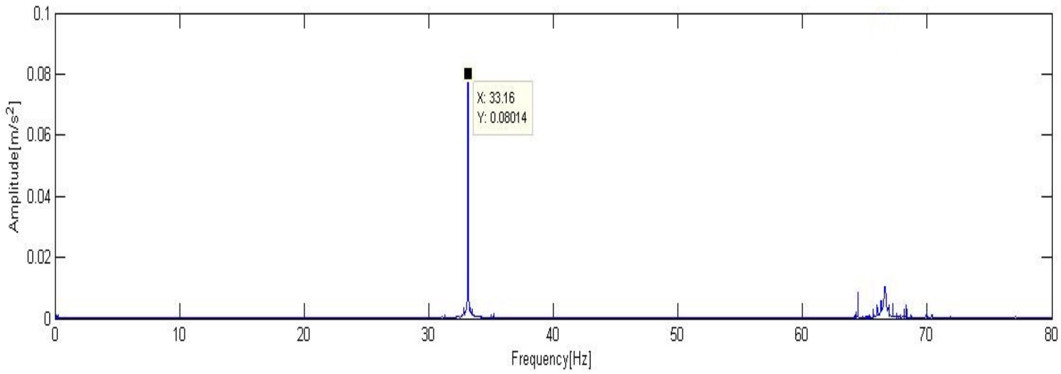


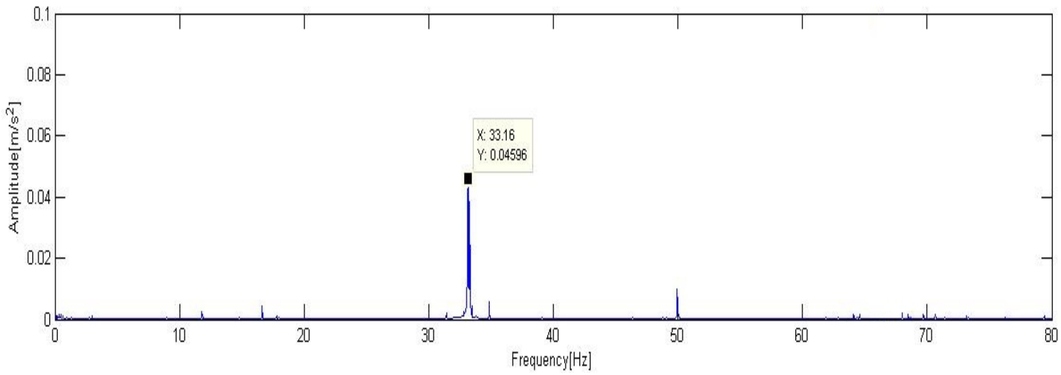
Figure 12: FFT Analysis for 1500 RPM

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

- Rigid Coupling



- Jaw Coupling



- Flexible Coupling

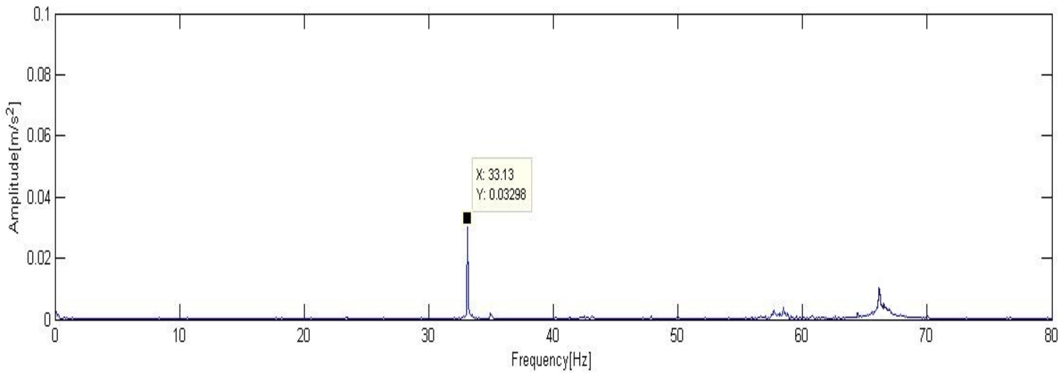


Figure 13: FFT Analysis for 2000 RPM

4.2 Modal Analysis

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

5 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.

1)The vibration amplitude of rigid coupling increases suddenly at 1000 RPM and then decreases at 1500 RPM and follows a slow increasing pattern afterwards.

2)The vibration amplitude of jaw coupling increases at 1000 RPM and then decreases gradually and almost becomes constant.

3)The vibration amplitude of flexible coupling is the lowest of all which increases at 1000 RPM and starts decreasing after 1500 RPM.

4)The Trend Graph obtained is: -

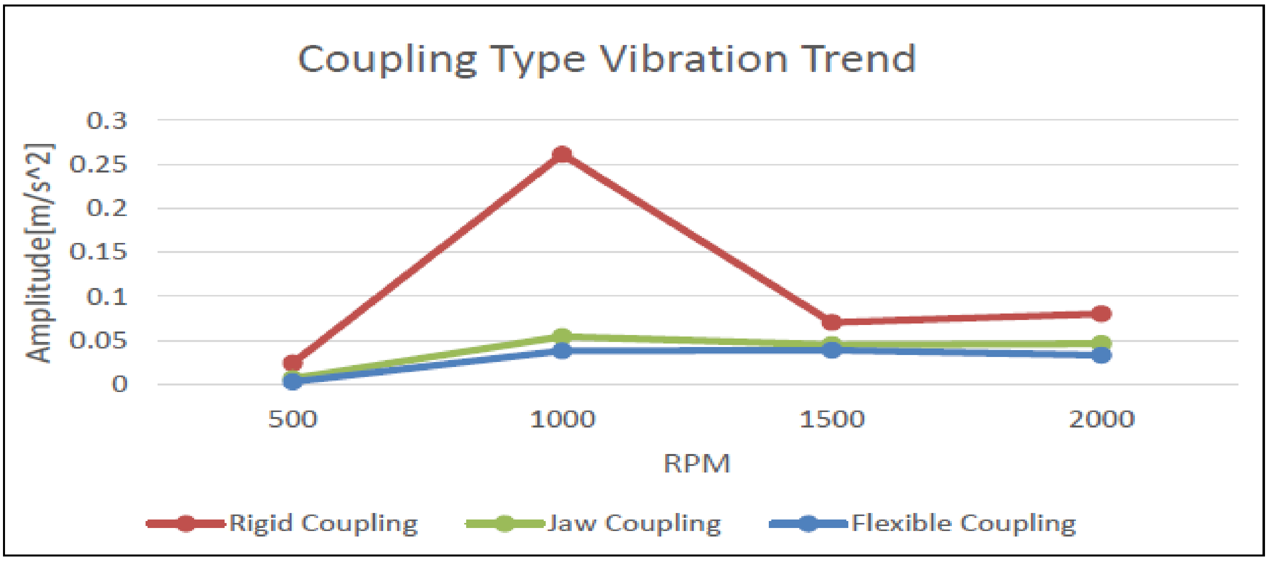


Figure 14: Vibration Trend Graph

6 Conclusion

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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