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Vibration Analysis of Misaligned Shafts

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ABSTRACT

Vibration is one of the common sources of machinery failure. Shaft misalignment is one of the sources of vibration. Correct shaft alignment gives surety of smooth, efficient transmission of power from the prime mover to the driven machine. Misalignment produces excessive vibration, noise, coupling, and bearing or shaft failure. Shaft misalignment can be divided into two components: offset misalignment, and angular misalignment. Offset (parallel) misalignment occurs when the centerlines of two shafts are parallel but do not meet at the power transfer point. Angular misalignment occurs when centerline of two shafts intersect at the power transfer point but are not parallel. Often misalignment in actual machinery exhibits a combination of both types of misalignment. In this work, vibration analysis of misaligned shafts is done by experimentation and finite element analysis. Vibration accelerations were measured using single channel vibration analyzer for baseline and the misalignment condition. The experimental results are in good agreement with the finite elements analysis results. The peak values are observed on the multiple of rotational frequency. This work will be helpful to condition monitoring of rotary machines which fails due to the misalignment between shafts. It will help for predictive maintenance and to optimize breakdown period.

***Keywords*— Condition monitoring, Flexible Coupling, Misalignment, Rotor Shaft, Vibration Analysis.**

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I. INTRODUCTION

José M. Bossio, Guillermo R. Bossio and Cristian H. De Angelo(2009) described the problem of angular shaft misalignment in motors. The load system coupled through flexible couplings is analyzed in this work. A model for the analysis and diagnosis of angular misalignment in induction motors is presented.

This work shows that the angular misalignment produces oscillations of torque and speed. A motor working under the misalignment of a shaft undergoes perturbation frequency that doubles that of rotation. Such perturbations due to misalignment are produced by the previously mentioned oscillation. In the similar way oscillation effects are also observed on the instantaneous active power consumed by the motor. For a three phase motor at constant load by the motor is constant but due to the misalignment angle, the instantaneous active power undergoes perturbations at the misalignment frequency. The effect of angular misalignment on the current spectrum shows to sidebands around the fundamental component [01].

VaggeeramHariharan and PSS Srinivasan (2011) have done experimental studies on a rotor dynamic test apparatus to predict the vibration spectrum for shaft misalignment. A self-designed simplified 3–pin type flexible coupling was used in the experiments. Vibration accelerations were measured using dual channel vibration analyser for baseline and the misalignment condition.

The rigid and pin type flexible coupling with shaft parallel misalignment is simulated and studied using the both experimental investigation and simulation. Finally the

author other concluded that experimental and simulated frequency spectra are similar, the experimental predictions are in good agreement with the ANSYS results. Both the experiment and simulation results prove that misalignment can be characterized primarily by second harmonics (2X) of shaft running speed. He also found that by using new newly designed flexible coupling, the vibration amplitudes due to the shaft parallel misalignment are found to reduce by in percentage [02].

Piotrowski. J. (2006) described importance of misalignment phenomenon as Industry worldwide is losing billions of dollars a year due to misalignment of machinery. The heart and soul of virtually every industrial operation pivots on keeping rotating machinery in good working order. Countless processes are dependent on the successful operation of rotating machines that produce electric power, fuels, paper, steel, glass, pharmaceuticals, the food we eat, the clothes we wear, the buildings we live and work in, and the vehicles that transport us across the surface of the Earth. Just about everything you see around has somehow been influenced by rotating machinery of some kind. The primary objective of accurate alignment is to increase the operating life span of rotating machinery. To achieve this goal, machinery components that are most likely to fail must operate well within their design limits. Despite popular belief, misalignment can disguise itself very well on industrial rotating machinery. He observed that the secondary effects of misalignment as it slowly damage the machinery over long periods of time. Some of the common symptoms of misalignment are as follows:

1. Premature bearing, seal, shaft, or coupling failures.
2. Elevated temperatures at or near the bearings or high discharge oil temperatures.
3. Excessive amount of lubricant leakage at the bearing seals.
4. Certain types of flexible couplings will exhibit higher than normal temperatures when running or will be hot immediately after the unit is shut down. If the coupling is an elastomeric type, look for rubber powder inside the coupling shroud.
5. Similar pieces of equipment seem to have a longer operating life.
6. Unusually high number of coupling failures or wear quickly.
7. The shafts are breaking (or cracking) at or close to the inboard bearings or coupling hubs [03].

M. LI, and L. YU (2001) found that misalignment of a gear coupling in a multi rotor system is an important problem; it can cause various faults. In this work the non-linear coupled lateral torsion vibration model of rotor-bearing-gear coupling system is developed based on the engagement conditions of gear couplings. From the theoretical analysis author concluded that the forces and moments acting on gear couplings due to the initial misalignment are from the inertia forces of the sleeve and the internal damping between the meshing teeth, and depend on the misalignment, internal damping, the rotating speed, and the structural parameters of the gear coupling [04].

A.W. Lees (2007) described that misalignment of multi bearing rotor systems is one of the most common fault conditions yet it is still not fully understood. There are numerous (and sometimes confusing) accounts in the literature asserting the presence of harmonics in the vibration signal, but no quantitative descriptions are offered. Harmonics may arise, of course, from the nonlinearities in fluid film journal bearings or from the kinematics of flexible couplings, but in this paper only rigidly coupled rotors mounted on idealized linear bearings are considered [05].

Arun Kr. Jalan, A.R. Mohanty (2009) described that vibration monitoring is one of the primary techniques of condition monitoring of rotating machines. Shaft misalignment and rotor unbalance are the main sources of vibration in rotating machines. In this work a model based technique for fault diagnosis of rotor–bearing system is described. Using the residual generation technique, residual vibrations are generated from experimental results for the rotor bearing system subject to misalignment and unbalance, and then the residual forces due to presence of faults are calculated. These residual forces are compared with the equivalent theoretical forces due to faults. The fault condition and location of faults are successfully detected by this model based technique[06].

J.K. Sinha, M.I. Friswell and A.W. Lees, (2004) found out, earlier studies have suggested that the reliable estimation of the state of unbalance (both amplitude and phase) at multiple planes of a flexible supported rotating machine from measured vibration data is possible using a single machine run-down. This paper proposes a method that can reliably estimate both the rotor unbalance and misalignment from a single machine run-down. This identification assumes that the source of misalignment is at the couplings of the multi-rotor system, and that this will generate constant synchronous forces and moments at the couplings depending upon the extent of the off-set between the two rotors, irrespective of the machine rotating speed. A flexible foundation model is also estimated.

A method to estimate both the rotor unbalance (amplitude and phase) and the misalignment of a rotor–bearing–foundation system has been presented. The estimation uses a priori rotor and bearing models along with measured vibration data at the bearing pedestals from a single run-down or run-up of the machine[07].

Mohsen Nakhaeinejad and SuriGaneriwala (2009) investigated dynamic effects of angular and parallel misalignments in rotating machinery on machine behavior. Different levels of angular and parallel misalignments were applied to the SpectraQuest Machinery Fault Simulator™ (MFS) rigid, rubber and helical beam couplings. For each case, shaft speed, motor and bearings vibrations, and bearing forces were measured. Forces and vibration signals were studied in time and frequency domain. Results indicate strong effect of coupling stiffness on vibrations and forces. Severe parallel and angular misalignments can generate low frequency modulations in vibration signals. Regarding the axial forces, higher forces with more variations can be generated in a misaligned rotor coupled with a helical beam coupling than rigid coupling. Investigating axial forces in frequency-domain reveals significant 3X and 5X harmonics for angular and 3X and 6X harmonics for parallel misalignments [8].

Jin WookHeo and Jintai Chung (2004) described that the dynamic characteristics and responses of a flexible rotating disk are analysed, when the disk has angular misalignment that is defined by the angle between the rotation and symmetry axes. Based on the von Karman strain theory and the Kirchhoff plate theory, three equations of motion are

derived for the transverse, radial and tangential displacements when the disk has angular misalignment. The derived equations are fully coupled partial differential equations through the transverse, radial and tangential displacements. The effects of angular misalignment on the natural frequencies, the mode shapes and the dynamic responses are investigated. The analysis shows that the angular misalignment causes the natural frequency split and the out-of-plane mode with only one nodal diameter and no nodal circle has the largest frequency split. It is also found that the angular misalignment yields the amplitude modulations in the transverse, radial and tangential dynamic responses.

G.R. Rameshkumar, B.V.A. Rao, K.P. Ramachandran (2012) described, shaft misalignment in rotating machinery is one of the major industrial concerns. When the power supply to any rotating system is cut-off, the system begins to lose the momentum gained during sustained operation and finally comes to rest. The exact time period between the power cut-off time and the time at which the rotor stops is called Coast Down Time. In this paper an experimental study was conducted to investigate the effect of angular misalignment in forward curved centrifugal blower test setup. Tests were conducted for various level of angular misalignment at different shaft cut-off speeds. The results show that the coast down time decreases with increase in level of angular misalignment. At higher speed and at higher level of angular misalignment, the impact on percentage reduction in CDT is very high and there is a specific correlation between the percentage reduction, cut-off speeds and the level of introduced angular misalignment.

In this work CDTs were recorded for different running speeds under normal operating conditions, and are then used as reference for analysis. The CDT decreases as angular misalignment increases, the percentage reductions in CDT, which increases with increase in angular misalignment and rotational cut-off speeds. There is a specific correlation between the reduction percentage in CDT and the level of angular misalignment with rotational speed. This experimental investigation technique provides a simple method of evaluating the effect of angular misalignment on forward curved centrifugal blower using coast down time analysis and shown great potential to use this technique to predict mechanical malfunction. The traditional vibration analysis is performed along with CDT analysis to identify the angular misalignment, find the root cause and further corrective action can be initiated to avoid serious damage and machinery failure. The industrial case study presented demonstrates how a CDT can be used as a monitoring tool to detect the shaft misalignment fault. This gives supports to the earlier findings that the shaft misalignment fault have an effect on CDTs i.e., CDT value decreases with increase in mechanical faults. Therefore it proves that CDT could be used as a diagnostic parameter in condition monitoring of industrial rotating machinery [10].

Alok Kumar Verma, SomnathSarangi and M.H. Kolekar(2013) inspects the misaligned of shaft by using diagnostic medium such as current and vibration. Misalignments in machines can cause decrease in efficiency and in the long-run it may cause failure because of unnecessary vibration, stress on motor, bearings and short-circuiting in stator and rotor windings. In this study, authors investigate the onset of instability on a shaft mounted on journal bearings. Shaft displacement and stator current samples during machine run up under misaligned condition are measured, analysed and presented here. Verification of shaft alignment is done by precision laser alignment kit.

In this paper, variation in the displacement of shaft and current samples for the study of misalignment and unbalance of shaft instability in journal bearings is presented using a spectra quest machine fault simulator. Variation in rotor displacement and stator current samples during machine run up under loading condition were measured, analysed and presented. Misalignment of shaft and especially unbalance appears in both test conducted and harmonics of both the test are also observed. Some other vibrations components occur with this experiment were also observed and the reason for these vibration components is needs to be investigated in the future study [11].

Ferrando Chacon, EstefaniaArtigaoAndicoberry, and VassiliosKappatos (2014) describes, shaft angular misalignment (SAM) is a common and crucial problem in rotating machinery. Vibration analysishas been traditionally used to detect SAM; however, it presents some drawbacks i.e. high influence of machine operational conditions and strong impact of the coupling type and stiffness on vibration spectra. This paper presents an extensive experimental investigation in order to evaluate the possibility of detecting SAM, using acoustic emission (AE) technique. The test rig was operated at under different operational conditions of load and speed in order to evaluate the impact on the AE and vibration signature under normal operating conditions. This work shows that AE technique can be used as a reliable technique for SAM detection, providing enhancements over vibration analysis [12].

II. METHODOLOGY

A. Simulation (Finite Element Analysis)

Rotary shaft, coupling and disk are modelled using CATIA V5 R19 with the same dimensions which areused in the experimental setup. The figure 1 shows the assembly. The following table I shows the dimensions used for making a component models.

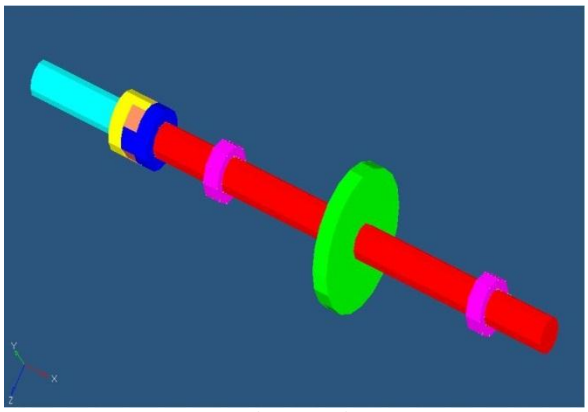

Fig. 1: Assembly of rotor shaft and coupling

TABLE I
DIMENSIONS OF SHAFT AND COUPLING ASSEMBLY

Then this assembly is imported in hypermesh software and meshing is carried out.

While doing meshing or even developing the model, it is important to decide which type of meshing is more suitable – a free mesh or a mapped mesh for the analysis. A free mesh has no restrictions in terms of element shapes, and has no specified pattern. A mapped mesh on the contrary, is restricted in terms of the element shape it contains and the pattern of the mesh. A mapped area mesh contains either quadrilateral or triangular elements, while the mapped volume mesh contains hexahedron elements. In addition, a mapped mesh typically has a regular pattern, with obvious rows of elements. In this type of mesh, first it is necessary to build the geometry as a series of fairly regular volumes and/or areas and the mapped mesh. In the present model, free mesh has been used with the element type of PSOLID. The meshed model is shown in Figure 2.

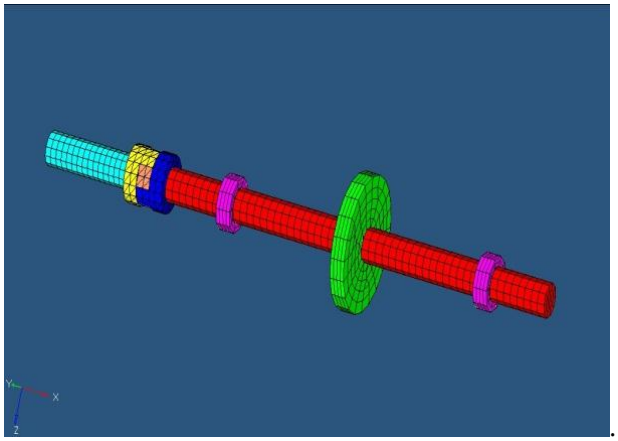

Fig. 2: Meshed Model

The table II shows the properties of material used for analysis.

The rotating shaft is supported between two identical ball bearings of 190 mm span on non-drive end and one bearing on the drive end. The bearing P 204 type is represented by COMBIN 40 element and the stiffness of the bearing is 1.5 x 10 4 N/mm.The rotor shaft model rotates with respect to global Cartesian X-axis. The angular velocity is applied with respect to X-axis. The degree of freedoms along UX, UZ, ROTY, ROTZ are used at bearing ends. Different angular velocities are given as input and corresponding accelerations are measured at the non-drive end because at this location the maximum vibrations are generated.

TABLE II
MATERIAL PROPERTIES

Properties	Brass	Mild	Rubber
Young's Modulus(MPA)	**1.05×10^5**	**2.1×10^5**	**30**
Poisson's Ratio	**0.33**	**0.290**	**0.49**
Density(kg/m^3)	**8500**	**7800**	**1100**

The solver used for the analysis is Nastran. Hypergraph is used for post processing and results are obtained.

B. Experimentation

Figure 3 shows the experimental set up used to study effects of shaft misalignment. It consists of a 3 phase AC induction motor of 0.75 kW.

Sr. No.	Description	Value
1	Input Shaft Diameter	19mm
2	Output Shaft Diameter	19 mm
3	Disk Diameter	75 mm
4	Disk thickness	10 mm
5	Coupling Inner Diameter	19 mm
6	Keyway depth	
	In shaft	4 mm
	In hub	3 mm
	Keyway cross section	
	Height	6mm
	Width	6mm
7	Bolt diameter	6mm

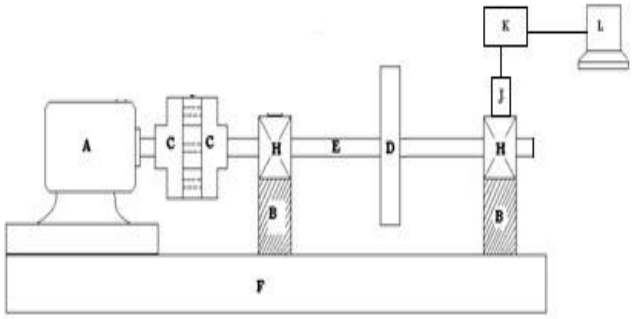

Fig. 3: Experimental setup

A: AC Motor, B: Bearing Support, C: Coupling, D: Disk, E: Shaft, F: Base Plate, H: Ball Bearing, J: Accelerometer, J: Accelerometer, K: FFT analyser, L: Computer.

The electric motor shaft is connected to driven shaft through flexible jaw coupling. The driven shaft has length 280 mm and diameter 19 mm. The driven shaft is supported between two identical ball bearings having span of 190mm. These two ball bearings are mounted on two supports. A circular disk of diameter 75 mm and thickness 10 mm is mounted on the driven shaft.

The motor end is adjustable in horizontal direction so that different misalignments between the shafts are created. Three phase A.C. auto transformer is used for voltage controlleradjust the powersupply so that motor speed can be changed.

Practical methods are developed for monitoring alignment of systems especially at static condition. Due care should be taken during the installation of all systems and their reconditioning, to ensure precise alignment between relative position of assembled system.

Major methods to diagnose alignment condition are as follows.

i) Reverse indicator method
ii) Face & rim indicator method
iii) Laser method
iv) Double radial method
v) Shaft to coupling spool method
vi) Face-face method.

Out of all above methods, face and rim method is used to correctthe shaft alignment.

C. Face and rim method

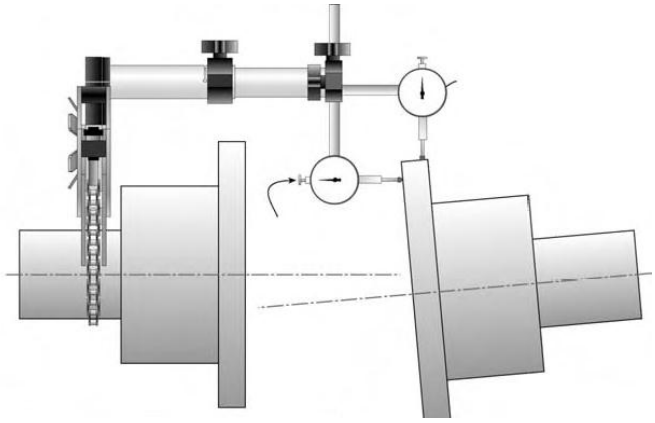

Fig. 4: Face and rim method set up

The basic setup is as shown in Figure 4. A clamp is fixtured to the rotating shaft or a part rigid with it, like the coupling hub. An extension bar is spanned to the other coupling hub where two dial indicators are attached one to take rim reading and another to take a face reading. The second shaft can be rotating or stationary better if it rotates with first shaft so that coupling imperfections are not added into the measurement. The coupling can be disassembled or not better if it is not.

The indicator is always attached or fixture onto the rotating shaft and reads on a rotating or stationary shaft. If second shaft rotates than dial indicator registers only misalignment data. If the second part is stationary, then the dial indicator measures the sum effect of misalignment plus run out of surface being read. Therefore, it is important that the surface, i.e. the coupling half, be round and concentric with the shaft axis if the second shaft is stationary. A stationary indicator reading on a rotating surface will only show run out. For this reason, the indicator fixture is never clamped to stationary shaft unless the purpose is a run out measurement.

The ideal practice is to rotate both shafts together such that the indicator tips, always read on the same spot. This way the condition of the coupling is not relevant, and in fact, it is possible to accurately alignment the shaft even if the coupling is not true or is mounted crooked. The face and rim setup takes two independent measurements. The rim reading measures parallel offset while the face reading measures angularity. Because the two conditions are measured separately, the ability exists to easily separate the two effects in the field with simple observations of the readings. This is the beauty, or simplicity of the face and rim method.

By using this method alignment is done at stationary. After this a different misalignmentsbetween the shafts are created and measured using the dial gauge indicator. The misalignments are given to the motor end at the speed range of 720 rpm to 300 rpm. Signals are acquired using accelerometer at the second bearing end atwhich maximum vibration energy is transferred. The instruments used in the experiment include dual channel vibration analyser (FFT Analyser) and accelerometer.

III. RESULTS AND DISCUSSIONS

Table III, IV and V shows the simulated vibration amplitudes in m/s^2 of non-drive end at different speeds. The numerical frequency spectra of non-drive end for parallel alignment is shown in the figure 5, for angular misalignment is shown in the figure 6 and for combined misalignment is shown in the figure 7. From this figures it is observed that peak amplitude values are multiple of rotational frequency(X). It also shows that as misalignment increases the vibration amplitudes are also increases.

TABLE III
OVL FOR PARALLEL MISALIGNMENT

Sr.No.	Misalignment 'mm'	Speed(RPM)			
		OVL (m/s2)			
		720	900	1440	2880
1	0	0.25	0.275	0.39	0.48
2	0.05	0.31	0.325	0.414	0.477
3	0.1	0.405	0.456	0.533	0.654
4	0.2	0.413	0.478	0.62	0.786
5	0.4	0.434	0.45	0.685	0.89

TABLE IV
OVL FOR ANGULAR MISALIGNMENT

Sr.No.	Misalignment 'Degree'	Speed(RPM)			
		OVL (m/s2)			
		720	900	1440	2880
1	0	0.25	0.275	0.39	0.48
2	0.029	0.284	0.296	0.453	0.55
3	0.058	0.312	0.285	0.426	0.65
4	0.087	0.35	0.33	0.489	0.766
5	0.116	0.43	0.444	0.51	0.844

TABLE V
OVL FOR COMBINED MISALIGNMENT

Sr.No.	Misalignment 'Combined'	Speed(RPM)			
		OVL (m/s2)			
		720	900	1440	2880
1	0	0.25	0.275	0.39	0.48
2	0.05 and 0.029	0.356	0.456	0.879	0.945
3	0.05 and 0.058	0.469	0.546	0.89	0.876
4	0.1 and 0.029	0.657	0.776	0.91	0.934
5	0.2 and 0.029	0.884	0.493	0.786	0.76

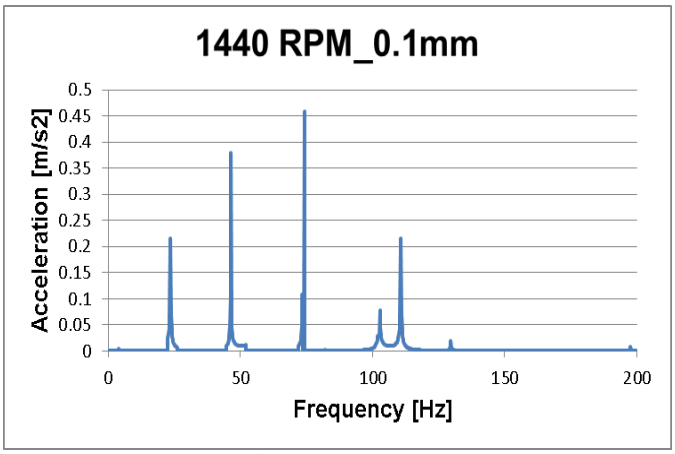

Fig.5: Vibration Spectra for 0.1 mm misalignment at 1440 RPM

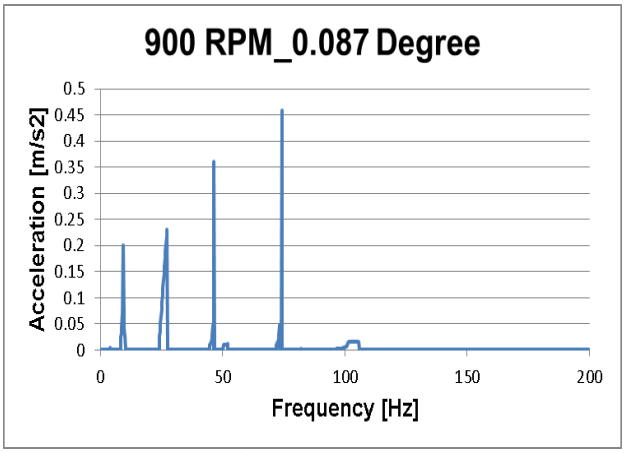

Fig. 6: Vibration Spectra for 0.07 degree misalignment at 1440 RPM

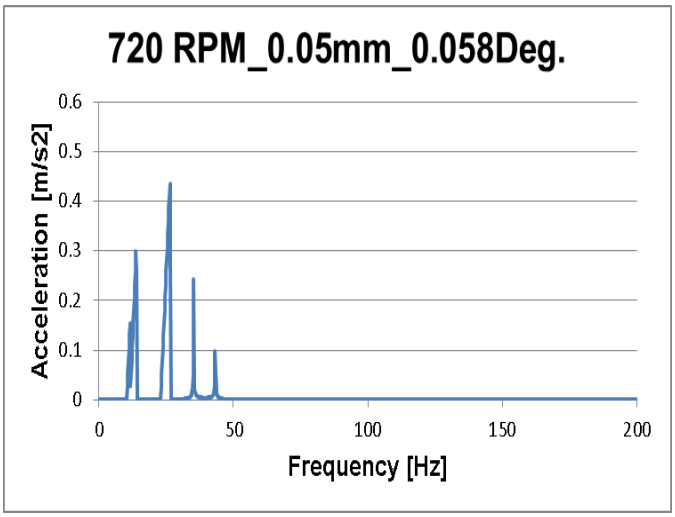

Fig.7: Vibration Spectra for 0.05 mm and 0.058 degree misalignment at 720 RPM

IV. Conclusions

In these work the effect of different misalignments was studied for different speeds by finite element method using software. It is found that the vibration amplitudes the 2X and 3X are gradually increases with increase in misalignment and rotational speed of shaft. The vibration graphs shows that the peak values are multiple of the rotational frequency(X). As vibration analysis is one of the most parameter in condition monitoring, this work will helpful for condition monitoring of rotary machines. By using one of the methods of alignment, the future failure of the rotary machine can be avoided. So this work will be helpful for predictive maintenance.

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