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Experimental Investigations of Whirl Speeds of a Rotor on Hydrodynamic Spiral Journal Bearings Under Flooded Lubrication

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Abstract. The purpose of the paper is to study Dynamic characteristics and whirl speeds of the hydrodynamic spiral journal bearing. Spiral bearings are of two types (i) Convergent Spiral Journal bearing (CSJB) and (ii) Divergent spiral journal bearing (DSJB). Short bearing is considered for the analysis which is governed by the Reynolds equation. The paper deals with stability of the rotor vibration in a spiral journal bearing. The spectra of the rotor run up and coast down are employed to evaluate a magnitude as a function of the rotor speed. The fluid induced vibration called self excited vibration occurs when the rotor rotation speed reaches the threshold speed. The bearing Dynamic characteristics and threshold frequencies have been estimated for different preload factors (m). Further, a comparative study of performance characteristics of CSJB and DSJB along with the circular bearing is carried out so as the designer can select the spiral journal bearing for experimentation. The selected bearings were tested on the test rig based on the flooded lubrication mechanism for whirl frequencies by implementing condition monitoring programs. Non-circular bearing are critical components in the equipment and hence it is needed to monitor their condition. In this paper the comparative study of parameters used for condition monitoring of bearings using time domain vibration analysis is described. The experimental result shows distinct peaks representing the presents of whips and whirls

Keywords: Non circular bearings, multi lobe bearings,

1. Introduction

In the present-day engineers interested in rotor dynamics that hydrodynamic oil film bearings play an important role in rotating machinery, because they behave like a complicated arrangement of springs and dampers and strongly influences the critical speed and imbalance response. Furthermore, Hydrodynamic oil film forces generated in the bearing gap can cause rotor instability which results in significant self-excited vibration. The analysis of the dynamic behavior of hydrodynamic bearings requires that the hydrodynamic oil film forces are related to the internal forces required to accelerate the rotor and to the additional forces due to gravity or imbalance. Hence, the authors are effectively searching new type of bearings moving from circular to non- circular bearing.

Recent test results for dry friction whip and whirl by Dara W Childs and Avijit Bhattacharya[1] deficient in predicting their observed transition speeds from whirl to whip and the associated precession frequencies of whirl and whip motion. The authors present whirl frequencies of Jeffcott- Rotor/ point-mass stator to multi rotor systems. Initial whip frequency persist s to quite high running speeds and does not transition to higher frequencies. Hiromu Hashimoto and Masayuki ochiai [2,3] described the stabilization method of a small bore journal bearing based on the combination mechanism with starved lubrication and orientation angle change. And also the same authors described the stabilization method of a small bore journal bearing using starved lubrication. The experimental setup is quite useful for condition similar experiment on other types of geometric profile bearings using the above two experiments. Jiri Tuma, jan Bilos [4]conducted the experiments stability of rotor vibration in a journal bearing. The vibration signal describing the rotor motion is a complex signal. The real part of the kind of signal is a rotor displacement in the x direction while the imaginary part is a displacement in the perpendicular direction. The self excited vibration called fluid induced vibration occurs when the rotor rotation speed crosses a certain threshold.

The present paper focuses on the unbalance responses of spiral bearings. The static and dynamic performance characteristics of the spiral bearing [5]. Considering the Short bearing, static characteristics such as load carrying capacity, attitude angle and friction parameter and dynamic characteristics such as direct stiffness and damping coefficients and Crossed couple stiffness and damping coefficients are evaluated by the Diwakar Reddy and et al. [5 & 6]. On the other hand, the authors determined theoretically the whirl speeds of the spiral journal bearing considering the stiffness coefficients of the bearing under undamped condition. These critical speeds are verified with experimental results conducted on Jeffcott rotor under flood lubrication. Also, in the present study is on the unbalance responses of the two different bearings of same preload factor i.e Divergent and Convergent spiral bearing are done and the results are compared with the Circular bearing.

2. Dynamics Characteristics

The fluid film stiffness and damping coefficients of the bearing are described by the given below matrix [7].

$$\begin{bmatrix} K_{zz} & K_{zy} \\ K_{yz} & K_{yy} \end{bmatrix} = \begin{bmatrix} 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_z r\cos(\theta)dxd\theta & 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_x r\cos(\theta)dxd\theta \\ 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_z r\cos(\theta)dxd\theta & 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_x r\cos(\theta)dxd\theta \end{bmatrix} \tag{1}$$

$$\begin{bmatrix} C_{zz} & C_{zy} \\ C_{yz} & C_{yy} \end{bmatrix} = \begin{bmatrix} 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_{z^\circ} r\cos(\theta)dxd\theta & 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_{x^\circ} r\cos(\theta)dxd\theta \\ 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_{z^\circ} r\cos(\theta)dxd\theta & 2\int_{\alpha_0}^{\alpha_0+\pi}\int_0^{L/2} p_{x^\circ} r\cos(\theta)dxd\theta \end{bmatrix} \tag{2}$$

The linearized equations of the disturbed motion of the journal center are

$$\mathrm{m}\ddot{\mathrm{y}} + \mathrm{C}_{\mathrm{ZZ}}\dot{\mathrm{z}} + \mathrm{C}_{\mathrm{Zy}}\dot{\mathrm{y}} + \mathrm{K}_{\mathrm{ZZ}}\mathrm{z} + \mathrm{K}_{\mathrm{Zy}}\mathrm{y} = 0 \tag{3}$$

$$\mathrm{m}\ddot{\mathrm{y}} + \mathrm{C}_{\mathrm{yz}}\dot{\mathrm{z}} + \mathrm{C}_{\mathrm{yy}}\dot{\mathrm{y}} + \mathrm{K}_{\mathrm{yz}}\mathrm{z} + \mathrm{K}_{\mathrm{yy}}\mathrm{y} = 0 \tag{4}$$

The characteristics equations (22) & (24) is a quadratic equation of the form

$$A_1 s^4 + A_2 s^3 + A_3 s^2 + A_4 s + A_5 = 0 \tag{5}$$

Where "s" is a complex Characteristics root. The As in equation (5) is functions of the journal mass, the stiffness coefficients and the damping coefficients. The stability margin of the journal bearing system can be established in terms of the critical mass [3,5].

3. Theoretical estimation of critical speeds

In order to study the effect of support stiffness and damping on the dynamic behavior of a rotor, Jeffcott model is considered, instead of a multi-mass model for the system [7]. This simplifies the analysis and gives fairly good results for symmetrical rotors in the fundamental mode region. We first determine the equivalent stiffness of the rotor, with the total mass of the rotor lumped at the mid span of an equivalent Jeffcott type rotor, which has the same critical speed of the system, supported on fluid film bearing at its end. In the analysis, we consider the stiffness coefficients of the bearings with undamped. For the rotor system, the following equations are adopted [7].

$$\mathrm{M}\frac{\mathrm{d}^2}{\mathrm{dt}^2}\left(\mathrm{z} + \mathrm{a}\cos(\omega\ \mathrm{t})\right) + \mathrm{K}(\mathrm{z} - \mathrm{z}_0) = 0 \tag{6}$$

$$\mathrm{M}\frac{\mathrm{d}^2}{\mathrm{dt}^2}\left(\mathrm{y} + \mathrm{a}\cos(\omega\mathrm{t})\right) + \mathrm{K}(\mathrm{y} - \mathrm{y}_0) = 0 \tag{7}$$

and

$$\mathrm{K}(\mathrm{z} - \mathrm{z}_0) = 2\mathrm{K}_{\mathrm{zz}}\mathrm{z}_0 + 2\mathrm{K}_{\mathrm{zy}}\mathrm{z}_0 \tag{8}$$

$$K(y - y_0) = 2K_{zz}y_0 + 2K_{zy}y_0 \qquad (9)$$

The frequency equations can be written as

$$\omega_1 = \sqrt{\frac{K_z(K_{zz}, K)}{M}} \qquad \omega_2 = \sqrt{\frac{K_y(K_{yy}, K)}{M}} \qquad (10)\ \&\ (11)$$

The theoretical evaluations of the above frequencies are shown in the Table – I with variation of the preload factor (m) and eccentricity ratio (ε). From the Table –I it is clear that for some of the preload factors does not have any critical frequencies. Since most of the couple stress components have negative values and therefore the frequencies are undermined and imaginary.

4. Experimental Test Rig and Methodology

The resonance of the oil whirl occurs when the fluid related natural frequency of the rotor is exited. The natural frequency of the rotating system increases as the rotor system increases and hence the excitation frequency and the natural frequency coincide over a range of rotor speed, which is the whirl vibrations continue to be active as the rotor speed increases at increasing frequency.

To predict the fluid induced unbalance responses, a Test rig is meant for predicting these frequencies of the shaft running with fluid film bearings and also observes the oil whips, oil whirls and threshold of instability. In general, the oil whips is observed below the half whirl frequencies, Whirl frequencies are observed at 1x frequencies and above which threshold of instability. A schematic of a Divergent and Convergent bearing used in the present experiment are shown in Figs. (1), respectively (The direction of the journal rotation gives the type of the bearing i.e. Convergent or Divergent). Each experimental bearing is provided with an oil supply groove. The position of an oil supply groove can be changed by changing the bearing orientation angle γ. In both convergent and divergent spiral journal bearing the oil supply position is fixed on the step.

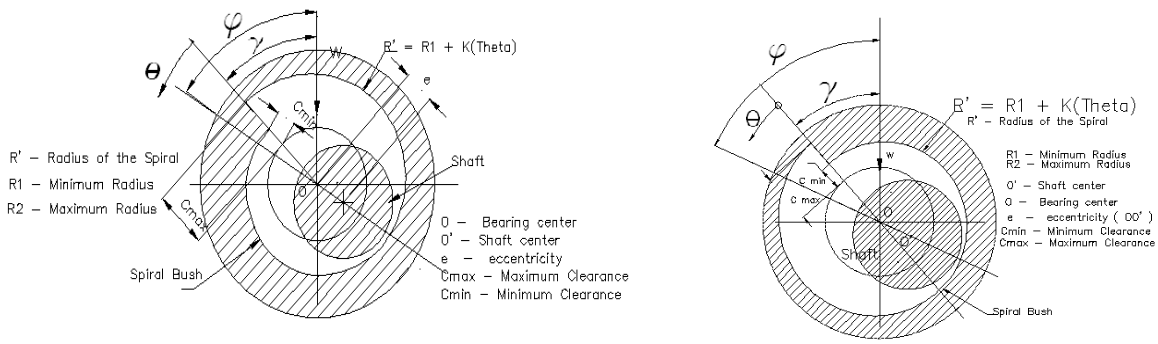

Figure (1) : Geometric configuration of spiral journal bearing

The Figure (2) shows the geometry of the experimental test rig for the small bore journal bearings used in the present experiment, along with the overall view of the test rig and the photograph of the test rig and a bearing. The rotating shaft is driven by a motor and the speed can be varied from 0 to 6000 rpm. The rotating shaft is supported by two bearings and the test bearing is installed on right-hand-side. A rotor is installed on the central part of the shaft, the mass of the rotor is 4.905N. the spring constant of the shaft is $4.44\text{x}10^4$ Nm and Natural frequency is 47.6 Hz. An oil tank is placed on the top of the bearing, and lubricating oil is supplied through a control valve. The viscosity grade of lubricating oil is that of VG32 and temperature of the oil supplied at standard room temperature conditions.

In the present experiments, vibrometer is used for measuring amplitude, RMS and Peak – Peak (Pk-Pk) with the incremental speed of 100 rpm from 0 to 3000 rpm. The amount of oil supply under flooded lubrication is $1.1425\text{x}10^{-6}m^3/s$. The amplitude, velocity and acceleration of RMS and Pk – Pk were shown with various preload factors. The preloaded factors considered for experimentation is 0.74 and there geometry is shown in the Table - I. Fabrication of lower preload factor may require high precession and increase the manufacturing cost and also the journal behaves as a circular journal bearing. AT higher preload factors the journal and bearing is having large clearances.

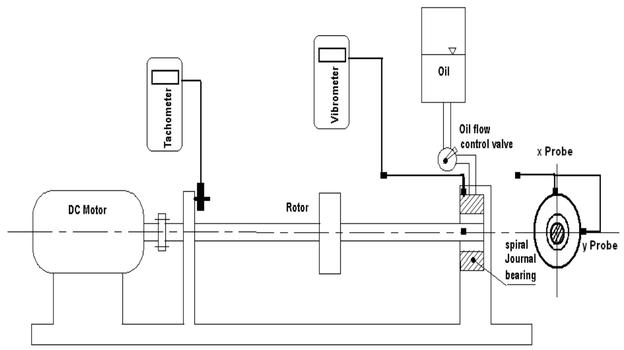

Figure – 2: Geometry of Experimental Test rig

5. Results and Discussion

Figure – 3, shows the variation of the mass parameter at different preload factors for CSJB. It is noticed that infinite stability is minimum for higher preload factor, at low Sommerfeld's numbers and also at higher Sommerfeld's number the bearing is unstable for all values of the preload factor. Also, it is observed that at higher preload factors, the mass parameter is more than the circular bearing at a particular instant. For the preload factor m = 0.7 onwards the mass parameter is infinite stable. Hence, this is the one of the reason for choosing the preload factor for experiments. At lower Sommerfeld`s number, that is, under heavy loads, the CSJB is better than the circular bearing. As the preload factor increases, clearances between the bearing and journal increases and at lower speeds the stability of the CSJB decreases. The CSJB is most stable at heavy loads and moderate speeds (lower Sommerfeld`s Number).

Figure – 4, shows the variation of the mass parameter at different preload factors for DSJB. It is observed that as the preload factor increases the mass parameter also increases. At lower Sommerfeld's number all preload factors are converging. And also the mass parameter is higher than the circular bearing. And also the Figure – 4 clearly depicts that the DSJB is less stable than the circular bearing.

In the present paper, for the analysis purpose the design parameters chosen for short bearing are as follows: Length to diameter ratio (λ) = 0.3; The value of preload factor (m) considered for the test is 0.75 and m = 0 refers to circular bearing; and the eccentric ratio (ε) which depend on preload factor (m).The Critical speeds by theoretical evaluation using equations (7 & 8) are obtained and results are shown in Table – I. The table shows for the some of the preload factors the critical speeds are unpredictable due to complex numbers and may consider stable under those conditions. These results are acceptable with the experimental results.

The archival literature is abundant in experimental and field descriptions of severe instabilities induced by fluid film bearings on rotating machinery. As an example of tests conducted at the author's laboratory on a test rig, Figures – (5 to 7) depicts recorded amplitudes of rotor motion in Peak to Peak versus operating shaft speed in a rigid rotor supported on journal bearings. The Time Wave form indices are measured; correspond to rotor motions along the vertical and horizontal positions.

The Time-Waveform index is a single number calculated based on the raw signal and used for trending and comparisons. For plotting Time Waveform index plots the following method is adopted. The raw signal whose amplitude is maximum, which is considered for a particular speed. A signal waveform of a sin wave is considered for the single maximum amplitude, for which amplitude is evaluated considering 100 observations per second at that particular speed. Such plots are not incorporated. These graphs are used to predict the oil whip, oil whirls and threshold instability.

Figure – (5), depict the amplitude, response of a rotor motion under operating Speeds for circular bearing. From these plots it is shown the three distinct peaks are observed. In which the critical speed is observed at 1350 rpm. Sub-harmonics speeds are observed at 490, 900 and 1800 rpms incase of vertical position of the probe. Whereas in case of horizontal position similar positions is observed with little variation.

Oil whirl is the form of instability that occurs at little less than half the running speed, which leads to the more violent oil whip when the oil whirl frequency coincides with the first critical speed. The main difference between oil whirl and whip lies in the fact that the oil whirl is speed dependent, while whip locks up at the first critical speed. The amplitudes response for convergent spiral journal bearing is depicted in Figure – (5). It is observed that only a single critical speed and threshold instabilities are observed. There is no sub-harmonic critical speed before a critical speed peak. This indicates the absence of oil whip in CSJB.

The amplitudes, for divergent spiral journal bearing is depicted in Figure – (6). It is observed that only a single critical speed and threshold instabilities are observed. There is one sub-harmonic critical speed before a critical speed peak. This indicates the mild oil whip in DSJB. Time domain methods use statistical analysis techniques on direct or filtered time signals to detect parametric or pattern changes as transmission components wear. The crest factor (CF), is a simple measure of detecting changes in the signal pattern due to impulsive vibration sources. Figures – (8 to 10) shows the one of the parameters crest factor for displacement amplitudes are used for three different bearings. The Results show that there is significant difference between the circular and non-circular bearings.

Table - 1 Circular, Convergent & Divergent Spiral Journal bearing theoretical critical speeds

Convergent Spiral Journal bearing

	first Critical Speed					Second Critical Speed				
m	**0.1**	**0.3**	**0.5**	**0.7**	**0.9**	**0.1**	**0.3**	**0.5**	**0.7**	**0.9**
e = 0.1	764.21	#NUM!	#NUM!	1108.70	1739	1085.09	#NUM!	3106.06	2174.15	2032.62
0.2	822.97	#NUM!	#NUM!	#NUM!	1627	1059.04	#NUM!	#NUM!	793.19	1903.62
0.3	909.55	#NUM!	#NUM!	#NUM!	#NUM!	1037.48	#NUM!	#NUM!	#NUM!	480.42
0.4	1014.67	#NUM!	#NUM!	#NUM!	#NUM!	1025.09	#NUM!	#NUM!	#NUM!	58.55
0.5	990.18	#NUM!	#NUM!	#NUM!		1174.82	#NUM!	#NUM!	#NUM!	
0.6	964.80	#NUM!	#NUM!			1373.49	#NUM!	#NUM!		
0.7	939.83	#NUM!				1657.59	#NUM!			
0.8	917.85					2131.46	**#Num! – indicates complex number**			

Divergent Spiral Journal Bearing

	first Critical Speed					Second Critical Speed				
m	**m = 0 Circular bearing**	**0.1**	**0.3**	**0.5**	**0.7**	**m = 0 Circular bearing**	**0.1**	**0.3**	**0.5**	**0.7**
e = 0.1	563.96	602.18	642.12	3478.01	#NUM!	771.48	#NUM!	666.55	4387.18	#NUM!
0.2	598.07	633.95	689.98	790.37	760.79	764.72	1616.02	792.25	803.39	873.39
0.3	653.25	692.62	757.29	833.51	923.69	753.91	864.76	802.22	875.19	979.91
0.4	728.84	745.32	790.38	816.43	923.44	739.62	4387.18	864.76	1021.87	1206.82
0.5	722.63	725.02	767.25	776.786	877.40	826.78	1246.20	1017.17	1246.20	1616.02
0.6	703.65	700.74	736.33	717.80	#NUM!	954.08	760.34	1237.03	1616.74	
0.7	683.40	672.92	698.06	3478.01		1128.0	1249.16	1599.64		
0.8	662.53	640.26	642.12			1393.2	1601.22			
e = 0.9	641.53					1900.7				

6. Conclusions

The experiment is conducted in the investigation phase. It is observed that the primary critical speeds are maximum in the case of plain circular journal bearing and DSJB. When compared with the amplitude there is reduction of 28% in CSJB. The secondary critical speed is almost acting at the same speed for both circular and CSJB. But the amplitude is less in case of CSJB compared with the circular bearing (around 33%).All these factors leads to more stability in the case of CSJB.

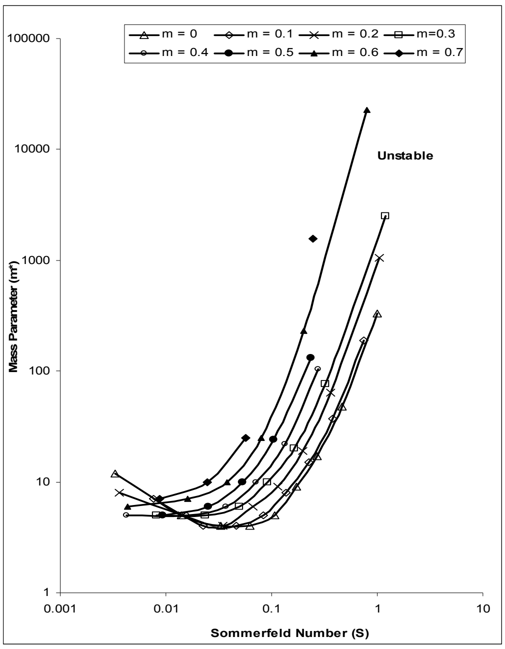

Figure – 3: CSJB
(Dimensionless mass parameter Vs Sommerfeld Number(S))

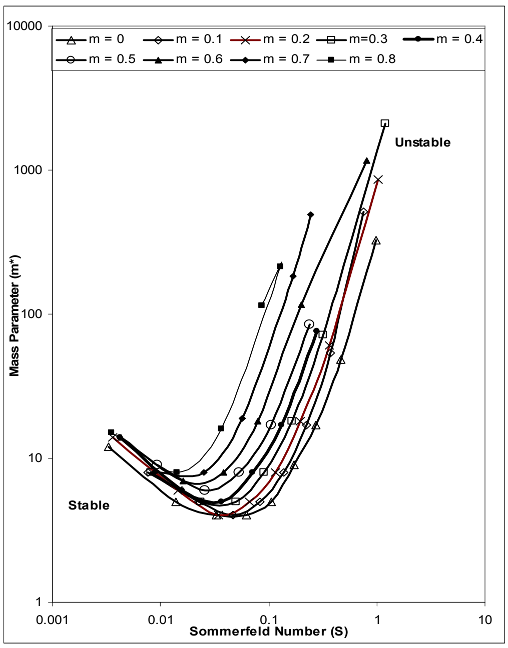

Figure – 4: DSJB
(Dimensionless mass parameter Vs Sommerfeld Number(S))

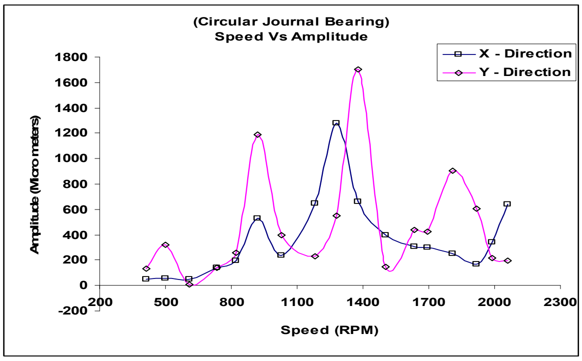

Figure – 5 Oil whip trend in Circular Journal bearing

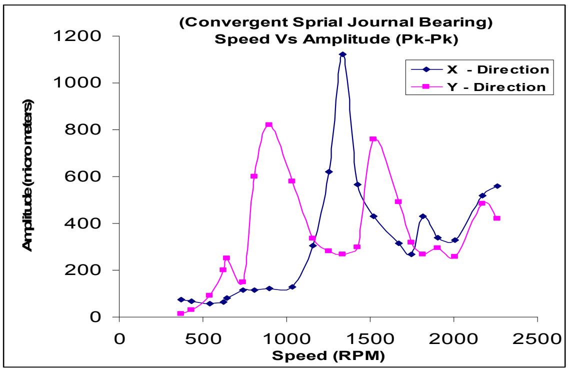

Figure – 6 Oil whip trend in CSJB

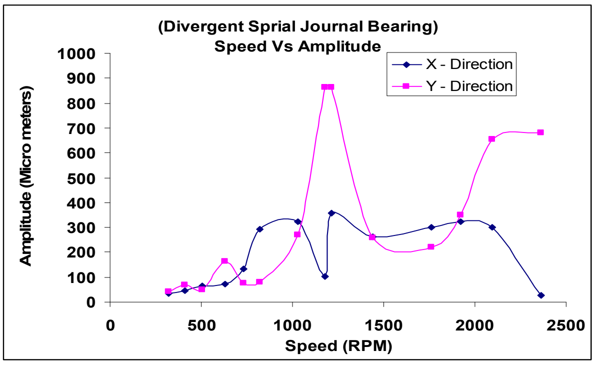

Figure – 7 Oil whip trend in DSJB

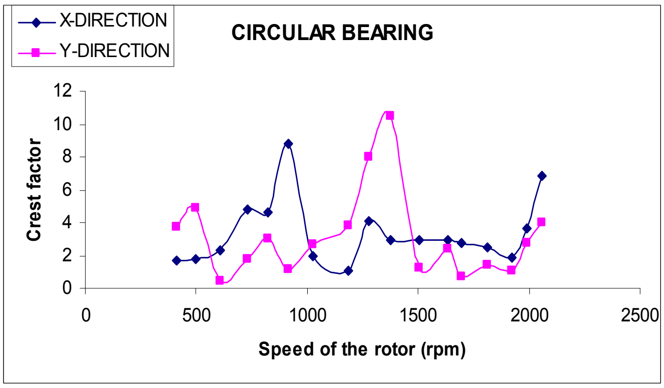

Figure - 8 Crest Factor Vs Speed(Circular J. Bearing)

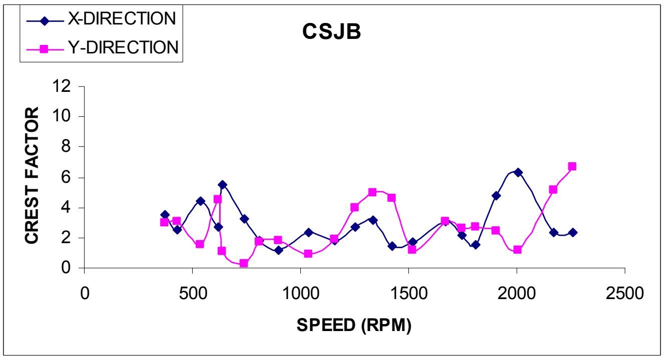

Figure -9 Crest Factor Vs Speed(CSJB)

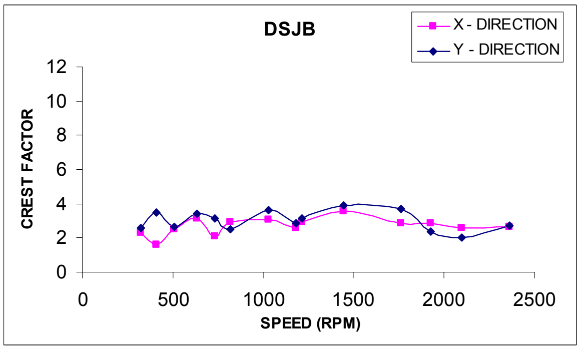

Figure -10 Crest Factor Vs Speed(DSJB)

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