Experimental Study of a Rotor Supported by Deep Groove Ball Bearing

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An experimental analysis has been done to analyze the effect of ball bearing support for a rigid, horizontal rotor. The effect of radial internal clearance of a ball bearing on its stiffness is studied by setting up a rigid, horizontal Jeffcott rotor supported by same type of ball bearings with known radial internal clearance. The nonlinear dynamic characteristics of the ball bearing are also studied by an experimental setup. The signal is acquired by displacement and acceleration transducers. Analysis of the acquired data with the help of Bode plots, cascade plots, time response plots, phase plane plots and frequency spectra brings out the non-linear nature of the ball bearing support. The effect of clearance on the varying compliance frequency, rotational frequency and the resultant sum-difference frequencies can be inferred from the various plots. The anisotropic nature of the ball bearing support shows up as the backward whirl components in full vibration spectra cascade plot. These experimental results validate theoretical results reported in literature.

Keywords: Ball bearing; Anisotropic stiffness; Varying compliance; Sub-harmonics; Super-harmonics; Phase plane

The ball bearing is a strong source of nonlinearity for a rotor system specially when the rotor is lightly loaded. The radial internal clearance of a ball bearing imparts strong nonlinearity. The geometric and elastic properties of a ball bearing also change because of radial internal clearance and constant radial force, which introduces periodically varying

stiffness offered by the balls as these balls roll around the inner and outer races. The varying stiffness gives the parametric effect which is also called the varying compliance effect. Another source of nonlinearity is the Hertzian ball-race contact deformation. Theoretical analysis of the ball bearing supporting a rigid, balanced horizontal rotor has been done by Sunnersjo (1978), Fukata (1985), Mevel and Guyader (1993), Sankaravelu (1994), and Tiwari (1999). Sunnersjo (1978) in his study of bearing defects, studied experimentally the varying compliance effect of a roller element bearing. Tamura and Taniguchi (1960) experimentally studied the motion of the center of a horizontal rotor as the inner race of the supporting ball bearing is turned in steps. The work reported by Tamura and Taniguchi (1960) was effectively a quasi-static experimental analysis and the motivation for this experimental work was provided by the theoretical work done by Perret (1950) and Meldau (1951), and the results of the two studies match fairly well.

In this work an experimental setup has been devised to analyze the effect of varying compliance and unbalance excitation. The control parameters for experimental study are radial internal clearance, rotational speed and level of unbalance. The theoretical results reported by Tiwari (1999) validate the experimental results reported here.

DESCRIPTION OF THE TEST RIG

The Bently Nevada rotor kit configuration is modified to study the effect of ball bearing supports. There are two configurations of the rotor system which have been used in the present study. In configuration-a, Figure 1a, both the rotor supports are same, i.e., ball bearing supports and the disk is placed at midspan. The objective is to have a rigid Jeffcott rotor on supports with anisotropic stiffness. In the second configuration-b (Figure 1b), the support at the motor end is a bush bearing and the other support is the ball. The disk is placed closer to the ball bearing support so

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FIGURE 1 Experimental rig (a) Configuration-a (b) Configuration-b.

as to have most of the weight of the system on the ball bearing support. The first configuration is used for estimating the bearing stiffness in the horizontal and vertical directions. The second configuration is used for studying the dynamic response of the rotor supported by a ball bearing. The bearing supports have been made sufficiently rigid so that the effect of ball bearing stiffness can be studied.

The rotor is coupled to the motor by a flexible coupling. This motor is a variable speed type with a maximum speed of about 10,000 RPM. A feedback controller provides accurate control of speed and ramp rate. The disk has accurately machined threaded holes at 22.5 degrees at a

pitch radius of 30.3 mm for adding unbalance mass. The mass of the disk is 819 gm. The rotor bearing system is mounted on a heavy base plate so as to provide a rigid base for the whole system.

Bearings

The bearings used are SKF 6002 deep groove single row ball bearings provided by SKF R&D, France. These bearings have undergone precise measurements at the manufacturers to provide value of relevant parameters such as radial internal clearance, inner and outer groove radius, the deviation of the circularity of the groove radius,

TABLE I Bearing dimensions

Brgs type/no.	Radial int. cl (μm)($2\gamma_0$)	Inner race groove radius (mm)/Ra(μm)	Outer race groove radius	Ball dia. (mm)	No. of balls
C5/4	39	2.440/0.028	2.585/0.030	4.7620	9
C5/5	40	2.437/0.027	2.588/0.028	4.7620	9
C5/6	39	2.437/0.025	2.584/0.025	4.7620	9
C2/1	7	2.442/0.026	2.590/0.039	4.7620	9
C2/2	6	2.437/0.024	2.585/0.026	4.7620	9
C2/3	5	2.439/0.027	2.575/0.027	4.7620	9

FIGURE 2 Response plots for 3X and 4X frequency components.

ball diameter and the surface roughness of the grooves of the inner and outer races. The bearing dimensions provided by the manufacturers (SKF R&D France) are as given in Table I.

INSTRUMENTATION

In the present experiments, different instrumentation has been used in various experiments, which is described below.

Transducers

The proximity transducers (Bently, Nevada) are used to measure shaft displacement in horizontal (y) and vertical (x) directions. The proximity transducers are eddy current type with a sensitivity of $8 \text{ mV}/\mu\text{m}$ ($200 \text{ mV}/\text{mil}$). The signal from the proximity probes goes through the proximator and the Dual Channel Digital Vector filter (DVF2, Bently Nevada). The DVF is used primarily for order analysis. The dual channel DVF has filters which can band pass or band stop $1/2X$, $1X$, or $2X$ frequency. Since the key phasor signal is an input in the DVF, it displays the amplitude and phase for $1X$ signal along with rotor speed.

For vibration measurement on the bearing pedestal and the rotor base frame, accelerometer (B&K 4371) is used. The accelerometer signal is fed into a charge amplifier (B&K 2635). The output from the charge amplifier can be acceleration, velocity or displacement. A dual channel spectrum analyzer(DSA) HP 3582A is used for spectral analysis of the signals. Both the instantaneous and the averaged spectra are obtained depending upon the requirement. The spectrum analyzer HP 3582A permits spectral analysis in a wide frequency range. The DVF 2 and the spectrum analyzer are also used as part of the data acquisition and analysis system ADRE (Bently Nevada).

Data Acquisition and Processing

Data acquisition and processing of signals is done in two ways. While ADRE3 (Automated Diagnostics for Rotating

TABLE II Critical frequency for SKF6002

Bearing supports	Lower peak (Hz)	Higher peak (Hz)
C2	280 ~ 300	420 ~ 510
C5	180 ~ 210	330 ~ 420
Difference between C2 and C5 (Hz)	~ 100	~ 100

FIGURE 3 Rap test frequency spectra.

Equipment, Bently, Nevada) is used for Bode plots, cascade plots and orbital analysis, a PC based Data Acquisition system with a GPIB card is used for time response plots and frequency spectra.

The data acquisition for ADRE 3 is done by the 108 DAI (Data Acquisition Instrument of Bently, Nevada). Automatic data collection is done by programming change in speed (Δ RPM) at which the sampling is done as the system coasts up or coasts down. After the 108 DAI has acquired and stored data from all the channels, each channel is downloaded on to a PC with the help of RS232 interface and the ADRE for Windows software (Bently, Nevada).

PRELIMINARY TESTS

Prior to detailed response analysis of the bearing/rotor, some preliminary tests were carried out to assess the bearing clearance, defect condition, its stiffness and load.

Estimation of Bearing Clearance

The radial internal clearance is estimated when the bearings are assembled on to bearing pedestals. This is done by assembling similar type of bearings on the pedestals and mounting the shaft without disk in the center of the shaft. The proximity probes are positioned so as to measure the

vertical probe gap voltage. The shaft is very lightly pushed up by hand and the change in the gap voltage is measured on DVF. It is seen that for C2 type bearings, the change in gap voltage is not indicated on DVF which records a minimum of 0.1 V whereas for the C5 type bearings the gap voltage changes by 0.3 volts. With the probe sensitivity of 8 mV/ μ m it is clear that for the C2 type bearing the radial clearance is less than 12.5 μ m whereas for the C5 type bearing the radial clearance is about 37 microns. Measurement of the gap voltage on oscilloscope for C2 bearing indicated a radial clearance of about 5–6 μ m.

Assessment of Bearing Condition

The varying compliance frequency coincides with the frequency associated with outer race point defect which has been shown by Sunnersjo (1985) and Tandon and Nakra (1993). To make sure that the bearings are free of point defects, these were tested with a Shock Pulse Meter (SPM). Also it may be noted that since the loading which the bearing is subjected to in the present case (≈ 6 N) is very low as compared to the loading (≈ 600 N) on the same type of bearing studied by Tandon and Nakra (1993) for outer race defects, the chances of development of points defects are very less. The test result from SPM revealed absence of outer race point defect.

Estimation of Bearing Stiffness

An experimental set up is used for determining the bearing stiffness, Figure 1a. The setup has been made so that we have a horizontal Jeffcott rotor supported on anisotropic bearings. The shaft is supported by deep groove ball bearings of known radial internal clearance. The frequency

TABLE III Natural frequencies of the rotor

	Higher freq. (Hz)	Lower freq. (Hz)
Theoretical	369.6	214.4
Experimental	300	200

FIGURE 4 Experimental setup for bearing load estimation.

spectra of the displacement signal has 2X, 3X, and 4X components which peak at $1/2$, $1/3$, and $1/4$ times the critical speed respectively. This characteristic is seen from theoretical results reported by Tiwari (1999) and it is because of the system having parametric excitation, clearance non-linearity and the resulting anisotropy of the bearing support. It is observed from the DSA that the amplitude of 2X and higher harmonics peak at two frequencies. These two frequencies give us an estimate of the two natural frequencies corresponding to the stiffness in the vertical and horizontal directions. The shaft is rigid with bending stiffness of $1.6 \times 10^8 \text{ N/m}$, which is about 10 times more than the linearized stiffness of the bearings found out theoretically from Tamura's (1985) method. The maximum rotor speed is 10000 RPM so the 3X and 4X components are tracked to give an estimate of the vertical and horizontal natural frequency. RMS averaging is done for 64 samples in the DSA for a number of steady state speeds.

From Figures 2a–b for C2 type bearing (bearing of smaller clearance of about $5 \mu\text{m}$), the amplitudes of 3X and 4X peaks are at two frequencies, i.e., 300 and 520 Hz for vertical direction and 280 and 420 Hz for the horizontal direction. For C5 type bearing (higher clearance of about $40 \mu\text{m}$), Figures 2c–d, the peaks appear at 310 Hz for vertical displacement and around 400 Hz for horizontal displacement. Increased clearance results in many frequency components, as a result of the interaction of these frequency components the peaks of both 3X and 4X do not form very well. In Figure 2c the higher peak coincides at 310 Hz for both 3X and 4X. The lower peak in Figure 2c is not visible here except slightly in 3X at about 200 Hz. The horizontal displacement response, Figure 2d, for C5 does not have very good coincidence of peaks. The lower peak appears to develop slightly around $180 \sim 240 \text{ Hz}$. The higher peak, which is the dominant of the two but not sharp enough appears around 330 to 420 Hz. The change in critical frequencies has been summarized in Table II. The ratio between the vertical and horizontal critical frequency is about 0.6, which agrees reasonably with the value suggested by Krämer (1993). The shifting of critical speed with clearance has also been shown experimentally by

Flowers and Wu (1996) that for a rotor support with clearance (not ball bearing), increase in clearance shifts the critical speed down.

Static Tests

Rap test was carried out on the experimental rig (configuration-b, Figure 1b) in static (nonrotating) condition in order to determine the prominent frequencies. The point of impact and the measurement (placement of accelerometer (B&K 4374)) point were changed so as to be able to give an idea of important structural frequencies. The resulting spectra are given in Figures 3a–d, Figures 3a–c are for C2 bearing support and Figure 3d for C5 bearing support. From Figures 3a–b when the accelerometer is mounted on the disk, the two natural frequencies of the rotor disk system at 200 Hz and 300 Hz are clearly observed. In Figure 3a the point of impact is on the shaft near the disk which is on the center of gravity of the rotor disc system. From Figure 3a we see the dominant frequency is about 200 Hz along with a higher frequency of about 300 Hz. There are some frequencies above 500 Hz. From Figure 3b, when the point of impact is one end of the rotor, the dominant frequency is 300 Hz while the 200 Hz frequency is also present. Higher frequencies ($> 500 \text{ Hz}$) stay more or less at the same level as in the previous case. From Figure 3c, it is seen that higher frequencies of the base plate and bearing pedestal are excited strongly where the point of impact is on the base plate. For C5 type bearing Figure 3d the frequency spectra is found to be the same as for C2 type, Figure 3a.

The rotor system is also analyzed using a transfer matrix procedure for its natural frequencies. The theoretical and

TABLE IV Load on the bearing

L (mm)	Readings of the load cell (kg)	
	Without motor	With motor
120	0.66	0.566
140	0.735	0.674

FIGURE 5 Bode plot, high unbalance, direct signal, vertical displacement (a) C2 type (b) C5 type.

FIGURE 6 Bode plot C2, low unbalance, direct signal, vertical displacement (a) C2 type (b) C5 type.

experimental values of the shaft disc system are compared in Table III. The theoretical value of natural frequencies are higher compared to the experimental value. Theoretical

modeling of the bearing is by linearized stiffness which does not take into account the clearance whereas clearance lowers the stiffness in experimental tests.

TABLE V Comparison of displacement response of C2 and C5 type bearing

Unbalance level (kg-m) (C2/C5)	Peak amplitude (μm)		Speed (RPM)	
	Vertical displacement response		Vertical displacement response	
	C5	C2	C5	C2
$8 \times 10^{-7} / 9 \times 10^{-7}$ (lower)	250	80	8753	8960
$2 \times 10^{-6} /$ 2.4×10^{-6} (higher)	210	150	8560	9772

FIGURE 7 Cascade plot (full vibration spectrum), high unbalance, 1X notched (a) C2 type (b) C5 type.

Estimation of Load Carried by Bearing

The shaft is supported at three points: the ball bearing, the bush bearing, and the flexible coupling between the motor and shaft. This system is statically indeterminate. To find

the load carried by the bearing, a simple experiment was devised, as shown in Figure 4.

The ball bearing pedestal is mounted on a load cell. The two remaining supports (bush bearing pedestal and motor with coupling) are mounted on a base plate. The base plate

FIGURE 8 Cascade plot (full vibration spectrum), low unbalance, 1X notched (a) C2 type (b) C5 type.

is jacked up with help of screw jacks and with a spirit level the load cell mounting surface and the base plate are leveled. The load cell directly gives the weight which is coming on the bearing pedestal. With the help of a height gauge, the deflection in the load cell surface was estimated as the load is applied on the ball bearing pedestal. The deflection of the load cell was found to be negligible. It can be assumed that the error due to tilting of the shaft-disk system as a result of load cell deflection is negligible. All three supports were on the same level as deflection in the load cell is very small.

The span $S = 20.0$ mm remains constant. It is the nearest distance the disk can be placed to the ball bearing pedestal. The total mass of 0.943 kg gets divided between the three supports. The load cell readings for two values of span with and without the motor attached to the rotor are given in Table IV. For a span $L = 140$ mm the weight taken up by

the ball bearing is $F = 6.6$ N. This is taken as the constant radial force acting on the ball bearing.

ANALYSIS OF RESULTS AND DISCUSSION

The rotor is supported by bush bearing and the ball bearing, configuration-b, Figure 1b. The disk is placed as close as possible to the ball bearing support so that the major part of the weight of the shaft-disk system is on the ball bearing.

The bush bearing with negligible clearance approximates an isotropic bearing support very well. This is shown by Gupta et al. (1993) and Gupta (1989) who also estimated the stiffness of this bush bearing as 2.2×10^5 N/m.

FIGURE 9 Response plot for SKF 6002 type C5.

Following parameters are varied in the present experimental study.

- (a) Type of bearing with different radial clearance
- (b) Unbalance level
- (c) Speed of rotor

Two different levels of radial internal clearances are taken up for study; SKF 6002 type C2 with radial clearance of about $5\mu\text{m}$ and SKF 6002 type C5 with radial clearance of $40\mu\text{m}$ (Table I). The two unbalance levels are $8 \times 10^{-7}\text{kg-m}$ as the lower level and $2 \times 10^{-6}\text{kg-m}$ as the higher level for C2 type ($9 \times 10^{-7}\text{kg-m}$ and $2.4 \times 10^{-6}\text{kg-m}$ for C5 type). Rotor speed is varied up to 10500 RPM.

Direct Signal

On comparing the Bode plot for type C2 with high unbalance level in Figure 5a and low unbalance level in Figure 6a we see that for high unbalance the maximum response

is achieved at about 9772 RPM where as for low unbalance, it is at 8960 RPM. For C5 type bearing the peak amplitude is attained at a speed lower than that for C2 type bearing (Table V). The lowering of critical speed is because of increased radial clearance. Theoretically and experimentally, Flowers and Wu (1993 and 1996) have shown that increased radial clearance at the rotor support leads to decrease in critical speed. This has also been the conclusion by Tiwari (1999) from theoretical analysis.

The peak in the Bode plot for C5 type, Figure 6b, develops at 8753 RPM for low level of unbalance. In Figure 6b between 4500 and 5000 RPM, a weak second peak can be seen. The second peak is developed due to 2X component at half the speed of 8753 RPM. This component is not present for lower radial internal clearance of C2 type bearing because the strength of 2X with respect to 1X for C2 type bearing is weaker compared to that in C5 type bearing. For C5 with high level of unbalance, Figure 5b the peak develops at 8560 RPM.

1X Signal Band Rejected

The signal from the horizontal and vertical displacement probes are input in the Digital Vector Filter. The DVF is set for band rejecting (bandwidth 2 Hz) 1X in both the channels. The output signal has 1X component to a negligible level. The absence of 1X makes it easy to study higher super-harmonics and sub-harmonics in the frequency spectra.

Cascade Plots

Cascade plots were generated for 1X band rejected signal. A comparative analysis of the 1X band rejected cascade plots of bearing type C2 with bearing type C5 brings to light some important observations discussed below.

For high level of unbalance we see that C2 type bearing has a dominant peak near 9000 RPM (Figure 7a) whereas the second peak at about 5400 RPM is not well developed. Between 7400 to 8000 RPM, there is lot of speed fluctuation and resulting key phasor error. Above 7000 RPM, 1X starts developing (although signal is 1X band reject filter) pointing to the dominance of very strong 1X due to high unbalance and lower radial internal clearance. On comparing these dynamic characteristics with the ones observed for type C5 from Figure 7b, we see some logical differences.

For the C5 type bearing with higher unbalance level, Figure 7b, 2X clearly develops two peaks, compared to this for lower radial internal clearance, Figure 7a only one peak develops strongly with in the experimental range of measurement. This points to decrease in stiffness with increased radial internal clearance. For C2 type bearing we also see a weakly developed 2X reverse vibration component as compared to the well developed 2X reverse vibration component for the type C5 bearing, in Figure 7b, which is because of increased anisotropy due to increased radial internal clearance. Increased anisotropy at the bearing support increases the level of backward whirl component as shown by Muszyńska (1996).

In both the bearings, varying compliance frequency component 3.6X is well developed but the amplitude for C5 is more. Super-harmonic components such as 3X, 4X develop much better for larger clearance as in C5 as compared to C2. Also the linear combinations of 1X and varying compliance 3.6X, which are 2.6X, 1.6X, 4.6X, etc., develop better in larger clearance C5 bearing. The direct relationship between clearance and strength of varying compliance is a good match with the findings of theoretical simulation in the present analysis. For C5 (Figure 7b) between 3000 to 5000 RPM we see appearance of a number of higher ($> 3.6X$) frequency components. They are noticeably absent in the Figure 7a for C2 bearing. It can

be observed without much effort that for a larger clearance the frequency spectrum is very "rich".

Reduced level of unbalance Figures 8a–b results in smaller amplitudes as can be seen by comparing cascade plots in Figures 7a–b with Figures 8a–b. The 2X component has very well developed two peaks. Also the

FIGURE 10 Time response plots (vertical displacement).

cascade plots become "neat", and there is no fluctuation of speed and the resulting key phasor error. On comparing Figure 8a (C2 type) with Figure 8b (C5 type), it can be inferred that higher radial internal clearance seems to result in stronger frequency components. This holds very well for frequency components more than order 2X. Also for type C5 bearing, frequency components above 3.6X (4X, 4.6X, 5X, 5.6X, ...) appear clearly in the speed range 3000 to 4600 RPM (Figure 8b). The 3.6X component has a strong forward vibration component for higher clearance, Figure 8b. Higher clearance results in more super-harmonic and sub-harmonic frequency components which is also supported by the theoretical study.

The appearance of super-harmonics such as 2X, 3X, and 4X is characteristic of clearance nonlinearity in the bearing support and the anisotropy of the bearing support. This has been shown theoretically by Fukata et al. (1985). Flowers and Wu (1993, 1996) have shown this theoretically and experimentally for a rotor system supported on springs which has clearance (not a ball bearing). The super-harmonics peak at the frequency corresponding to critical frequency which is clearly observed in the cascade plots Figures 7 and 8. The attainment of maximum amplitude of these super-harmonics at the critical frequency is again an observation which has been reported by Fukata et al. (1985) based on the theoretical analysis of a single ball

bearing supporting a balanced rotor and by Flowers and Wu (1993, 1996) for a rotor support having clearance.

Analysis of Time Response Plot

To study the steady state dynamic behaviour of the rotor system the vertical and horizontal response signals were fed in the two channels of the DVF, the output of which was in turn fed to a two channel Tektronix digitizing oscilloscope (TDS 320). For certain steady state speeds the time response plot of the signal was taken. The DVF was used to filter out frequencies and study the time response signals. The total speed span over which the rotor system was run was 1000 to 10400 RPM.

Two cases of (a) low level of unbalance (9×10^{-7} kg-m) and (b) high level of unbalance (2.4×10^{-6} kg-m) for configuration-b (Figure 1a) with C5 type bearing were studied. The peak to peak amplitude of the vertical displacement response measured from the plots obtained from the Oscilloscope have been presented in Figures 9a-b as speed response plots. For increasing values of steady state speeds at which the readings were taken the peak to peak value increase, from 4266 RPM (Figure 10), the periodic nature of the response changes to amplitude modulated response. From 5000 RPM there is a lot of

FIGURE 11 Orbit plots.

fluctuation in the peak to peak displacement response. At 6900 RPM a jump was observed and recorded (Figure 10). The peak to peak response which has been recorded reaches its maximum at about 8000 RPM but the amplitude modulation continued till a speed of 8890 RPM when the response seems to again become near periodic.

For high unbalance level, Figure 9a, the time response undergoes the same kind of changes from periodic to amplitude modulated. The peak is reached at about 8000 RPM but no jump was detected in this case. The dominance of 1X for high unbalance level produces lesser fluctuation in the peak to peak values.

The amplitude modulation in the time response plots (Figure 10) is also seen from the orbit plots shown in Figure 11. In these plots, the long time period because of the multi-frequency response results in a "net" structure of the orbits. The orbits gradually develop an elliptical

structure for increase in speed because of the growing strength of the 1X component as the speed increases.

Phase Plane Plots

Phase plane plots have been experimentally generated and the structure of the phase plane plots is qualitatively analyzed. The instrumentation for this experiment consists of an accelerometer (B&K 4371) mounted on the bearing pedestal. The signal of this accelerometer is integrated in the charge amplifier (B&K 2635) to give a velocity output. This velocity output is fed into both the channels of the DVF. One channel of the DVF is put in integration so the output is displacement. The latter signal (displacement) and also direct signal from the second channel (velocity) are fed in a digitizing oscilloscope (TDS 320) which is set to $x-y$ mode.

FIGURE 12 Phase plane plots, C5 type.

FIGURE 12 (Continued).

FIGURE 13 Phase plane plots, C2 type.

Phase plane plots for SKF6002 C5 and C2 type bearing support for high level of unbalance and low level of unbalance are generated. The DC value of the velocity signal is negligible and it is difficult to stabilize the signal on the screen. Therefore it was difficult to take the plots at desired speeds. However, following four sets of phase plane plots are generated

- (1) C5 type bearing, low unbalance (9×10^{-7} kg-m)
- (2) C5 type bearing, high unbalance (2.4×10^{-6} kg-m)

- (3) C2 type bearing, low unbalance (8×10^{-7} kg-m)
- (4) C2 type bearing, high unbalance (2×10^{-6} kg-m)

On comparing the phase plane plots for C5 type bearing with low unbalance level (Figures 12a–h) with high unbalance level (Figures 12i–p), it can be seen that the plots for low unbalance level have a structure (lines crossing each other in a net structure) which points to the interaction of more than one frequency resulting in amplitude modulated time response waveform. The plots for high level of

unbalance show a tendency to be elliptical which is because of the dominant 1X frequency producing a strong single harmonic output for which the phase plane plot is an ellipse or circle. There seems to be a tendency for more than one stable solutions as shown by the 'cleft' structure (plots in Figures 12e–g) of the phase plane plots dividing the phase plane space into more than one area of attraction. Near the critical speed, about 7000 RPM to 8000 RPM, for the high unbalance case we see very good elliptical phase plane plots which points strongly to the prominent 1X signals resulting in very strong harmonic response. This feature is not so clear for the low unbalance case where we do not see elliptical structure of the phase plane plots.

Comparing the plots for C2 type bearing for low unbalance case (Figures 13a–d) with high unbalance case (Figures 13e–g), we see a similar structure. Both cases have elliptical phase plane plots which indicate weak sub-harmonics or no signal other than 1X. This compares well with theoretical and other experimental findings which show a decrease in sub-harmonics and super-harmonics with decreasing clearance (i.e., decreased nonlinearity). On comparing these with phase plane plots obtained for C5 type (higher clearance), it can be inferred that the higher clearance has more frequency components in its response.

Analysis of Frequency Spectrum for SKF6002

For two levels of unbalance, frequency spectra were obtained at different operating speeds for two different setups of experimental rig configuration-b (Figure 1b). In one setup the support had C2 type bearing in the other

setup C5 type bearing. The horizontal and vertical displacement signals were fed in the DVF which was set to 1X band rejection (2 Hz band width). These two signals were then fed in the dual channel spectrum analyzer. The spectrum plots were acquired after averaging 64 samples in the RMS mode, and the passband shape in the dual channel spectrum analyser was set to Hanning.

The rotor was run up to the required speed and reading was taken only after DVF indicated no fluctuation in speed. The plots were then down loaded on to the PC with the help of GPIB interface and Labview software.

The frequency spectra for low level of unbalance has frequency components for the varying compliance and the rotational frequency. For high clearance levels the varying compliance frequency is stronger compared to the low level of clearance value. For C2 type of bearing (low clearance) the super-harmonic (2X, 3X) dominate, there is very little appearance of sub-harmonics which also includes 3.6X the varying compliance frequency. For C5 type of bearing (high clearance) varying compliance (3.6X) is clearly visible which interacts with the rotational frequency to give sum and difference frequencies which results in the amplitude modulation of the time response. Low level of unbalance and C5 type bearing support generates spectra which is "rich" in sub-harmonics, as the speed nears 6000 RPM, there is also appearance of 1/2X. The response plots in Figure 9b also show strong appearance of sub-harmonics in the range of 5000 RPM to 7000 RPM which cause a lot of fluctuation in the response plot. After the speed crosses 7000 RPM the spectra again has dominant rotational frequency and 2X, 3X. For a speed of 5020 RPM the frequency spectra are shown for C2 and C5 type bearings (Figures 14a–d). The C2 type bearing has less sub-harmonic components (Figures 14a–b) where as C5

FIGURE 14 Frequency spectra 5020 RPM, low unbalance (a–b) C2 type (c–d) C5 type.

(Figures 14c–d) has VC, 2X, 3X and the linear combination of these frequencies. The C5 type bearing also generates a 1/2X component, Figures 14c–d.

CONCLUSIONS

The experimental analysis shows that bearing clearance changes the response of a rotor significantly because of the change in dynamic stiffness of the bearing. This has been shown by theoretical analysis also. Most estimates of bearing stiffness do not take into account the change in dynamic stiffness due to clearance, for example Tamura (1985); Gargiulo (1980). Experimental results of the present study have clearly indicated that higher clearance generates more sub-harmonic and harmonic components in response compared to the case when the bearing has lesser clearance. The synchronous component 1X appears strongly for the case of lower bearing clearance in comparison to the case of larger bearing clearance. The nonlinear characteristic are significant for lower unbalance and high bearing clearance. The jump phenomena could be experimentally seen for lower unbalance and higher clearance. The orbit plots, time response plots and phase plane plots all show the amplitude modulated nature of the response.

The orbit plots have a “net” structure which point towards a long time period due to multifrequency response. The phase plane plots also substantiate the increased strength of non-linearity for higher clearance and lower unbalance. Experimental results have confirmed some observations from published literature like reduction in critical speed with increased clearance, unequal/different critical speeds in horizontal and vertical directions and existence of cross-coupling between horizontal and vertical stiffness. From the present experiments on a rigid, horizontal Jeffcott rotor, chaotic response could not be observed with certainty.

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