Fluid Induced Vibrations in Rotors Supported by Journal Bearings: A Case Study

Manish Agrawal and R. V. S. Krishnadutt

Abstract Rotors are supported by oil in fluid film bearings. Usually the oil film has a damping effect on the rotor. However, for some rotors, the oil film may become source of self-induced vibrations and cause instability in the system. Unlike the forced vibration response whose frequency is equal to the excitation force, self-exciting vibrations have a frequency equal to one of the rotor modes. As the source of vibrations in not linked to external excitation, vibration diagnostics of this kind of problems becomes complex. A case of a rotor failure due to self-exciting vibrations, its failure investigation and subsequent remedial measures are presented in this paper.

1 Introduction

Vibrations originating from fluid induced forces in the clearances between rotor and stator are associated with sub-synchronous frequency [1] and the phenomenon is called fluid whirl or fluid whip. The phenomenon is responsible for self-excited vibrations resulting in rotor instability. Fluid whirl frequency is at about 0.43–0.49 times the rotational speed as per Oliver [7] whereas fluid whip frequency gets locked to the first bending mode of the rotor. However, fluid whirl and fluid whip belong to the same phenomenon and are generated by the same source, i.e. fluid induced forces. Fluid whirl shows up if the instability threshold occurs at low speeds and transits into fluid whip at speed about twice the frequency of first bending mode. Fluid whip continues as the speed goes up and the frequency of sub synchronous vibrations gets locked to the frequency of first bending mode. Oil whip was first reported by Newkirk

M. Agrawal (✉) · R. V. S. Krishnadutt
Machine Dynamics and Failure Analysis, Bharat Heavy Electricals Limited
Corporate Research and Development, 500093 Hyderabad, India
e-mail: manishagrawal@bhel.in

R. V. S. Krishnadutt
e-mail: krishnadutt.rvs@gmail.com

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and Taylor [6]. A detailed description of self-excited vibrations and evaluation of stability zones has been explained by Muszynska [3–5].

In a hydrodynamic fluid film bearing, the lubrication oil rotates along with the rotor at some circumferential velocity. This circumferential velocity is equal to the rotor velocity at the rotor surface and gradually decreases towards the bearing surface. The faster is the rotor speed, the higher is the fluid average velocity. Average circumferential velocity of the fluid in the annular space between rotor and bearing is approx. half of the rotor speed. Muszynska has shown that the destabilizing fluid force, pushing on to the rotor is proportional to the fluid average circumferential velocity. The magnitude of the sub synchronous vibrations due to this fluid force are self-exciting and grow till the magnitude of vibrations becomes equal to the oil clearance and the shaft touches the bearing surface.

Oil-whirl is usually a weak vibration phenomenon and not observed as the response is dominated by unbalance. Therefore, the oil whip frequency is commonly considered as the threshold of instability. However, it is essential to accurately identify the instability threshold at design stage to avoid any unexpected vibration behavior during operation, for high speed flexible rotors. This paper presents a case where oil instability induced vibrations caused large vibrations resulting in a rotor failure and the remedial measures taken up to mitigate the instability.

2 Problem Statement: Failed Rotor and Large Vibrations

The rotor was under balancing at the balancing tunnel in the factory. The rotor synchronous vibrations were within acceptable limits and it was decided to take up the over-speed test. However, the rotor experienced catastrophic failure at over-speed. Figure 1 shows the broken rotor assembly after the failure. Figure 2 shows the run up plot before the failure and run down plot recorded until complete failure. It can be seen from these plots that the synchronous vibrations do not seem to be the cause of failure and have gone up subsequent to the failure.

The failure investigation pointed out at the shrink fit release resulting in large unbalance and amplified response due to vicinity of over-speed to second critical

Fig. 1 Rotor assembly after failure at balancing tunnel for the failed rotor

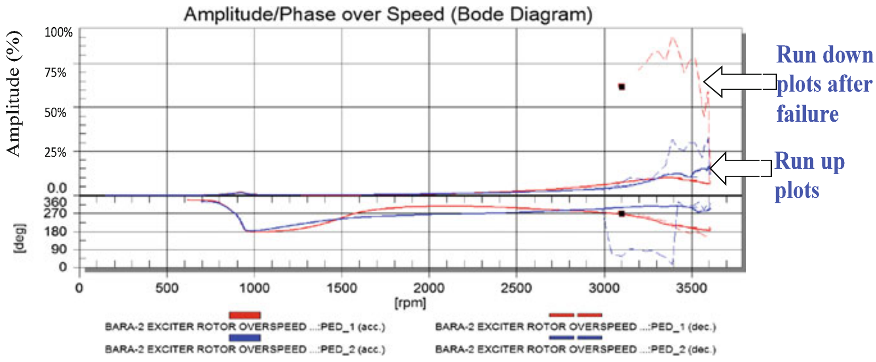

Fig. 2 Run up and run down plots showing amplitude of synchronous component for the failed rotor

speed of the rotor bearing system as possible causes of failure. The shrink fit was improved and critical speed was pushed up by reducing the bearing span. This time, the output of the vibration sensors was connected to a vibration analysis instrument and the output was recorded for diagnostic purposes.

The rotor could successfully clear the over-speed test with acceptable synchronous vibrations, however, extremely large overall vibrations were observed at higher speeds. Run up and run down plots for this rotor are shown in Fig. 3. The synchronous component is well controlled. The gap in OA and 1X vibrations is attributed to sub synchronous component, as shown by waterfall plots, Fig. 4. The sub-synchronous component shows up at around 2400 rpm and steadily grows with increasing speed. The frequency of the sub synchronous component is observed to be fixed and not varying with speed.

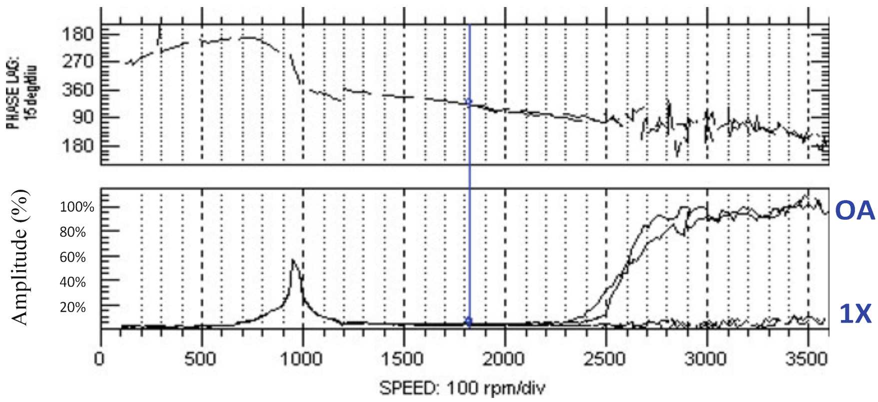

Fig. 3 Measured run up and run down plot for a successful rotor showing unusually large gap between OA and 1X, 2500 rpm onwards

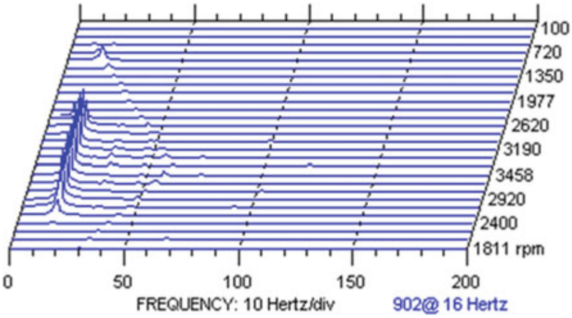

Fig. 4 Waterfall plot (non drive end) showing large sub synchronous component

3 Rotordynamic Analysis: Fluid Induced Instability

In view of the presence of a large sub synchronous component in the measurements, the rotordynamic behaviour of the rotor bearing system was reviewed. The primary aim of the analysis was to detect the instability threshold and identification of the response frequency, expected during the unstable behavior. MADYN2000 has been used for numerical simulations.

Figure 5 shows the first flexural mode of the rotor bearing system. The rotor is supported on elliptical bearings. The first bending mode at 890 rpm (14.8 Hz) is shown to have negative damping, indicating an unstable mode. Negative damping is confirmed by the Campbell and modal damping diagram, shown in Fig. 6.

Journal bearings are generally preferable for reducing the vibration response, due to the damping effect, however, insufficient design poses a risk of self-exciting vibrations above one of the rotor critical speed, usually the first critical speed. Run up plot for this case (Fig. 3) shows the harmonic resonance at around 950 rpm. When the speed increases beyond first resonance, self-exciting vibrations, known as oil whirl occurs. In the oil whirl condition, the rotor whirls like a rigid rotor in forward direction with small amplitude. The frequency of oil whirl is usually from 0.39 to 0.48 of rotational speed. The actual value is dependent upon type of bearing and the shaft location in the bearing. With further increase in the speed, the rotor

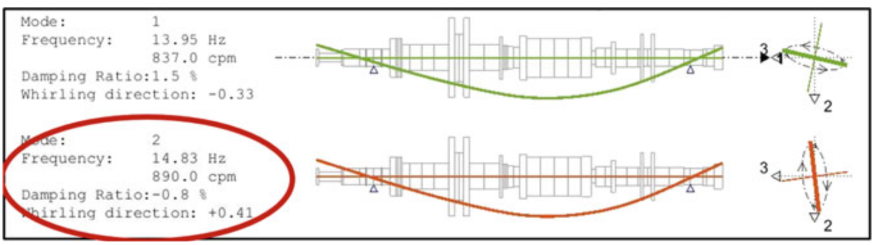

Fig. 5 Critical speeds of the rotor bearing system from eigenvalue analysis

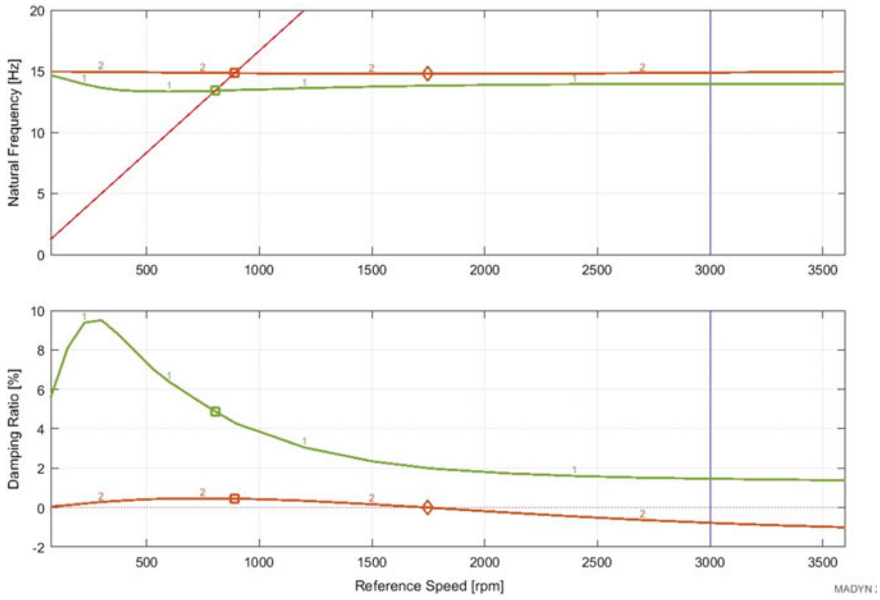

Fig. 6 Campbell and modal damping diagram showing negative damping for first bending mode

crosses the threshold of instability, which is usually at about twice the first critical speed. Transition of self-exciting from oil whirl to oil whip takes place. Frequency of self-exciting vibrations due to oil whip is locked to the frequency of first mode. For bearings with medium to heavy load, the oil whirl may not show up and oil whip may occur directly. This statement is shown to be true for the rotor under investigation. The bearings have specific load of approx. 1.0 N/mm^2 thus falling in the category of bearings with medium load. Measurements have shown the occurrence of oil whip directly without the appearance of oil whirl. Measurements show the threshold of instability at around 2200 rpm and the sub synchronous component rises violently with the increase in speed beyond this.

Nature and threshold of instability have been captured through non-linear transient rotordynamic analysis. Figure 7 shows amplitude versus time plots at 2550 rpm. The vibrations are self-exciting and very large in amplitude, reaching equal to the available oil clearance in the bearing.

4 Instability Due to Cross Coupling Stiffness Coefficients

Destabilizing force arising from the cross coupling stiffness terms must be suppressed by the dissipative force arising from the damping. However, acceleration of rotor results in rapid increase in cross coupling stiffness terms and the destabilizing force may surpass the dissipative force leading to instability of rotor [2].

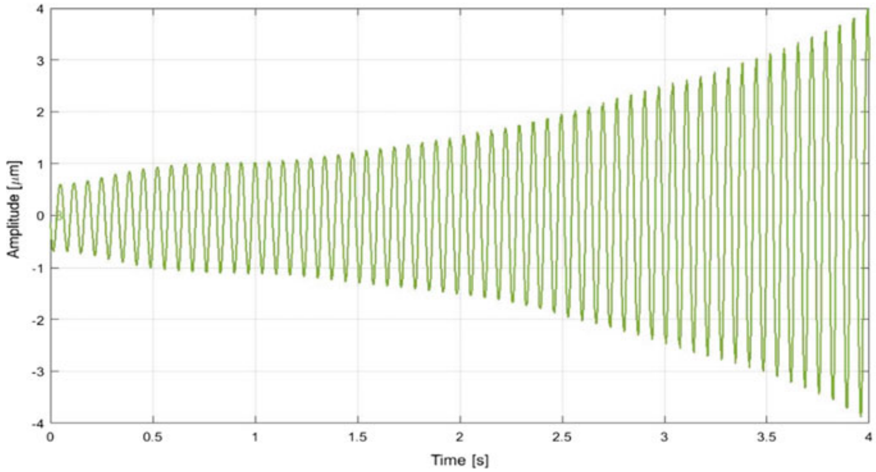

Fig. 7 Amplitude versus time plots at 2550 rpm, obtained from non-linear transient analysis

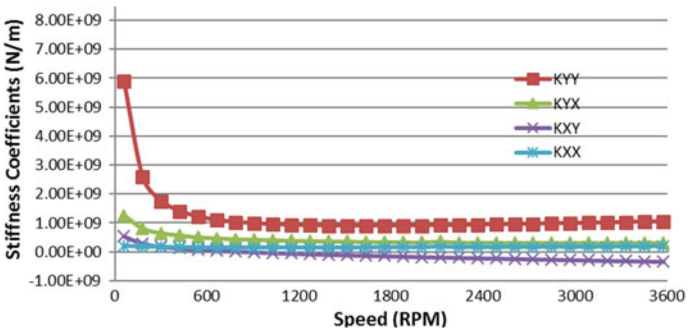

Fig. 8 Stiffness coefficients for elliptical bearing

Figure 8 shows the stiffness coefficients versus rotor speed graph for the elliptical bearing. Amplitude of the cross coupling stiffness term KXY increases rapidly with speed. This leads to destabilizing force dominating the dissipative force, causing the instability.

5 Remedial Measure: Use of Tilting Pad Bearings

The possible remedial measures to avoid instability include increasing the natural frequencies of the system by reducing the bearing span or increasing the shaft diameter, increasing the instability threshold, increase the bearing load, increase the eccentricity ratio (eccentricity ratio = eccentricity/oil clearance) and using a tilting pad bearing. Tilting pad bearings do not have significant cross coupling terms and the destabilizing tangential force from oil film on the rotor.

Figures 9 and 10 show the eigenvalue analysis results and the Campbell and modal damping diagram for the rotor with tilting pad bearings. Damping ratio for the first bending mode, which is negative with original elliptical bearings, has now turned positive. However, the damping ratio at 0.4% indicates insufficient damping and hence, the sub-synchronous component is expected to be present though with a small amplitude.

Figure 11 (Estimated amplitude versus time plots at operating speed) shows that the self-exciting sub synchronous vibrations do not diminish, thus confirming the insufficient damping. Figure 12 shows the stiffness coefficients vs rotor speed graph. It can be seen that cross coupling stiffness terms KYX and KXY are insignificant as compared to the elliptical bearing.

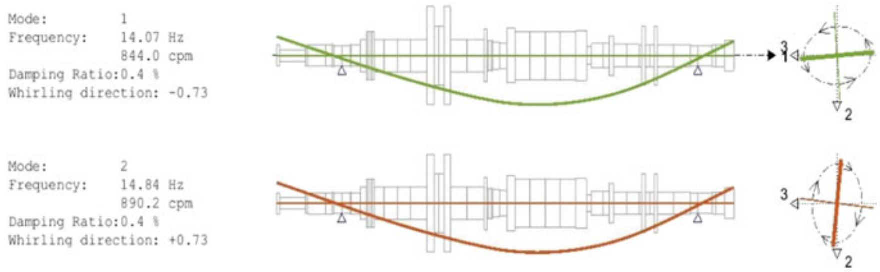

Fig. 9 Critical speeds of the system with tilting pad bearings

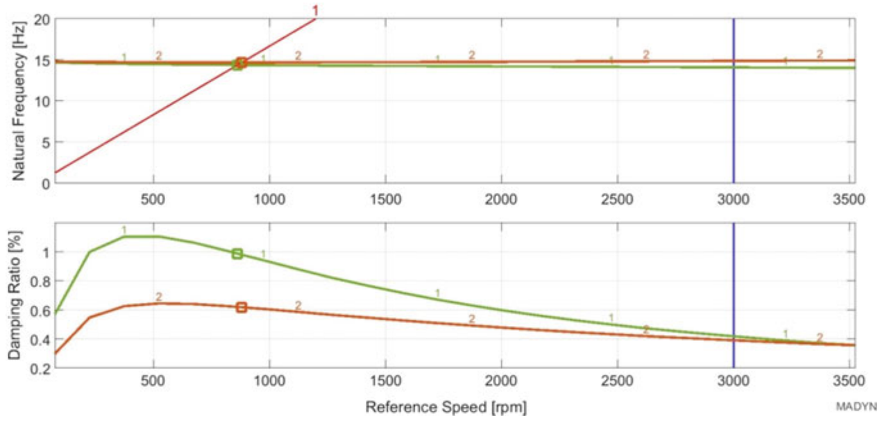

Fig. 10 Campbell diagram for the system with tilting pad bearings

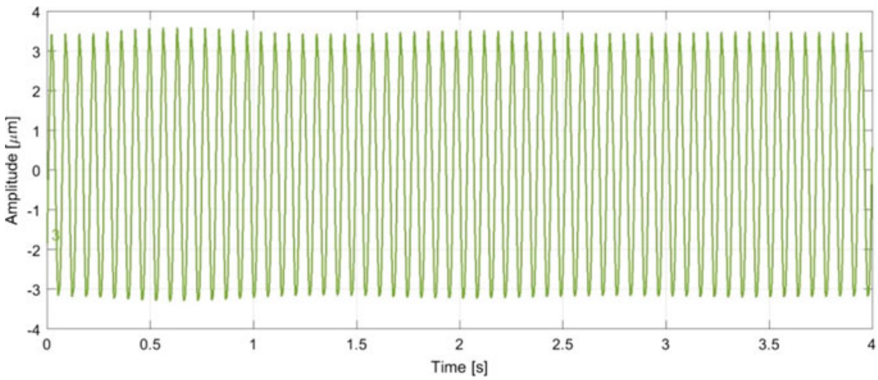

Fig. 11 Amplitude versus time plots at operating speed (3000 rpm)

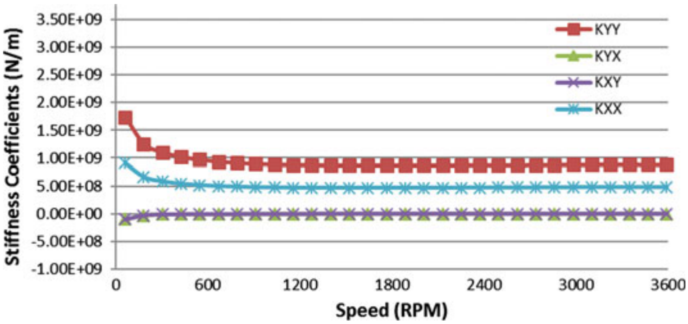

Fig. 12 Stiffness coefficients for tilting pad bearing

The bearings were replaced by tilting pad bearings and the rotor was operated in the balancing tunnel. Figures 13 and 14 show the measured run up and waterfall plots. The waterfall plot shows the presence of sub synchronous component in small amplitude, above 2000 rpm. This validates the results obtained from non-linear analysis.

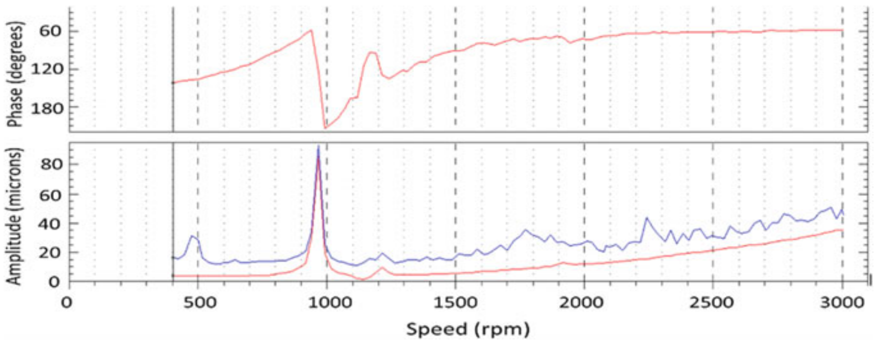

Fig. 13 Measured run up plot for the rotor with tilting pad bearings

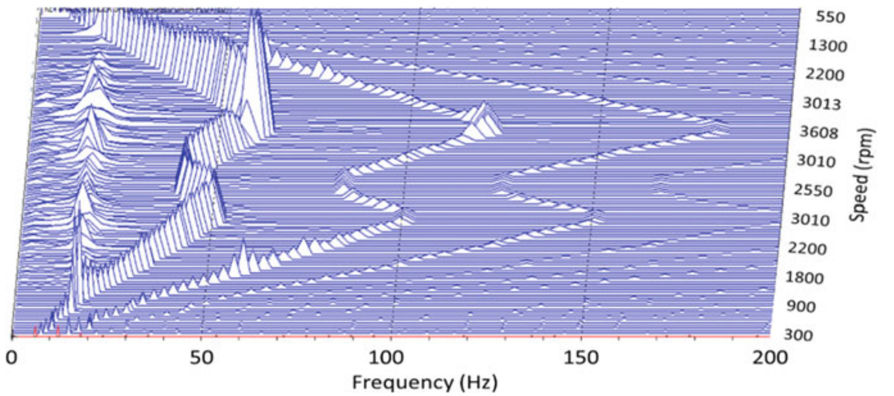

Fig. 14 Measured waterfall plot for the rotor with tilting pad bearings

6 Conclusions

Following conclusions are drawn from this case study:

1. Thresholds of instability and frequency of the unstable mode have been predicted through the rotordynamic analysis and are in good agreement with the measurements. The work successfully reiterates the need of rotordynamic analysis considering the stability of the rotor bearing system and prediction of any instability, present in the system at design stage to avoid the unpleasant surprises during operation.
2. The elliptical bearings with preload are known to be more stable than the cylindrical bearings. However, in this case, use of elliptical bearings has not helped in preventing the occurrence of instability.
3. Tilting pad bearings have been introduced replacing the two lobe elliptical bearings to mitigate the instability. However, sub synchronous component originating from oil whip is present even in case of the rotor supported on tilting pad bearings, though in small amplitude. Thus, it can be inferred that the self-exciting vibration mechanism cannot be eliminated by changing the bearings alone. This requires modification of the rotor design.
4. In the balancing tunnels world over, balancing teams consider the synchronous vibrations alone and usually do not monitor the overall vibration levels. However, it may be a good practice to monitor the overall vibrations and in case of significant non synchronous component, the diagnostic team can be alerted.

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