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PERFORMANCE ANALYSIS OF HYDRODYNAMIC JOURNAL BEARING WITH THE EFFECT OF WHIRL INSTABILITY

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ABSTRACT

Hydrodynamic journal bearings require greater care in design and operation than hydrostatic bearings. They are also more prone to initial wear because lubrication does not occur until there is rotation of the shaft. Oil whirl is when a lubrication wedge can't form, but instead "whirls" around the bearing, causes tremendous self excitation. This leads to direct contact between the journal and the bearing, which quickly wears out the bearing. This paper represents instability analysis of journal bearings using stiffness estimation at different journal speeds & loads. In this theoretical analysis is presented on journal bearing at 450 N and 1440 r.p.m., which shows that the bearing will remains stable upto a speed of 2378 r.p.m.

Keywords: *Hydrodynamic Journal Bearing, Whirl Instability, Eccentricity Ratio, Stiffness and Film Thickness.*

1. Introduction

Hydrodynamic bearings are common components of rotating machinery. They are frequently used in applications involving high loads and/or high speeds between two surfaces that have relative motion. Journal bearings are specific to surfaces that mate cylindrically with the applied load in the radial direction. In the study of journal bearings many aspects of engineering are present. Stress analysis, fluid dynamics, instrumentation, vibration, material properties, thermodynamics, and heat transfer are some of the common subjects encountered in understanding hydrodynamic bearings. The research on the stability of a journal bearings-rotor system is one of the most important subjects of rotor bearing dynamics. However, the mechanism of instability caused by oil whirl is not completely clear. This lack of good understanding of the nature of oil whirl and whip hinders the effective fault diagnosis of the journal bearing-rotor system. By analyzing the interrelation among whirl speed, eccentricity ratio, rotating speed, oil supply pressure, bearing load, oil film thickness, and observing the pressure distribution in the three phases of oil film attenuation, instability transition, and oil film load instability, it was found that in the whirl instability transition phase, the amplitude of rotor whirl speed exceeds half the value of rotor rotation speed, which causes the pressure change in the oil convergent wedge to change from positive to negative and the pressure change in the oil divergent wedge to change from negative to positive, and the change in oil film pressure distribution causes the instability of oil whirl. These changes in oil film pressure distribution are the indicator of oil whirl instability [9].

2. Background

The first reported discovery of self-excited vibration referred to as oil whirl was published in 1925 by Newkirk and Lewis [1]. In studying vibration in a cylindrical journal bearing, the experimenters noticed that oil flow to the bearing had an effect on the vibration of the rotor bearing system. Under the given conditions, the rotor would cease vibration when they stopped the supply of oil to the bearing. Kakoty [2] presented that, due to vibration, failure of machinery in applications such as aero engines, turbo machines, space vehicles, etc. creates enormous repair costs and more importantly may put human life in danger. Jang et al. [3] presented an analytical method to investigate its stability by transforming the equations of motion to the eigenvalue problem with Hill's infinite determinant. The validity of this research is proved by the comparison of the stability chart with the time response of the whirl radius obtained from the equations of motion. Jun Sun et al. [4] analyzed hydrodynamic lubrication characteristics of a journal bearing, taking into consideration the misalignment caused by shaft Deformation. Ogrodnik et al.[5] stated that Oil whirl instability is a serious problem in oil lubricated journal bearings. The phenomenon is characterised by a sub-synchronous

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vibration of the journal within the bush and is particularly apparent in turbo generators, aero engines and electric motors. Elmadany et al. [6] described the development of an optimal control law design for the lateral vibration suppression and stabilization of a rotor system with anisotropic fluid-film bearings and fluid leakage. Majumdar et al. [7] analyzed the stability characteristics of the system. A linearized perturbation theory about the equilibrium point can predict the threshold of stability; however it does not indicate post whirl orbit detail. Abdou et al. [8] examined the dynamic behavior of fluid film bearing of finite width as a result of sinusoidal variation of the applied load and/or the sinusoidal variation of the journal rotational velocity. A change in load or rotational velocity of the journal is always accompanied by waves exciting the bearing fluid layer. These waves produce fluid forces on the journal. A perturbation technique is applied to obtain the governing equations for dynamic conditions. Chen Ce et al. [9] presented a theoretical analysis of the mechanism of oil whirl, oil whip, and hysteresis in oil whip. A qualitative analysis of the interrelation between whirl frequency, natural frequency, and eccentricity ratio versus speed was conducted using the rotor-bearing load balance equation. According to Subbiah [10], It is well known in rotor dynamic community that oil whirl/whip occurrences in rotors supported on cylindrical type fluid-film bearings are, in general, speed-dependent and not usually turbine megawatt (MW) load dependent. It is also known that oil whip conditions in rotating machinery can be addressed by replacing the cylindrical type bearings with tiltpad type. However, it was found that under certain MW loading or operating conditions, oil whirl occurred in rotor systems that were supported by tilt-pad type bearings. Oil whirl to whip also occurred at increased turbine MW loading which meant oil whirl/whip could also be load dependent. Bently Nevada Corporation [11] offers insight into the difference between whirl and whip in their article archives. Whirl and whip are both self-excited vibrations that occur when the fluid forces generated in the lubricant tend to rotate the rotor within the bearing. The key to distinguishing the difference between the two lies in the understanding of the stiffness of the rotor bearing system. The stiffness of the rotor and the stiffness of the bearing, or fluid, act in series. The weaker stiffness controls the overall stiffness of the rotor bearing system. During whirl, the bearing's stiffness is weaker than the shaft's stiffness.

3. Principle of Journal Bearing

When a Journal bearing which has an adequate supply of lubricant is carrying a load it normally runs with the geometric centres of the shaft and housing displaced so that a region of convergent flow is established. In this region large hydrodynamic pressures are set up within the oil film and these pressures when summated over the total bearing surface are found to completely support the load. If bearing conditions change; for instance the load may vary, the displacement or "attitude" of the centres changes so that the new pressure distribution is sufficient to support the new load. Fig. 1 illustrates the basic principles [13].

Various non-dimensional load parameters are used to assess the performance of a bearing. The parameters are formed from terms such as the oil viscosity and the load and speed of the bearing. For a given bearing it is found that there is a critical value of load parameter at which the convergent film is unable to support the total bearing load and the surfaces touch. Under these conditions the friction torque suddenly starts to rise and boundary lubrication occurs in the region of minimum film thickness. Fig. 2 illustrates these phenomena. Based on his theoretical investigation of cylindrical journal bearings, Professor Osborn Reynolds showed that oil, because of its adhesion to the journal and its resistance to flow (viscosity), is dragged by the rotation of the journal so as to form a wedge-shaped film between the journal and journal bearing (Fig. 3) [13].

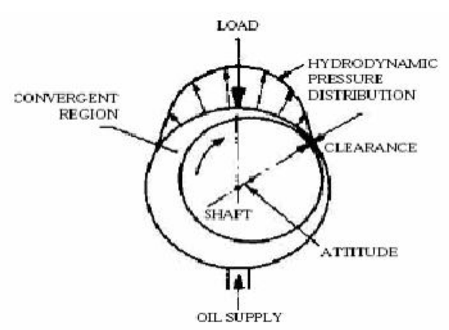

Fig. 1 Convergent Region (Pressure Distribution)

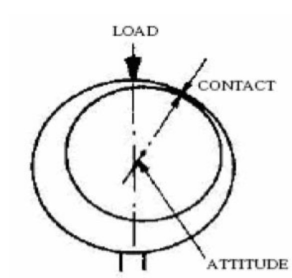

Fig. 2 Metal to Metal Contact

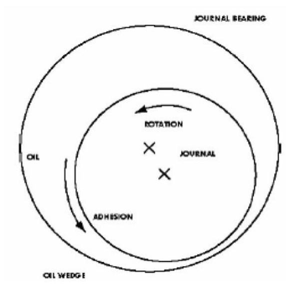

Fig. 3 Hydrodynamic Principle

4. Analysis of Hydrodynamic Journal Bearing

In these bearings the load-carrying surfaces are separated by a stable thick film of lubricant that prevents the metal-to-metal contact. The film pressure generated by the moving surfaces that force the lubricant through a wedge shaped zone. At sufficiently high speed the pressure developed around the journal sustains the load. This is illustrated in Fig. 4.

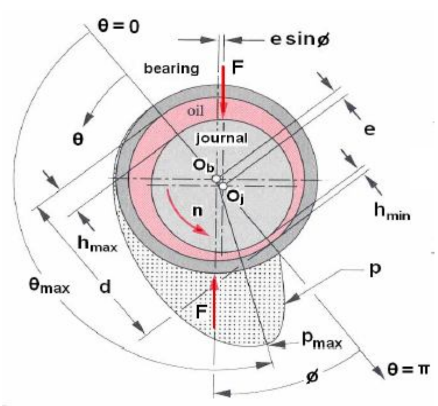

Fig. 4 Bearing Geometry

4.1. Operating parameters

Table 1: Operating Parameters

Type of lubricant	ISO VG 46 μ = 98.1 CP
Diameter of journal	40 mm
Length of bearing (L)	40 mm
L/D ratio	1
Diametrical clearance	0.185 mm
Speed (N)	1440 r.p.m.
Load (W)	450 N

4.2. Calculation of parameters

Table 2: Calculated Parameters

Bearing pressure	0.5625 MPa
Summerfield	0.04891
Eccentricity	0.6467

4.3. Analysis of whirl speed

Consider partial arc bearing ($\beta = 90^0$)

Let, m = Diametrical clearance/Radius of journal

$= 9.25*10^{-3}$

Table 3: Values of η and ϵ for L/D as 1 [12]

ϵ	A	η
0.2	1.6	0.42
0.3	2.4	0.4375
0.4	3.2	0.455
0.5	4	0.4825
0.6	5.18	0.51
0.6467	6.10	0.5321

Using the mathematical relations as [12],

Pavg = = 0.0861 A η ……………………(1)

Where, μ = 1.45*10-7*Z.

W = Pavg * L * Dj = 137.88 A η ……… (2)

ho = m r (1-ϵ) = 0.185 (1-ϵ) ……………. (3)

Selecting the values of A and η from Table 3,we come to know that applied load 450 N will produce an eccentricity of 0.6467 with a minimum thickness of 0.065 mm.

4.4. Analysis of whirl speed

The hydrodynamic journal bearing is analyzed for different values of eccentricity as shown in Fig. 5, by using equations (1), (2) and (3), so as to determine an average pressure, load and a minimum oil film thickness.

To estimate stiffness of bearing, the stiffness of shaft is combined with the stiffness of bearing that supports journal. It determines the whirl instability or critical speed. Stability of journal bearing is obtained from the graph of Load V/s eccentricity ratio so as to estimate a minimum oil film thickness. A tangent is drawn to the obtained curve at operating eccentricity ratio as shown in fig.5 and slope is determined. The estimated magnitude slope of tangent indicates the stiffness at that point [12]. Slope can be estimated as,

Stiffness = $\Delta W / \Delta ho$ …………………….. (4)

From graph and equation (3),

The minimum film thickness ho at ϵ = 0.8 is 0.037 mm and at ϵ = 0.4 is 0.111 mm. Hence the stiffness k becomes,

k = 2844621.204 N/m

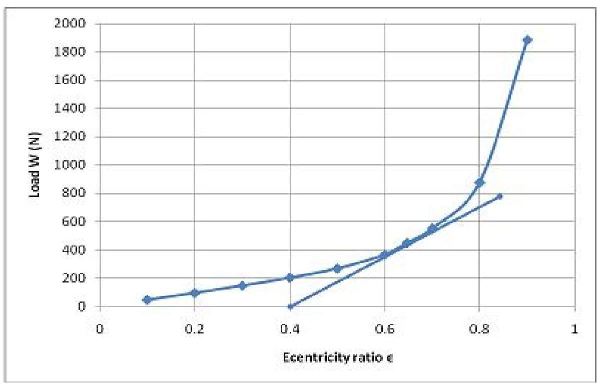

Fig. 5 Bearing Performance

The frequency of the first critical speed (called synchronous whirl or whirl speed) would be,

Assuming rigid body conditions,

$$f = \frac{1}{2\Pi}\sqrt{(k/m)} \quad \text{……………………...(5)}$$

= 39.63 cycles / sec

= 2378 cycles / min.

5. Conclusion

Based on the concept of stiffness coefficients, stability speed of journal bearing is found to be 2378 r.p.m. operating at 1440 r.p.m. and 450 N load, which indicates that bearing, is stable up to 2378 r.p.m. and it proves that whirl speed is more than half of rotor speed. It also means that, upto this whirl speed, pressure distribution is regular.

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