NEGATIVE PRESSURE IN THE OIL-FILM OF JOURNAL BEARING

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ABSTRACT

In the present paper, negative pressure developed in the oil-film of journal bearing is discussed. Assuming dilatational surface viscosity, negative pressure found experimentally could be reproduced. Parameter study has shown that, when negative pressure is developed: the load capacity increases and the friction coefficient of bearing decreases slightly near the eccentricity 0.6, but this effect reverses at larger eccentricity. Oil flow rate decreases under negative pressure. The locus of center of journal in the bearing clearance is pushed outwards and horizontally under vertical static load. This means that the stability of a turbo-rotor can be reduced, when negative pressure is developed.

KEYWORDS: negative pressure, dilatational surface viscosity, journal bearing, oil-film, bubble theory

1. INTRODUCTION

Characteristics of journal bearing, such as load capacity, frictional power loss, oil flow rate etc, are determined by pressure distribution in the oil-film of the bearing. Therefore, in order to get the characteristics correctly, the pressure distribution has to be calculated under realistic boundary condition. In the literature, the oil film pressure has been usually calculated under the condition that the oil-film can not sustain negative pressure. However, several experimental results with negative pressure are also reported for the oil film of journal bearing [1, 2, 6-9]. In the present paper, distribution of pressure including negative one should be calculated and compared with experimental results for a journal bearing with a circumferential oil groove as shown in Fig.1. It should be also discussed how the development of negative pressure influences the characteristics of journal bearing.

Fig. 1 Journal bearing with a circumferential oil groove.
2. EXPERIMENTAL EVIDENCE OF NEGATIVE PRESSURE

Figure 2 gives examples of measured circumferential distribution of pressure at land center in a bearing shown in figure 1, at shaft speed of 1500 min\(^{-1}\) and four bearing load \(W\). The results show clearly that there exists negative pressure and its absolute value increases with bearing load. Figure 3 gives another example of negative pressure, where a pair of parallel plates with oil-film in-between is pulled off. Both types of pressure pickup yielded negative pressure.

![Fig.2 Measured pressure distribution](image1)

![Fig.3 Negative pressure at reverse squeeze](image2)

3. THEORETICAL INVESTIGATION ON NEGATIVE PRESSURE

Using a “bubble theory” explained below, pressure distribution in journal bearing should be calculated and compared with experimental results. It is assumed that the oil contains uniformly dispersed bubbles with a constant radius \(R_a\) under atmospheric pressure \(p_a\). When this oil is introduced into bearing gap and experiences positive or negative pressure as it flows in the gap space in the bearing, the bubbles will contract or expand. It is further assumed that the bubbles remain spherical in the oil film and no interference, no combination and breakup between bubbles takes place. Also no evaporation and condensation of oil and no diffusion of gas should occur. The gas of constant mass contained in the bubble will experience isothermal change during the expansion and compression of bubble [10].

Figure 4 depicts the forces acting on the surface of a bubble in oil-film. When the bubble radius changes, surface force due to dilatational viscosity [4, 11, 12] of the oil will oppose the radius change. This surface dilatational viscosity is related to Marangoni effect [5]. That is, when the surface of a liquid enlarges, adsorption taking place in the solution is decelerated by the resisting force, the surface dilatational viscosity. Therefore, when the bubble expands under negative pressure, surface dilatational viscosity force \(\Delta \sigma\) will resist the bubble expansion and the bubble can withstand greater negative pressure without rupture than when only surface tension \(\sigma\) is acting and no surface dilatational viscosity acts.

According to Scriven [11, 12], equation (1) holds for \(\Delta \sigma\) with surface area \(A\) and surface dilatational viscosity \(\kappa\). Taking this force into account, equation (2) is derived for the ratio \(\chi=R/R_a\) of a bubble radius \(R\) [8, 11, 12].

Considering the change of viscosity \(\eta\) and density \(\rho\) due to inclusion of bubbles, a modified Reynolds equation (3) for the gauge pressure \(p\) is derived. As for void fraction \(\alpha\) and density ratio \(\delta = \rho / \rho_1\) of the oil-bubble mixture, equation (4) and (6) can be derived [6]. For viscosity ratio \(K = \eta/\eta_i\), an experimental formula (5) according to [6] was used, where \(\eta_i\) stands for viscosity of oil alone. For the dimensionless oil-film thickness \(H = h/\Delta r\), equation (7) holds with radial clearance \(\Delta r\). In equation (2) dimensionless numbers for surface tension \(\sigma\), oil viscosity \(\eta_i\) and surface dilatational viscosity \(\kappa\) are introduced by equations (8)–(10):
\[ \Delta \sigma = \kappa \frac{1}{A} \frac{dA}{dt} \]  

\[ (1 + \frac{p}{p_0}) \chi^3 + (B + C \frac{d\chi}{d\phi}) \chi^2 + \hat{D} \frac{d\chi}{d\phi} \chi - (1 + \hat{B}) = 0 \]  

\[ \{ \frac{\partial}{\partial \phi} (\frac{\delta H^3}{K} \frac{\partial p}{\partial \phi}) + \frac{\partial}{\partial \zeta} (\frac{\delta H^3}{K} \frac{\partial p}{\partial \zeta}) \} \psi^2 = 6(H \frac{\partial \delta}{\partial \zeta} + \delta \frac{\partial H}{\partial \phi}) \]  

\[ \alpha = \frac{\alpha_0 \chi^3}{1 - \alpha_0 + \alpha_0 \chi^3} \]  

\[ K = 1 + 0.5062 \alpha + 9.044 \alpha^2 - 46.83 \alpha^3 + 60.13 \alpha^4 - 23.85 \alpha^5 \]  

\[ \delta = 1 - \alpha \]  

\[ H = 1 - \varepsilon \cos(\varphi - \gamma) \]  

\[ \hat{B} = \frac{2 \sigma}{p_a R_a} \]  

\[ \hat{C} = \frac{4 \eta_1 \omega}{p_a} \]  

\[ \hat{D} = \frac{4 \kappa \omega}{p_a R_a} \]  

4. INFLUENCE OF NEGATIVE PRESSURE ON BEARING CHARACTERISTICS

Having obtained the positive results from bubble theory above, some parameter study should now be made to investigate how the bearing characteristics will be influenced by the development of negative pressure [13~16].

Figure 6 depicts the specific bearing load \( \bar{p} \) by bubble theory and conventional theory as function of bearing eccentricity \( \varepsilon \). The curve for the ratio \( Z \) of both \( \bar{p} \) is also drawn. It can be seen from this figure that the load capacity increases slightly near \( \varepsilon = 0.6 \), when negative pressure is developed in the oil-film. At larger \( \varepsilon \) the effect reverses.
Fig. 5 Pressure distribution in journal bearing (1500 min\(^{-1}\)).
Figure 7 shows the reduced bearing coefficient of friction $\mu' = \mu/\psi$ by both theories and their ratio $Z$. The curves show that the bearing friction decreases slightly near $\varepsilon=0.6$, when negative pressure is developed. At larger $\varepsilon$ the effect reverses.

Figure 8 depicts the journal centre orbits calculated by both theories under vertical load. The orbit is pushed horizontally and towards bearing clearance circle when negative pressure is developed. This means that the oil-film force becomes more “unstable”, that is the equilibrium of a rotor carried by journal bearings can become more unstable due to the oil-film force, when negative pressure is developed.

Figure 9 depicts the oil flow rate $Q$ out of both sides of the bearing. The oil flow rate decreases by $\Delta Q$, when negative pressure is developed, because the oil flowed out of the bearing sides is sucked again by the negative pressure.

CONCLUSIONS

In this paper, existence and influences of negative pressure are investigated for the oil-film of journal bearing with a circumferential oil groove. The results can be summarized as follows.

1) Existence of negative pressure could be verified experimentally in the oil-film.

2) Taking into consideration the surface dilatational viscosity, the negative pressure could be reproduced.

3) When negative pressure is developed in oil-film, the bearing load capacity increases slightly and the frictional coefficient decreases near $\varepsilon=0.6$. At larger $\varepsilon$ these changes reverse.
(4) When negative pressure is developed, the static journal orbit is pushed horizontally and towards the bearing clearance circle. Rotors running in journal bearings can become then more unstable.

(5) Oil flow is reduced by the amount of oil sucked again by negative pressure, when it is developed.

REFERENCES