Modelling of Outer and Inner Film Oil Pressure for Floating Ring Bearing Clearance in Turbochargers

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Abstract. Floating ring bearing is widely used in turbochargers to undertake the extreme condition of high rotating speed and high operating temperature. It is also the most concerned by the designers and users alike due to its high failure rate and high maintenance cost. Any little clearance change may result in oil leakage, which in turn cause blue smoke or black smoke according to leakage types. However, there is no condition monitoring of this bearing because it is almost impossible to measure the clearance especially the inner clearance, in which the inner oil film directly bears the high speed rotation. Instead of measuring clearance directly, this paper has proposed a method that uses film pressure as a measure to monitor the bearing clearance and its variation. A non-linear mathematical model is developed by using Reynolds equations with non-linear oil film pressure. A full description of the outer and inner film is provided along both axial and radial directions. A numerical simulation is immediately carried out. Variable clearance changes are investigated using the mathematical model. Results show the relationship between clearance and film pressure.

1. Introduction

Floating ring bearing is widely used in automotive turbocharger to support its rotating part [1]. Compared with plain journal bearing, floating ring bearing comprises two fluid films in series, which could improve lubricant quality, decrease relative rotational speed between journal and bearing. It is therefore more suitable for turbocharger’s extreme working condition of high speed and high temperature [2]. However high failure rate of bearings is also concerned by designers and users because of such extreme conditions as abrasion caused by lubricant impurities and starvation, high temperature of exhaust gases, high rotational speed, etc [3]. Bearing damage would have given rise to rotor imbalance, shaft oscillation, impellers broken and might cause serious accidents. Monitoring of bearing working status is therefore, quite meaningful for fault detection and prognosis of turbochargers [4].

A number of researchers have paid attention to turbocharger floating ring bearings. Clark [5] presented a steady-state model of floating ring bearing to simulate temperature distribution inside bearing. Thermal effects on the performance of floating ring bearing was also considered by Andres [6], who developed a model for prediction of floating ring bearing forced response. It is pointed out that the clearances between the solid parts of bearing are affected by the thermal effects and thus influence the rotational speed of the floating ring and the oil film pressure distribution. Chun [7] works
on aeration effects on turbocharger journal bearing. Aerated and pure lubricant have been compared in terms of effects on oil pressure and temperature distribution inside bearings. Relevant experiments have been done by Andres [8] on a small turbocharger supported on floating ring bearings. Shaft motions of turbocharger have also been investigated.

Pressure distribution of plain journal bearing have been studied by some researchers, e.g. infinite long bearing by Wang [9] and infinite short bearing by Castro [10]. However so far it is still lack of an effective approach to monitor pressure distribution, especially for finite length floating ring bearing with two fluid films. The present paper proposes a novel method to solve this problem that uses oil film pressure distribution to monitor bearing status instead of measuring its clearances directly. In this paper, structure of floating ring bearing of turbocharger is firstly studied as well as the lubricant system. A nonlinear mathematical model is developed to analyze fluid pressure distribution in floating ring bearing. Finite difference method has been adopted to solve Reynolds equations of outer and inner film simultaneously. Following model development, pressure distribution have been simulated under various simulation parameters, for example different lubricant supply pressure, different lubricant viscosity, bearing wear, etc. Simulation results have then been analyzed to find out relationship between pressure distribution and such working circumstances.

2. Analysis

2.1. Floating ring bearing of turbocharger

Since plain journal bearing could not bear such high rotational velocity, most of automotive turbochargers incorporate floating ring bearings to support the rotor [11]. The existence of floating ring decreases greatly relative speed of journal. Figure 1 shows the structure of floating ring bearing, which comprises two fluid films separated by the rotating floating ring, i.e. outer fluid film between bearing and floating ring, inner fluid film between journal and floating ring. While turbocharger is working, floating ring will be rotating at a certain speed driven by shear-driven torques of two fluid films.

Approximate 1/7 amount of engine lubricant is supplied into turbocharger housing [2]. Oil is fed into outer fluid film through a hole on the top and inner fluid film is lubricated through six supply holes distributed in the middle of floating ring. Following heat transfer and lubrication, lubricant flow out bearing from two ends and enter into turbocharger casing.

![Diagram](image_url)
When working in steady state, both journal and floating ring will rotate at their static equilibrium positions or inside stable limit cycles. So far prediction of exact rotational speed of floating ring is complicated which depends upon lots of factors, such as lubricant temperature, viscosity and bearing structure, etc. To simplify calculation, a model is adopted in this paper to predict rotational speed of floating ring based on short bearing theory in which speed ratio of floating ring to journal is considered as a constant [6]. The relationship is given as follows.

$$\frac{\Omega_R}{\Omega_J} = \frac{1}{1 + R_o^3 / R_i^3 \cdot C_i / C_o}$$  \hspace{1cm} (1)

Existence of the floating ring make pressure distributions of two fluid films affect each other. If status of one of fluid films changing, instantaneous displacement of floating ring will be changed and pressure distribution of another fluid film will be influenced as well. Reynolds equations are expressed for two fluid films $P_i$ and $P_o$ as follows.

$$\frac{1}{R_j^2} \frac{1}{\partial \theta} \left( \frac{h_i^3}{12\mu} \frac{\partial P_i}{\partial \theta} \right) + \frac{1}{\partial z} \left( \frac{h_i^3}{12\mu} \frac{\partial P_i}{\partial \theta} \right) = \frac{\Omega_j + R_j / R_i \cdot \Omega_R}{2} \frac{\partial h_i}{\partial \theta} + \frac{\partial h_i}{\partial t}$$  \hspace{1cm} (2)

$$\frac{1}{R_o^2} \frac{1}{\partial \theta} \left( \frac{h_o^3}{12\mu} \frac{\partial P_o}{\partial \theta} \right) + \frac{1}{\partial z} \left( \frac{h_o^3}{12\mu} \frac{\partial P_o}{\partial \theta} \right) = \frac{\Omega_R}{2} \frac{\partial h_o}{\partial \theta} + \frac{\partial h_o}{\partial t}$$  \hspace{1cm} (3)

where $R_j, R_i, R_o$ are the radius of journal and inner and outer radius of floating ring respectively, $\mu$ is lubricant oil viscosity, $\Omega_j, \Omega_R$ are rotational speed of journal and floating ring. $\theta$ is circumferential coordinate starting from y axis. As shown in Figure 1, inner and outer film thickness are given in equation (4) and (5).

$$h_i = C_i + (x_j - x_R) \sin \theta - (y_j - y_R) \cos \theta$$  \hspace{1cm} (4)

$$h_o = C_o + x_R \sin \theta - y_R \cos \theta$$  \hspace{1cm} (5)
where $C_i, C_o$ are inner and outer clearances of floating ring bearing, $x_j, y_j$ are journal displacement on x and y direction, $x_R, y_R$ are floating ring displacement on x and y direction.

The present paper adopts Finite Difference Method to solve Reynolds equations. Figure 2 illustrate grid scheme of the whole oil film lands of two fluid films, where i and j stand for nodes index on the circumferential and axial direction respectively.

![Grid schemes of oil film lands](image)

**Figure 2.** Grid schemes of oil film lands

Since lubricant is fed into outer fluid film through a round hole on the top of bearing, fluid pressure at where supply hole locates covered by oblique lines in Figure 2(a) is considered to be equal to the lubricant supply pressure. Inner fluid film is lubricated by six supply holes distributed in the middle of floating ring. Here centrifugal force of inner film because of rotation of floating ring is neglected. Consequently outer film pressure distribution obtained, fluid pressure of such nodes in the middle of floating ring on circumferential direction have the same values covered by oblique lines in Figure 2(b). Apart from those nodes, pressure distributions at other nodes could been solved iteratively.

3. Simulation results

3.1. 3-D pressure distribution of outer and inner films

Both outer and inner films have been divided into a great number of nodes, 200 nodes on axial direction and 360 nodes on circumferential direction. Simulation parameters are described as follows. Turbocharger is considered to be working at rotational speed of 3000 rad/s. Outer and inner clearance of floating ring bearing are 80 $\mu$m and 20 $\mu$m. Radius of inner and outer floating ring are 7 mm and 11 mm respectively. Bearing length is 11 mm. Lubricant viscosity is 15cp.

Figure 3 show 3-D pressure distribution of outer (L) and inner films (R). Lubricant is fed into outer film through a supply hole shown at both ends on circumferential direction. In inner film, circumferential pressure in the middle of bearing equal pressure value of at outer film the same position. Pressure are distributed symmetrically on both sides of the holes of floating ring. In the cavitation area where fluid pressure less than ambient pressure, fluid film would have no longer been continuous and might not be able to support loads.

3.2. Influence of lubricant supply pressure

Figure 4 shows pressure distribution on circumferential direction at a certain profile on the left hand side of lubricant supply hole. Figure 5 shows pressure distribution on axial direction on the bottom of bearing. Different supply pressure has been given from 1 to 8 times of ambient pressure. Normalized pressure is obtained by being divided by ambient pressure.
It can be seen that inner film pressure appear to be much higher than outer film, which is caused by its smaller thickness and higher rotating speed (journal speed plus floating ring speed). According to simulation results, influence of lubricant supply pressure is not obvious in terms of film pressure shapes. However, if supply pressure is so low as similar with ambient pressure, pressure distribution will be much lower than normal. In that case no enough lubricant would be fed into inner clearance and lubricant starvation might take place.

**Figure 3.** 3-D pressure distribution of outer and inner films

**Figure 4.** Pressure distribution on axial direction: (L) outer film (R) inner film
3.3. Influence of lubricant viscosity

Figures 6 and 7 show waterfall diagram of pressure distribution on circumferential direction and axial direction respectively. Viscosity mainly affects fluid pressure values. Fluid pressure of both outer and inner films becomes higher as lubricant viscosity increases. Moreover, increase extent of inner film pressure exceeds greatly that of outer film.

3.4. Influence of bearing clearance

Another important factor affecting fluid film pressure distribution is bearing clearance. Due to extreme working condition, bearing wear would have happened on turbochargers. Once this phenomenon lasting for a period, outer clearance will be changed and thus pressure distribution in outer and inner films will be influenced.

In this paper, such two cases of bearing wear have been considered. In first case, the radius of bearing increases and in the other case the lower half of bearing becomes ellipse instead of round. Relationship between bearing clearance and pressure distribution have been simulated in figures 8 to 11.

For case one, as shown in figures 8 and 9, it can be seen that bearing clearance mainly affects outer film pressure distribution on axial direction. The larger outer clearance is, the smaller peak value of outer film pressure will be. Inner film is affected by means of the change of boundary condition, i.e. pressure distribution of outer film in the middle of bearing. However, simulation results shows that influence seems to be too weak compared with outer fluid film.

Figure 10 illustrates schematic view of floating ring bearing in the case of lower half of bearing transforming into ellipse. Outer clearance is expressed as follows.

\[ C_o = \frac{r_B (r_B + \Delta B)}{\sqrt{r_B^2 \cos^2 \theta + (r_B + \Delta B)^2 \sin^2 \theta}} - r_o \]  

where \( r_B \) is radius of bearing and \( r_o \) is outer radius of floating ring respectively. \( \Delta B \) denotes difference between semi-major axis and semi-minor axis of ellipse, representing maximal value of bearing wear.
Outer film thickness is therefore expressed.

\[ h_o = \frac{r_B (r_B + \Delta B)}{\sqrt{r_B^2 \cos^2 \theta + (r_B + \Delta B)^2 \sin^2 \theta}} - r_o + x_B \sin \theta - y_B \cos \theta \]  

(7)

Substitute equation (7) into Reynolds equation, pressure distribution can be solved using Finite Difference Method.

**Figure 6.** Pressure distribution on axial direction: (L) outer film (R) inner film

**Figure 7.** Pressure distribution on circumferential direction: (L) outer film (R) inner film
Figure 8. Pressure distribution on axial direction: (L) outer film (R) inner film

Figure 9. Pressure distribution on circumferential direction: (L) outer film (R) inner film

Figures 11 and 12 show the relationship between $\Delta B$ and pressure distribution. In comparison with the first case, main difference is outer film pressure on axial direction. Fluid film pressure drops sharply as $\Delta B$ increases. When $\Delta B$ reaches $4 \times 10^{-2} \text{mm}$, fluid film pressure appears to be quite low already. Large clearance in the lower half of bearing makes it difficult to form squeeze film under the floating ring and bearing wear might therefore affect loading capacity of bearing.
4. Conclusion

In this paper, a nonlinear mathematical model has been developed to investigate fluid pressure distribution of outer and inner film after study of structure of turbocharger floating ring bearing and lubricant system. Finite difference method is adopted to solve Reynolds equations. Following model development, pressure distribution of outer and inner film have been simulated under various working circumstances.

Simulation results show that lubricant supply pressure does not affect fluid pressure obviously, except for the case of supply pressure same as ambient pressure which would lead to lubricant starvation. Major difference of lubricant viscosity is pressure peak value. Fluid pressure becomes higher as lubricant viscosity increases. Bearing clearance mainly affects outer film pressure distribution on axial direction especially for lower half of bearing transforming into ellipse. Fluid film pressure drops sharply as difference between semi-major axis and semi-minor axia of ellipse increase because large clearance make it difficult to form squeeze film.
Figure 12. Pressure distribution on circumferential direction: (L) outer film (R) inner film

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