Effect of design parameters on the noise of rotor-bearing systems

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Abstract

The purpose of the present paper is to investigate the effects of design parameters on the noise produced by rotor-bearing systems supported by oil lubricated journal bearings. To do this, the effects of radial clearance and the width of the bearing, lubricant viscosity, and mass eccentricity of the rotor on the noise of the bearing are investigated, and the numerical results are presented through the graph of the A-weighted sound pressure level of the bearing for various rotational speeds of the rotor. The results show that the A-weighted sound pressure level of the bearing is markedly influenced by the mass eccentricity of the rotor and the radial clearance and the width of the bearing. High viscosity of the lubricant slightly decreases the noise of the bearing, but its effect is relatively low at high speed. The results of the paper could aid in the design of low-noise rotor-bearing systems supported by oil lubricated journal bearings.

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Keywords: Hydrodynamic journal bearing; Imbalance; Noise; Rotor-bearing systems

1. Introduction

The bearings used for supporting rotating machinery are one of the crucial elements by which the safe operation of the machinery is ensured. In recent years, with continuing demands for increased performance, many rotating industrial machines are now being designed for operation at high speed, a trend which has resulted in increased mechanical vibration and noise problems. Many researchers have studied the vibration characteristics of bearings [1–3], but there is relatively little information regarding their acoustical properties. Most of the studies that have examined acoustic properties have focused on acoustic emission (f > ~ 100 kHz) only to monitor and detect damages in ball and journal bearings [4–5]. Therefore, in the range of audio frequency, bearing acoustical properties should be determined in order to solve bearing associated problems and develop quieter systems. From this point of view, Rho and Kim [6] investigated the acoustical properties of hydrodynamic journal bearings through frequency analysis of oil pressure fluctuation calculated from nonlinear transient analysis. Furthermore, using a transmission theory of plane waves, a method of calculating the noise caused by oil pressure fluctuation in hydrodynamic journal bearings was studied [7]. However, there have been no studies on the effects of design parameters on the noise of rotor-bearing systems. In practice, it is very important to know how much the bearing noise can be reduced by design parameters such as bearing width, radial clearance, oil viscosity, mass eccentricity of the rotor, and so on. In other words, it is very important to know what parameters are dominant on bearing noise. It is also expected that the acoustic properties of the bearing can provide diagnostic information on abnormal phenomena of the rotor-bearing system. For example, if the frequency characteristics and the corresponding variation of oil film pressure from a pressure transducer installed at the inner surface of the bearing were detected and monitored, it is expected that the information might...
provide diagnostic information on abnormal phenomena of the rotor-bearing system.

The purpose of the present paper is to investigate the effects of design parameters on the noise of rotor-bearing systems supported by oil lubricated journal bearings. To do this, the effects of radial clearance and width of bearing, lubricant viscosity, and mass eccentricity of the rotor are examined for various rotational speeds of the rotor.

2. Governing equations

The coordinate system of an oil lubricated journal bearing is shown in Fig. 1. It is assumed that the rotor and bearing are circular and rigid, and the rotor moves only in parallel mode. The bearing load is applied in the x direction, and the axial groove, which is located at the top of the bearing, is filled with a lubricant at constant pressure.

The equations of motion including rotor imbalance for a rotor-bearing system can be written as:

\[ m \ddot{x} = me \Omega^2 \cos \Omega t + f_{px} + W \]  
\[ m \ddot{y} = me \Omega^2 \sin \Omega t + f_{py} \]

where \( m \) is the mass of the rotor, \( x \) and \( y \) are the coordinates of the journal center, \( e \) is the mass eccentricity of the rotor, and \( W \) is the static load of the journal bearing. The reaction forces, \( f_{px} \) and \( f_{py} \), of the oil film can be obtained by integrating the oil film pressure of the bearing.

The Reynolds equation governing the pressure distribution of the oil film in a finite width bearing under unsteady conditions can be written as follows:

\[ \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( h^3 \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial p}{\partial z} \right) = 6 \mu \Omega \frac{\partial h}{\partial \theta} + 12 \mu \frac{\partial h}{\partial t} \]

The lubricant is assumed to cavitate at ambient pressure and supply at zero gauge pressure. In addition, the Reynolds condition at the film rupture boundary and

### Nomenclature

- \( A \): surface area of the bearing (m\(^2\))
- \( c_a \): speed of sound in air (m/s)
- \( c_s \): speed of sound in bearing (m/s)
- \( e \): mass eccentricity of rotor (kg)
- \( f \): frequency (Hz)
- \( f_{px}, f_{py} \): fluid film reaction forces in x and y directions (N)
- \( h \): film thickness (m)
- \( m \): mass of rotor (kg)
- \( N \): sound pressure level at outer surface of bearing (dB, dB A)
- \( N_b \): averaged sound pressure level of bearing (dB, dB A)
- \( p \): oil pressure (N/m\(^2\))
- \( p_f \): oil pressure fluctuation (N/m\(^2\))
- \( p_{fi} \): pressure of incident wave at outer surface of bearing (N/m\(^2\))
- \( p_{fr} \): pressure of reflected wave at outer surface of bearing (N/m\(^2\))
- \( p_m \): mean pressure of oil (N/m\(^2\))
- \( p_{ref} \): reference sound pressure (N/m\(^2\))
- \( R \): inner radius of bearing (m)
- \( R_o \): outer radius of bearing (m)
- \( T \): period of steady state response (s)
- \( t \): time (s)
- \( W \): static load of journal bearing (N)
- \( x, y, z \): coordinates (m)
- \( Z_a \): acoustic impedance of air (kg/m\(^2\) s)
- \( Z_s \): acoustic impedance of bearing (kg/m\(^2\) s)
- \( \mu \): oil dynamic viscosity (kg/m s)
- \( \theta \): angular coordinate (rad)
- \( \rho_a \): density of air (kg/m\(^3\))
- \( \rho_s \): density of bearing (kg/m\(^3\))
- \( \Omega \): rotational angular velocity of journal (rad/s)

\( 600 \)  
the JFO condition at the film reformation boundary are adopted to predict the cavitation region of the lubricant in the bearing. A cavitation algorithm [8] is used to implement the above cavitation boundary conditions, and the finite difference method, together with the column method [9], is also used for the analysis. The numbers of grid points in the circumferential and axial directions for half of each fluid film are 151 and 23, respectively.

3. Sound pressure level

For the sake of simplicity, it is assumed that the acoustic wave for the oil pressure fluctuation is transmitted in the radial direction through the bearing, as shown in Fig. 2, and the acoustic energy loss in the bearing is neglected. In addition, the effect of the cavitation noise generated in the cavitation region was neglected in predicting the bearing noise transmitted at the outer surface of the bearing. Because the acoustic characteristic impedance of the bearing is very much higher than that of the cavity, the noise generated in the cavitation region can be nearly reflected at the inner surface of the bearing.

The averaged sound pressure level transmitted to air in the radial direction at the outer surface of the bearing can be defined as [7]:

\[ N_b = 10 \log \left[ \frac{1}{A} \int_A 10^{0.1N} \, \text{d}A \right] \]  

(4)

where \( N \) is the sound pressure level transmitted to air.
in the radial direction at the outer surface of the bearing, and it can be expressed as:

\[ N = 20 \log \left( \frac{p_{ft}}{p_{ref}} \right) \]  

where \( p_{ft} \) is the pressure amplitude of the wave transmitted to air at the outer surface of the bearing, and \( p_{ref} \) is the reference sound pressure, standardized at \( 20 \times 10^{-6} \text{ N/m}^2 \).

The transmitted wave \( p_{ft} \) retains the propagation direction of the incident wave \( p_{fi} \), and the following relationship can be written [10]:

\[ p_{ft} = p_{fi} \frac{2Z_a}{Z_s + Z_a} \]  

where \( Z_s = \rho_s c_s \) and \( Z_a = \rho_a c_a \) are the characteristic acoustic impedance of bearing and air, respectively, and \( \rho \) and \( c \) are the density and speed of sound of the medium.

Considering the thickness of the bearing, the pressure amplitude of the incident wave at the outer surface of the bearing can be expressed as:

\[ \left( \frac{p_{ft}}{p_f} \right)^2 = \left( \frac{R}{R_o} \right) \]  

where \( p_f \) is the pressure amplitude of the oil pressure fluctuation of the bearing and \( R_o \) is the outer radius of the bearing.

For the steady state response obtained from the nonlinear transient analysis, the pressure amplitude of the oil pressure fluctuation can be obtained in root mean square form as follows:

\[ p_t = \sqrt{\frac{1}{T} \int_0^T (p - p_m)^2 \, dt} \]  

where \( p_m \) is the time averaged oil film pressure for the steady state response of the rotor.

4. Results and discussion

The specifications of the bearing and parameter values are listed in Table 1. For a given set of operating conditions, a two-dimensional Newton–Raphson search technique was used to calculate the static equilibrium position of the rotor. Starting from the static equilibrium position of the rotor, the nonlinear transient response of the rotor centre was obtained by numerical integration of its acceleration using the fourth order Runge–Kutta method. In the case of the rotor-bearing system considered here, the resonance of the system appears to be about 4000 rpm, and the instability threshold speed about 7500 rpm. In order to investigate the effects of design parameters on the noise of the rotor-bearing system, numerical results of the

![Fig. 3. Sound pressure level change with respect to mass eccentricity of the rotor.](image)

![Fig. 4. Sound pressure level change with respect to viscosity of the lubricant.](image)
parametric studies are presented through graphs of the A-weighted sound pressure level of the bearing for various rotational speeds of the rotor. The A-weighted sound pressure level is widely used to estimate the probability of hearing damage in industry, and it can be correlated with the annoyance caused by rotating machinery [10].

Fig. 3 shows the effects of the mass eccentricity of the rotor on the A-weighted sound pressure level of the bearing for various rotational speeds. The results show that the mass eccentricity of the rotor increases the sound pressure level of the bearing, as is expected. This is readily explained by the increase in vibration amplitude of the rotor as the mass eccentricity of the rotor increases, as is well known.

Fig. 4 shows the effects of the viscosity of lubricants on the A-weighted sound pressure level of the bearing for various operational speeds. The results show that high viscosity of the lubricant decreases the sound pressure level of the bearing, but its effects are relatively low at speeds above 5000 rpm. This is also explained by the increase in the vibration amplitude of the rotor with respect to the lubricant viscosity. To explain the results, the vibration amplitude of the rotor with respect to the lubricant viscosity is shown in Fig. 5. The results show that the vibration amplitude changes of the rotor with respect to the lubricant viscosity are relatively low at speeds above 5000 rpm. In general, the noise is closely related to the vibration. That is, it is expected that the bearing noise decreases as the rotor vibration decreases. However, according to Figs. 4 and 5, the results also show that the rotor vibration decreases, but the bearing noise increases as the rotational speed of the rotor increases above the critical speed of the system. That is, the bearing noise can be increased even if the rotor vibration decreases as the rotational speed of the rotor increases.

Fig. 5. Vibration amplitude change with respect to viscosity of the lubricant.

Fig. 6. Sound pressure level change with respect to width of the bearing.

Fig. 7. Vibration amplitude of the rotor and averaged area ratio of full film region.
Fig. 6 shows the effects of the width of the bearing on the A-weighted sound pressure level of the bearing for various operational speeds. As a general rule, the results show that the A-weighted sound pressure level of the bearing decreases as the width of the bearing increases. However, in the case of the bearing widths of 24 and 40 mm, the noise of the bearings 24 mm in width is lower than that of bearings with a width of 40 mm when the rotor rotates at above 7000 rpm. This is not surprising, considering the average area of the full film region by which the oil pressure fluctuation can be generated in the bearing for the steady state response of the rotor. In general, the vibration amplitude of the rotor decreases, but the average area of the full film region increases as the width of the bearing increases. Thus, the noise of the bearings 40 mm in width can be greater than that of bearings 24 mm in width due to the increase in the area of the full film region. To explain the results, the vibration amplitude of the rotor and the averaged area ratio of the full film region with respect to the bearing widths of both 24 and 40 mm for various operational speeds are shown in Fig. 7. The results show that, in the case of bearings 40 mm in width, the average area ratio of the full film region is larger than that of bearings 24 mm in width, although the vibration amplitude of the rotor is lower than that of bearings 24 mm in width at speeds above 6000 rpm.

Fig. 8 shows the effects of radial clearance of bearings on the A-weighted sound pressure level of bearings for various operational speeds. The vibration amplitude of the rotor with respect to the radial clearance of the bearings for various operational speeds is shown in Fig. 9. The results show that the A-weighted sound pressure level of the bearings and the vibration amplitude of the rotor increase as the radial clearance of the bearings increase.

5. Conclusion

In order to investigate the effects of design parameters on the noise of rotor-bearing systems supported by oil lubricated journal bearings, the effects of radial clearance and width of bearing, lubricant viscosity, and mass eccentricity of the rotor were numerically investigated for various rotational speeds. It is found that, as a general rule, the noise of the bearing decreases as the mass eccentricity of the rotor decreases, the lubricant viscosity increases, the width of the bearing increases, and the radial clearance of the bearing decreases. The results and discussion presented in this paper could aid in the design of a low-noise rotor-bearing system supported in oil lubricated journal bearings.

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