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Noise of Cylindrical Roller Bearings with Low Preload Assembly

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***Abstract*— A test rig was developed to reproduce acoustic noise in cylindrical roller bearings (CRBs) with a large radial clearance operated under low-speed, low-preload conditions. Experimental results showed that noise emerged at 2500 rpm and disappeared at 1500 rpm. FFT analysis revealed a correlation between the shafts speed, outer ring vibration, and noise peak frequencies. The outer ring vibration plays a role in noise generation.**

Keywords—Cylindrical roller bearing, Acoustic noise, Oil lubrication, Radial clearance, Low-preload

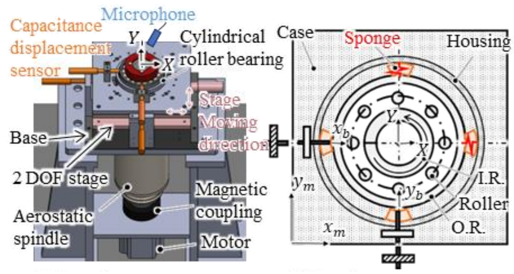
I. Introduction

In industrial equipment, the radial clearance of cylindrical roller bearings (CRBs) is sometimes set to a larger value to simplify assembly [1]. This design relies on the reduction of internal clearance caused by the radial expansion of the inner ring (I.R.) and the outer ring (O.R.) due to centrifugal force at high rotational speeds. At these speeds, the expansion creates sufficient preload, which ensures proper fixation of the bearing to the rotating shaft and housing [1, 2]. However, this approach does not reduce the bearing clearance at lower rotational speeds, leading to a low preload and potentially causing abnormal acoustic noise [3, 4].

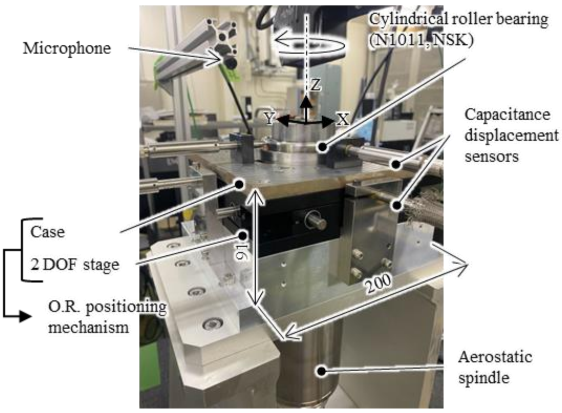
To address this issue, a prototype device was developed to simulate the noise generation in CRB at the low shaft speed condition. This comprehensive study focuses on reproducing abnormal acoustic noise and conducting vibration measurements during noise events.

II. Development of noise reproduction equipment

A test rig was designed and assembled to reproduce the abnormal acoustic noise when a cylindrical roller bearing with a large radial clearance is used under low-speed and low-preload conditions. Fig. 1 shows the noise reproduction test rig. Fig. 1(a) provides an overview of the setup. I.R. of the CRB was mounted on an aerostatic spindle to ensure high-precision rotation. A designed magnetic coupling was placed between the motor and the air spindle to prevent high-frequency torque ripple transmission. Fig. 1(b) shows the O.R. positioning mechanism using a 2-degree-of-freedom (2DOF) stage. Since the eccentricity of O.R. is considered a crucial factor in vibration and noise generation, the O.R. was softly supported with a sponge to minimize the influence of the stator structure. The 2-DOF stage consists of a planar 2-axis linear table, enabling in-plane positioning of O.R. with an accuracy of about 2 µm.



(a) Over view (b) Top view



(c)Fabricated and assembled equipment.

Fig. 1 Experimental setup.

O.R. displacements were measured as its relative movement within the sponge tape by two capacitance displacement sensors (Microsence, 6810) mounted on the 2-DOF stage. The measurement range of the capacitance displacement sensors is 50 µm with a resolution of under 10 nm. Due to the 2-DOF stage's low rigidity and lack of a locking mechanism, the stage could move relative to the shaft under load, so displacement measurements were taken for the stage during bearing operation by the same type of capacitance displacement sensors. Before each experiment, precise coaxial alignment of the I.R. and the O.R. was required to set the eccentric state. A digital dial gauge (SURUGA Seiki) with a 0.1 µm resolution was used during assembly to ensure this alignment. Fig. 1(c) shows the fabricated and assembled measurement test rig. After assembling the components, excluding the sensors, the 2-DOF stage was adjusted while confirming coaxial alignment with the dial gauge, and then the capacitance displacement sensors were installed. A microphone (GRAS, 146AE) was used to measure the noise.

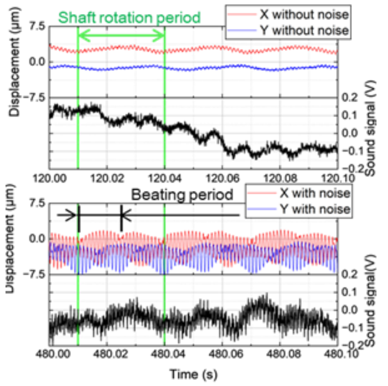


Fig. 2 Comparison of the outer ring displacement and bearing noise at speed of 2000 rpm: (a) No noise during increasing rotational speed (b) Noise observed during rotational speed decrease.

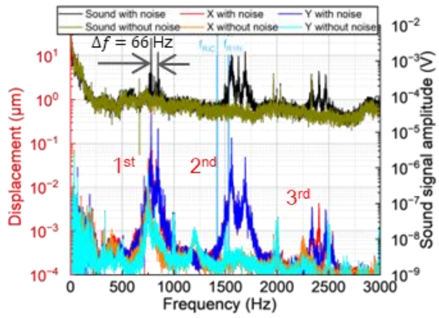


Fig. 3 FFT results showing frequency response differences at the same shaft speed of 2000 rpm, with and without noise.

Since the offset cannot be determined when attaching the sensors, the dial gauge reading is used as the eccentricity offset. x_0 and y_0 are the dial gauge readings, x_b and y_b are the relative displacement of the O.R., and x_m and y_m are the displacement of the 2-DOF stage. The displacement of the O.R. relative to the I.R., X and Y, is given by Equation (1).

$$(X, Y) = (x_0 + x_m + x_b, y_0 + y_m + y_b) \tag{1}$$

III. Experiment

A. Experimental setup

The CRB (N1011, NSK) was mounted, and the coaxiality of the I.R. and O.R. was adjusted to approximately 1 μm using a dial gauge and positioning mechanism. Radial clearance was measured with a dial gauge, and the value was 15 μm. Then, 0.2 mL of lubricant (VG32) was applied via drop lubrication to CRB. The shaft speed was swept from 1500 rpm to 2750 rpm, and then reduced in 250 rpm increments to 1250 rpm. Sound pressure and O.R. displacements were measured at each speed for 20 seconds with a sampling frequency of 40 kHz, followed by FFT analysis.

B. Result and discussion

Fabricated equipment setup successfully reproduced the noise like that observed in actual operating. The noise was first detected during the sweep-up at a spindle speed of 2500 rpm. The noise persisted thereafter and disappeared during the sweep-down at a spindle speed of 1500 rpm. Fig. 2(a) represents the results without noise, while Fig. 2(b) shows the results with noise. Comparing the two, it is evident that during the noise occurrence, the vibration amplitude of the O.R. significantly increased, exhibiting a beating vibration. The sound pressure waveform also showed similar behavior. Additionally, it was found that the position of the O.R. changed over time, suggesting that the O.R. position is related to the occurrence of the noise.

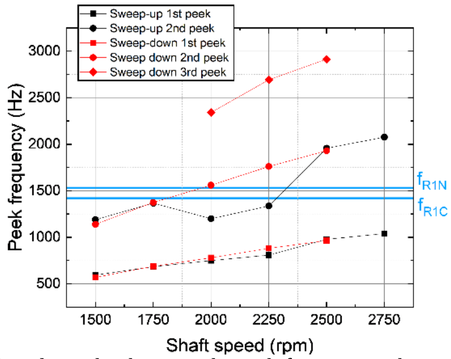


Fig. 4 The relationship between the peek frequencies obtained from the experimental results to the shaft rotation speed.

Fig. 3 shows the FFT analysis results of the O.R. displacement and noise. In the case of noise, peaks appeared in the sound pressure signal. Since the frequencies of these peaks are multiples of the first peak frequency, the first peak is referred to as the fundamental frequency (1st), and the subsequent peaks are the 2nd and 3rd harmonics. At all peak frequencies, with noise, the O.R. vibration amplitude also significantly increased, suggesting that the O.R. vibration is likely the main cause of the noise. Additionally, the beating frequency Δf was 66 Hz at a shaft speed of 2000 rpm.

Fig. 4 presents peak frequencies against the shaft speed. f_{R1N} and f_{R1C} represent the in-plane elastic natural frequency of O.R. and cage under free-free conditions obtained by FEM, while other elastic modes had much higher frequency. Fig. 4 shows a clear positive correlation between the peak frequency and shaft speed. We considered that the noise was caused not by the elastic vibrations of the components, but rather by the hydrodynamic effect of the oil film formed between the rolling elements [5].

IV. Conclusion and Future Plan

We experimentally simulated acoustic noise from a CRB with a large radial clearance, designed for high-speed machinery, under low preload and low-speed conditions. The test rig successfully reproduced the noise generation. Experiments revealed that the noise appeared during the sweep-up at 2500 rpm and persisted until disappearing during sweep-down at 1500 rpm. The results showed a significant increase in vibration amplitude of the O.R., and a clear correlation between the shaft speed and the noise peak frequency, implying that the noise was caused by O.R. vibration. We consider the noise is influenced by the hydrodynamic effect of the oil film. As a prospect, the relationship between O.R. position and noise occurrence will be studied. Focusing on peak frequencies, a dynamic bearing model, considering the oil film will be developed and compared with experimental results.

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