Squealing of Cylindrical Roller Bearing*

Mikio NAKAI**, Masayuki YOKOI***,
Masaru INOUE**** and Keizo KAWAKAMI*****

This paper describes an investigation into squealing sounds which occur in a large-sized cylindrical bearing under two testing conditions, a single bearing and a bearing fitted into an annular housing by an interference. Squeal of 4-6 kHz for the single bearing and two kinds of squeals of approximately 5 kHz and 6 kHz for the bearing fitted into the housing occur at the shaft speeds of 1 000-1 600 rpm. These squeal frequencies are not equal to the natural frequencies of the in-plane vibration of the outer race or of the housing including the outer race. The natural frequencies of the in-plane vibration of the outer race coupled with the stationary rollers were calculated, replacing the roller/outer race contact in the loading zone by a spring and considering the friction between the roller and the outer race. As a result, the squeal frequencies approximately agree with the natural frequencies, which indicate an oscillatory instability, and also are close to the natural frequencies of a single roller with a spring.

Key Words: Rolling Bearing, Self-Excited Vibration, Coupled Vibration, Sound, Frictional Vibration, Squeal, Ring

1. Introduction

Large-sized cylindrical roller bearings often emit high-level squealing sounds in grease lubrication. The squealing sound does not affect the function of the bearing, but it is a great nuisance.

Many experiments have been performed on this squealing sound by Iida⁽¹⁾, Ito⁽²⁾ and Kobayashi and Kurita⁽³⁾. Iida stated that the squeal frequencies are equal to the in-plane natural frequencies of the outer race. Igarashi⁽⁴⁾ and Nagamatsu and Fukuda⁽⁵⁾ studied noise in the ball bearing fitted into an annular housing. In their studies, frequency components of noise are also equal to the in-plane natural frequencies of the

outer race, although the bearing is fitted into the housing. No theoretical study has been done on this squealing sound. Comprehension of this sound is hence far from complete.

In this paper, we study the locations of the origins of the squealing sounds for the bearing under two testing conditions, only a single bearing and a bearing fitted into an annular housing, and investigate squeal frequency, considering the coupled vibration between the roller and the outer race.

2. Experimental Apparatus and Measurement Procedures

2.1 A single bearing

The experimental apparatus is shown in Fig. 1. Experiments were performed on a type NU 314 cylindrical roller bearing ① (inner radius : 70 mm, outer radius : 150 mm, width : 35 mm) with 13 rollers (diameter : 20 mm, length : 20 mm). Its radial clearance is very large (103 μ m) in order to easily produce squeal. The inner race ④ of the cylindrical roller bearing is assembled into a shaft ③ supported by two ball bearings ① and ②. Radial load is applied to the outer race ⑤ by an adjustable screw ⑦ attached to a

* Received 10 th September, 1990. Paper No.89-0637 A, 0899 B

** Faculty of Engineering, Kyoto University, Yoshidahonmachi, Sakyo-ku, Kyoto 606, Japan

*** Junior College, Osaka Sangyo University, 3-3-1 Nakagaido, Daitoshi, Osaka 574, Japan

**** Minolta Camera Co., Ltd., 3-91 Daisennishicho, Sakai, Osaka 590, Japan

***** Matsushita Electric Co., Ltd., 1006 Kadoma, Kadomashi, Osaka 571, Japan

plate spring ⑥. Sound pressure was measured by a 1/2" microphone located at a distance of about 20 cm from the center of the bearing. The accelerations of the outer race were measured by three light accelerometers located at three points A, B and C, where A : just above the outer race ; B : the inlet to the loading zone ; C : the outlet from the loading zone (Fig. 1).

To determine the locations of the origins of squeal, we measured sound intensity. The sound intensity radiated from the front of the bearing was measured in 7×7 fixed points with a distance of 2 cm and at a distance of 1 cm from the end face of the bearing.

2.2 A bearing fitted into the housing

Figure 2 shows the experimental apparatus and dimensions of annular housing. The housing is suspended from the frame by an adjustable coil spring through a supporting rod. The outer race is fitted into the housing with an interference of $12 \mu\text{m}$. Constant load is applied by the coil spring.

Fig. 1 Experimental apparatus for a single bearing and locations of accelerometers

Fig. 2 Experimental apparatus for a bearing fitted into housing and dimensions of housing

3. Experimental Results

3.1 A single bearing

We increased the shaft speeds from 400 rpm to 1500 rpm under a load of 980 N. Figures 3(a) and (b) show averaged spectra of sound and in-plane acceleration of the outer race at point A every 100 rpm. At a low shaft speed, sound levels are low and their frequency components are about 1 kHz. Sound levels increase with the increase of the shaft speed. When the shaft speed exceeds 1000 rpm, squeal begins to occur, and the sound levels at 4-6 kHz increase. These frequency components seem to be the main squeal frequencies. However, the acceleration levels at 4-6 kHz are lower than that at 8 kHz. This result is the same as at points B and C. Figure 4 shows spectra of the out-of-plane acceleration A of the outer race at the shaft speeds of 400-1400 rpm every 200 rpm. The acceleration levels at 2650 Hz and 7650 Hz are high at any shaft speed.

Fig. 3 Spectra of sound and in-plane acceleration of the outer race at point A versus different shaft speeds

Fig. 4 Spectra of out-of-plane acceleration of the outer race

Fig. 5 Spectra of sound and in-plane acceleration at 500 and 1400 rpm

First, to examine these frequencies in detail, Fig. 5 shows spectra of sound and in-plane accelerations of points A, B and C at the shaft speeds of 500 rpm for the absence of squeal and of 1400 rpm for squeal occurrence. As can be seen, sound levels of the main squeal frequencies of 4–6 kHz at 1400 rpm are markedly high compared with those at 500 rpm. Also included in Fig. 5 are mode shapes of the outer race measured through a narrow band-pass filter. For the in-plane vibration at 500 rpm, the modes associated with peak frequencies at 1 kHz, 1.6–1.9 kHz, 5 kHz and 8 kHz have 1–4 nodal diameters, respectively. For the out-of-plane vibration, the modes associated with two peak frequencies at 2.7 kHz and 7.5 kHz have two and three nodal diameters. However, since the acceleration levels at the main squeal frequencies of 4–6 kHz showed a violent fluctuation, we could not identify their mode shapes.

Figure 6(a) presents peak frequencies in the acceleration spectra of Fig. 3(b) as a function of shaft speed. When the shaft speed is more than 1000 rpm, the main squeal frequencies fluctuate between 4 and 6 kHz, but the other peak frequencies are constant. To determine the sources of these peak frequencies, only the outer race was suspended at one point with nylon strings and impacted in both the in-

Fig. 6

and out-of-planes by a hammer. These frequency response functions and mode shapes associated with natural frequencies are presented in Fig. 6(b). By comparison between Figs. 6(a) and (b), it can be found that the out-of-plane peak frequencies do not agree with the main squeal frequencies, but with the out-of-plane natural frequencies of only the outer race. Also, the in-plane natural frequencies of the outer race do not agree with the main squeal frequencies. This differs from the previous results⁽¹⁾ in which squeal frequencies are equal to the in-plane natural frequencies of only the outer race. Furthermore, the mode with one nodal diameter of the in-plane vibration is not observed in Fig. 6(b) for the free vibration of only the outer race, but is observed in Fig. 6(a). This indicates that the mode may be caused by the coupled vibration due to the contact between the roller and the outer race.

Next, we investigate the influence of radial load and cage material on the squeal frequencies. Figure 7 shows spectra of the squealing sound under different loads of 490–1 490 N. From these results, it can be concluded that the main squeal frequencies do not depend on the load. Figure 8 shows spectra of the squealing sound for brass and bakelite cages. The natural frequency of the bakelite cage is 700 Hz, which is lower than 4 525 Hz of the natural frequency

Fig. 7 Effects of radial loads on squeal frequency

Fig. 8 Effects of cage material on squeal frequency

of the brass cage. However, the squeal frequencies are the same for both cages. That means that cage material has no effect on squeal frequencies.

Lastly, Fig. 9 shows the sound intensity vector map at the frequency range of 4.5–6 kHz. There are three squealing sound sources located between the outer and the inner races: two sources where the roller enters and exits the loading zone, and one source at the upper unloading zone. Squeals at the loading zone are produced by the outer race/roller slip and the squeal at the upper unloading zone may occur because the roller impacts or rubs the outer race under the influence of the centrifugal force of the roller.

3.2 A bearing fitted into the housing

Figure 10 shows averaged spectra of sound at a shaft speed range from 900 to 2 000 rpm under a load of 980 N. When the shaft speed exceeds 1 000 rpm, squeal begins to occur and is dominated by about 5 kHz. To examine this frequency, spectra of squealing sound at 1 400 rpm are presented in Fig. 11. The spectrum of sound is also included for comparison in Fig. 11 when squealing stopped after pouring lubricating oil into the bearing. It is obvious from Fig. 11 that the sound level at a main squeal frequency of about 5 kHz is 20 dB higher than that for the absence of squeal. In this figure, mode shapes of the housing are also shown, where n indicates the number of nodal diameter.

We examine the relationship among these peak

Fig. 9 Sound intensity flow in squeal occurrence (: sources)

Fig. 10 Spectra of sound at different shaft speeds

frequencies and the natural frequencies of the outer race or the housing including the outer race. The free housing including the outer race was impacted by a hammer. Frequency response functions for both in- and out-of-plane vibrations and their mode shapes are shown in Fig. 12. Natural frequencies for $n=0\sim3$ modes coincide approximately with peak frequencies in Fig. 11. Furthermore, the natural frequencies of only the outer race, as shown in Fig. 6(b), do not exist in Fig. 11. These results suggest that the vibration for a housing including an outer race should be considered as a single body if the outer race is fitted into the housing with the same interference as in this paper. Furthermore, it can be seen from Figs. 6(b), 11 and 12 that a main squeal frequency of about 5 kHz does not agree with any natural frequency. The other squeal is sometimes observed under the same conditions as in Fig. 11. This sound spectrum is presented in Fig. 13. The main squeal frequency is approximately 6 kHz higher than that in Fig. 11.

The results mentioned above show that two kinds of squeals of about 5 kHz and 6 kHz occur, and squeal frequencies for the bearing fitted into an annular housing do not coincide with the natural frequencies of the housing including the outer race or of only the outer race. The natural frequency of the $n=1$ mode is

Fig. 11 Spectra of sounds for squeal occurrence and for the absence of squeal

Fig. 12 Frequency response functions of housing including outer race by hammering excitation

also observed in Fig. 11 as observed in Fig. 6(a). In order to investigate the squealing mechanism, we should thus consider the coupled vibration between the roller and the outer race fitted into the housing.

4. Analyses and Discussion

According to the experiments, squeal frequency does not depend on the out-of-plane vibration of the outer race or the cage material, but on the coupled vibration between the roller and the outer race. Assuming that the displacement of the inner race assembled into the shaft becomes small enough to be neglected, we investigate the squeal frequency by considering the coupled vibration.

4.1 Basic equations for outer race or housing including outer race

We assume that the outer race or the housing including the outer race is a ring of radius R , thickness h and width b . As the coordinate system is shown in Fig. 14, φ is taken in the circumferential direction of the ring and z in the normal to the middle surface, measured positive outward from the middle surface of the ring. The radial and circumferential displacements in z and φ coordinates on the middle surface from equilibrium are denoted by w and v , and the displacements w_z and v_z on the coordinate z normal to the middle surface are approximated by the forms

$$w_z = w, \quad v_z = v + az. \quad (1)$$

Substituting Eq. (1) into the basic equations of motion for a ring derived by the theory⁽⁶⁾ which contains the effects of shear deformation and rotatory inertia yields:

$$\xi(v'' + w') - \frac{D}{R^2}a'' + \eta\left(a - \frac{v}{R} + \frac{w'}{R}\right) + R p_\varphi = \rho h(Rv'' + h^2 a''/12) \quad (2)$$

Fig. 13 Spectrum of squealing sound with high frequency

Fig. 14 Coordinate system and displacements

$$\eta \left(\alpha' - \frac{v'}{R} + \frac{w''}{R} \right) - \xi (v' + w) + \frac{D}{R^2} \alpha' + R p_z = \rho R h w'' \quad (3)$$

$$\frac{D}{R} \left(-\frac{v''}{R} - \frac{w'}{R} + \alpha'' \right) - R \eta \left(\alpha - \frac{v}{R} + \frac{w'}{R} \right) + R m_\varphi = \rho h^3 (v'' + R \alpha'') / 12, \quad (4)$$

where $(\quad)' = \partial(\quad)/\partial t$, $(\quad)'' = \partial(\quad)/\partial \varphi$, $\xi = (N/R + D/R^3)$, $N = Eh$, $D = Eh^3/12$, $\eta = k^2 Gh(1 + h^2/12R^2)$; E : Young's modulus; G : shear modulus; k^2 : shear coefficient; ρ : the mass density; p_z, p_φ : the external forces acting along z and φ coordinates per unit width, respectively; m_φ : the bending moment acting on the middle surface per unit width.

4.2 Equations of motion for the rollers

Consider that l rollers (l : arbitrary integer) are in contact with the outer race and that the contact stiffness between the i -th roller and the outer race is approximately replaced by a spring with spring constant k_i , as shown schematically in Fig. 15.

If the i -th roller of mass m is in contact with the outer race on $\varphi = \varphi_0$, the equation of motion for the roller on the z coordinate is

$$m u_i'' + k_i u_i - k_i \{(w)_{\varphi=\varphi_i} - u_i\} = 0 \quad (i=1, \dots, l) \quad (5)$$

where u_i : the radial displacement of the i -th roller from equilibrium; $(w)_{\varphi=\varphi_i}$: the radial displacement of the ring on the $\varphi = \varphi_i$. Hence, the external force p_{z1} acting along the z coordinate of the ring per unit width due to l rollers is written by

$$p_{z1} = - \sum_{i=1}^l k_i \{(w)_{\varphi=\varphi_i} - u_i\} \delta(\varphi - \varphi_i) / (Rb), \quad (6)$$

where δ : Dirac's delta function.

In order to support the bearing, supporting equipment is attached to the outer race or the housing.

(i) A single bearing

Supporting equipment attached to $\varphi = \varphi_0$ of the outer race is replaced by the mass M and spring constants $\bar{k}_1$ and $\bar{k}_2$. Then the equation of motion for the supporting equipment is

$$M \bar{u}'' + \bar{k}_2 \bar{u} - \bar{k}_1 \{(w)_{\varphi=\varphi_0} - \bar{u}\} = 0, \quad (7a)$$

Fig. 15 Coupled vibration between roller and outer race

where $\bar{u}$ is the displacement of M from equilibrium.

(ii) A bearing fitted into the housing

Supporting equipment attached to $\varphi = \varphi_0$ of the upper point of the housing is replaced by the masses M_1 and M_2 , and spring constants $\bar{k}_1$, $\bar{k}_2$ and $\bar{k}_3$. Then the following equations of motion are obtained:

$$\begin{aligned} M_1 \bar{u}_1'' + \bar{k}_2 (\bar{u}_1 - \bar{u}_2) - \bar{k}_1 \{(w)_{\varphi=\varphi_0} - \bar{u}_1\} &= 0 \\ M_2 \bar{u}_2'' + \bar{k}_3 \bar{u}_2 + \bar{k}_1 (\bar{u}_2 - \bar{u}_1) &= 0 \end{aligned} \quad (7b)$$

where $\bar{u}_1$ and $\bar{u}_2$ are displacements of M_1 and M_2 from equilibrium.

The external force P_{z2} acting along the z coordinate of the outer race due to the supporting equipment is given by

$$p_{z2} = - \bar{k}_1 \{(w)_{\varphi=\varphi_0} - \bar{u}_1\} \delta(\varphi - \varphi_0) / (Rb). \quad (8)$$

p_z in Eq. (3) hence becomes

$$p_z = p_{z1} + p_{z2}. \quad (9)$$

Squeal is probably caused by the slip between the roller and the outer or inner race. The frictional force may therefore be produced between them as shown in Fig. 16. The frictional force F_i is then expressed as follows:

$$F_i = - \mu k_i \{(w)_{\varphi=\varphi_i} - u_i\}, \quad (10)$$

where μ is the coefficient of friction.

Furthermore, in addition to p_z , the bending moment m_φ and the external force p_φ in the circumferential direction of the ring due to the frictional force are applied to the ring. That is,

$$m_\varphi = \frac{h}{2} \sum_{i=1}^l F_i \delta(\varphi - \varphi_i) / (Rb) \quad (11)$$

Fig. 16 Frictional force acting on outer race

$$p_\varphi = - \sum_{i=1}^l F_i \delta(\varphi - \varphi_i) / (Rb) \quad (12).$$

4.3 Equation of motion for the coupled vibration between the roller and the ring

The solutions of Eqs. (2)~(4), v , w and α , may be written in the forms

$$\begin{aligned} v &= \sum_{n=0}^{\infty} \{v_{n1}(t) \cos n\varphi + v_{n2}(t) \sin n\varphi\} \\ w &= \sum_{n=0}^{\infty} \{w_{n1}(t) \cos n\varphi + w_{n2}(t) \sin n\varphi\} \\ \alpha &= \sum_{n=0}^{\infty} \{\alpha_{n1}(t) \cos n\varphi + \alpha_{n2}(t) \sin n\varphi\} \end{aligned} \quad (13),$$

where n is the number of nodal diameters of the ring.

Substituting Eqs. (2)~(4) into Eq.(13) and multiplying the resulting equations by $\cos n\varphi$ or $\sin n\varphi$ and integrating them from 0 to 2π yields

$$\begin{aligned} C_{n1}v_{n1} + C_{n2}w_{n2} + C_{n3}\alpha_{n1} + P_{\varphi c} \\ = \rho h(Rv_{n1}'' + h^2\alpha_{n1}'/12) \end{aligned} \quad (14)$$

$$\begin{aligned} C_{n1}v_{n2} - C_{n2}w_{n1} + C_{n3}\alpha_{n2} + P_{\varphi s} \\ = \rho h(Rv_{n2}'' + h^2\alpha_{n2}'/12) \end{aligned} \quad (15)$$

$$\begin{aligned} -C_{n2}v_{n2} + C_{n4}w_{n1} + C_{n5}\alpha_{n2} + P_{zc} \\ = \rho Rhw_{n1}'' \end{aligned} \quad (16)$$

$$\begin{aligned} C_{n2}v_{n1} + C_{n4}w_{n2} - C_{n5}\alpha_{n1} + P_{zs} \\ = \rho Rhw_{n2}'' \end{aligned} \quad (17)$$

$$\begin{aligned} C_{n3}v_{n1} - C_{n5}w_{n2} - C_{n3}R\alpha_{n1} + M_{\varphi c} \\ = \rho h^3(v_{n1}'' + R\alpha_{n1}')/12 \end{aligned} \quad (18)$$

$$\begin{aligned} C_{n3}v_{n2} + C_{n5}w_{n1} - C_{n3}R\alpha_{n2} + M_{\varphi s} \\ = \rho h^3(v_{n2}'' + R\alpha_{n2}')/12, \end{aligned} \quad (19)$$

where

$$P_{zc} = \frac{R}{\pi \varepsilon_n} \int_0^{2\pi} p_z \cos n\varphi d\varphi,$$

$$P_{zs} = \frac{R}{\pi \varepsilon_n} \int_0^{2\pi} p_z \sin n\varphi d\varphi,$$

$P_{\varphi c}$, $P_{\varphi s}$, $M_{\varphi c}$ and $M_{\varphi s}$ can be determined by substituting p_φ or m_φ into the above equations instead of p_z . The above equations can be analytically integrated by the substitution of Eqs. (9), (11) and (12). For example, P_{zc} is obtained as

$$\begin{aligned} P_{zc} = - \frac{1}{\pi \varepsilon_n b} \left[\sum_{i=1}^l k_i \left\{ \sum_{m=0}^{\infty} (w_{m1} \cos m\varphi_i + w_{m2} \sin m\varphi_i) \right. \right. \\ \left. \left. - u_i \right\} \cos n\varphi_i + \bar{k}_1 \left\{ \sum_{m=0}^{\infty} (w_{m1} \cos m\varphi_0 \right. \right. \\ \left. \left. + w_{m2} \sin m\varphi_0) - \bar{u} \right\} \cos n\varphi_0 \right] \end{aligned}$$

$$\varepsilon_n = 1 (n \neq 0) \text{ or } 2 (n = 0), \quad C_{n1} = -(n^2 \xi + \eta/R),$$

$$C_{n2} = n(\xi + \eta/R), \quad C_{n3} = (n^2 D/R^2 + \eta),$$

$$C_{n4} = -(\xi + n^2 \eta/R),$$

$$C_{n5} = n(D/R^2 + \eta).$$

For the value of $n=0$, Eqs. (15), (17) and (19) are excluded from Eqs. (14)~(19).

The contact position between the roller and the outer race is a function of time because of the revolution of the roller. However, assuming that the roller is stationary, we calculate the natural frequencies for the coupled vibration between the roller and the outer race.

The solutions of Eqs. (14)~(19) may now be written as

$$\begin{aligned} v_{n1} &= v_{n1}^* e^{\lambda t}, \quad v_{n2} = v_{n2}^* e^{\lambda t}, \dots, \quad \alpha_{n2} = \alpha_{n2}^* e^{\lambda t}, \\ u_i &= u_i^* e^{\lambda t}, \quad \bar{u} = \bar{u}^* e^{\lambda t}, \quad \bar{u}_1 = \bar{u}_1^* e^{\lambda t}, \quad \bar{u}_2 = \bar{u}_2^* e^{\lambda t}. \end{aligned} \quad (20)$$

From the characteristic equation obtained by substituting Eq.(20) into Eqs. (5), (7) and (14)~(19), eigenvalue λ can be determined. If an eigenvalue λ has a positive real part, the solution will exhibit an exponential growth; in such an unstable solution, a self-excited vibration (squeal) occurs.

4.4 Numerical results

4.4.1 Natural frequencies of the outer race or the housing including the outer race Natural frequencies of the outer race or of the housing including the outer race can be determined by omitting the external forces p_z , p_φ and m_φ in Eqs. (3)~(5) and substituting Eq.(14) into Eqs. (3)~(5). Table 1 lists these results as well as the natural frequencies of the out-of-plane vibration of the ring calculated by Bickford and Maganty's theory⁽⁷⁾ together with experimental data in Figs. 6(b) and 12. The data used in the numerical calculations are $R=6.91\text{cm}$, $h=1.23\text{cm}$, $b=3.5\text{cm}$ (for the outer race), $R=8.43\text{cm}$, $h=4.15\text{cm}$, $b=4.0\text{cm}$ (for the housing including the outer race), $E=206\text{GPa}$, and $k^2=\pi^2/12$.

Good agreement in both results proves that Eqs. (3)~(5) can be applied to such a thick ring as an outer race or a housing including an outer race.

4.4.2 Natural frequency of the coupled system between the roller and the outer race

(I) Only a single bearing

First, we calculate natural frequencies (the imaginary part of $\lambda/2\pi$) by taking the modes with zero through five nodal diameters ($n=0\sim 5$) when the frictional force between the roller and the outer race is not considered. In this calculation, we assume that there are three rollers in the loading zone and that both the second roller and the supporting equipment are located directly below the bearing.

Table 1 Natural frequencies of outer race and housing including outer race

Ring	Vibrational direction	Mode	Theoretical values Hz	Experimental values Hz
Outer race	In-plane vibration	n=2	1587	1575
		n=3	4364	4375
		n=4	8064	8150
	Out-of-plane vibration	n=2	2675	2650
Housing including outer race	In-plane vibration	n=2	3212	3225
		n=3	7898	8000
		n=0	9757	9825
	Out-of-plane vibration	n=2	2857	2850
		n=3	7213	7250
		n=0		

The natural frequencies of the system are plotted in Fig. 17 as a function of the nondimensional spring constant $K_2 = k_2 R^3 / (Db)$ of the second roller for the values of $K_1 = K_3 = K_2$, $M = 0.1$ kg, and $\bar{K}_1 = \bar{K}_2 = 12.3$. In this figure, the natural frequencies $f_{r0} = \sqrt{2k_i/m}/2\pi$ of only one roller are also plotted with a dotted line, when the vibration of the outer race is omitted. Figure 18 represents the mode shapes of the outer race for the value of $K_2 = 10.0$.

Without frictional force, all eigenvalues have negative real parts; no self-excited natural frequency occurs. Compare Fig. 18 with the experimental data (Fig. 5(a)) at a low shaft speed for the absence of squeal. We cannot observe natural frequencies (for

Fig. 17 Natural frequencies versus K_2 without frictional force for a single bearing (thin line : natural frequencies of the system, dotted line : natural frequencies of only a roller)

Fig. 18 Mode shapes of outer race for $K_2 = 10.0$

example, 307, 1 737 Hz) of the modes whose nodes are located directly below the outer race. Forces due to the revolution of the roller always act on the outer race in the loading zone and thus it is difficult to produce vibrational nodes in the loading zone.

For the modes whose nodes are not located directly below the outer race, Table 2 shows amplitudes $u_1 - u_3$ of the roller and $\bar{u}$ of the supporting equipment when the maximum amplitude $w_{\max}$ of the outer race is assumed to be 1.0. For these modes, calculated results show good agreement with the experimental frequencies. That is, (a), (b) and (d) in Fig. 5(a) correspond to 795 Hz, 1 902 Hz and 8 258 Hz, respectively, and (c) close to the natural frequencies of only the roller correspond to 4 208 Hz, 4 452 Hz, and 5 029 Hz, where the amplitudes $u_1 - u_3$ of the roller are large; see Table 2. Further, a frequency at 3 459 Hz observed at a higher shaft speed (see Fig. 6(a)) is the natural frequency of the supporting equipment, where the amplitude $\bar{u}$ of the supporting equipment is large.

Next, Fig. 19 shows natural frequencies as a function of K_2 when we consider the roller/outer race frictional force. Since a metallic contact between the

Table 2 Amplitudes of rollers and supporting equipment at $w_{\max} = 1.0$

Freq. Hz	795	1902	3459	4208	4452	5029	8258
$u_1 - u_3$	0.23	0.41	0.50	0.12	5.26	0.64	0.06
u_2	0.19	0.61	0.23	4.40	1.67	1.74	0.20
$\bar{u}$	0.19	0.71	3.82	0.97	0.02	0.47	0.11

Fig. 19 Natural frequencies with frictional force (thin line : natural frequencies of the system, thick line : self-excited natural frequencies, dotted line : natural frequencies of only an outer race)

roller and the outer race is partially caused by the roller/race slip in squeal occurrence, we used $\mu=0.5$. In this figure, the thick lines show the region for self-excited frequencies (squeal occurrence region) which have the positive real parts of λ . Also, the corresponding mode shapes calculated by excluding the real part of λ at $t=0$ are shown. Table 3 shows amplitudes u_1-u_3 and $\bar{u}$ at $w_{\max}=1.0$.

The self-excited frequencies are about 3 kHz, 4.5-6 kHz and 8 kHz. In the experimental data (Fig. 5(b)), the main frequency components of squeal were 4-6 kHz. Since the contact condition between the rolling roller and the race changes, the spring constant of the roller also changes. If the value of K_2 changes from 10 to 20, then the self-excited frequencies 4.5-6 kHz in Fig. 19 coincide with the main squeal frequencies. The in-plane acceleration of the outer race at the main squeal frequency components in the experiments was of low level in comparison with the sound. This can be verified from the fact that the displacement of the roller is about three times that of the outer race at 4.5-6 kHz (5758 Hz in Table 3).

Since the rotation slip of the roller becomes lar-

ger near the inlet of the loading zone, we calculated natural frequencies when the frictional force acts only on the first roller of the inlet. From these results shown in Fig. 20, it can be seen that the region for self-excited frequencies becomes narrower than that in Fig. 19.

(II) A bearing fitted into the housing

Figure 21 shows natural frequencies versus $K_2(=k_2R^3/(Db))$ considering frictional force. For comparison with numerical results for a single bearing, the spring constant is transformed into a dimensionless form by using the values of R , D and b for only the outer race. Results have been obtained with the following numerical data: $\mu=0.5$, $K_1=K_3=K_2$, $\bar{K}_1=10.0$, $\bar{K}_2=15.0$, $\bar{K}_3=5.12 \times 10^{-2}$, $M_1=0.98$ kg and $M_2=0.55$ kg. Also, natural frequencies of only the roller are plotted as the A-A curve and the mode shapes of self-excited frequencies are shown. Table 4 shows amplitudes of u_1-u_3 , $\bar{u}_1$ and $\bar{u}_2$ at $w_{\max}=1.0$. From Fig. 21, it is found that self-excited frequencies exist at 200 Hz, 3-4.8 kHz, 5.6-6.2 kHz and 8 kHz. If the value of K_2 changes from 10 to 20, as in the case of a single bearing, then 4.5-4.8 kHz of the mode with four nodes

Table 3 Amplitudes of rollers and supporting equipment at $w_{\max}=1.0$ for self-excited frequencies

K_2	5.0	15.0
Freq. Hz	3110	5758 8325
u_1	2.83	3.42 0.36
u_2	6.64	3.31 0.37
u_3	7.16	3.06 0.39
$\bar{u}$	0.48	0.25 0.11

Fig. 20 Natural frequencies when the frictional force acts on only the first roller

Table 4 Amplitudes of rollers and supporting equipment at $w_{\max}=1.0$ for self-excited frequencies

K_2	5.0	8.0	18.0
Freq. Hz	197	3042	4013 5930 7977
u_1	1.33	7.78	21.5 26.9 0.59
u_2	0.39	10.5	14.7 20.7 0.70
u_3	0.76	12.7	10.6 16.2 0.66
$\bar{u}_1$	0.85	0.15	0.03 0.01 0.01
$\bar{u}_2$	0.87	0.04	0.00 0.00 0.00

Fig. 21 Natural frequencies with frictional force

Fig. 22 Enlargement of the frame in Fig.21

and 5.6–6.2 kHz of the mode with six nodes approximately agree with the main squeal frequencies in Figs. 11 and 13. At these frequencies, the displacements of the rollers are very large in comparison with that of the housing; see Table 4. However, the squeal of self-excited frequency, 8 kHz, hardly occurs, which is obtained from the coupling between two vibration modes of the housing.

Enlargement of the frame in Fig. 21 is shown in Fig. 22. From this figure, it can be seen how two kinds of self-excited frequencies are produced by the coupling between two vibration modes.

5. Conclusions

From the measurements and analyses on the frequency of squeal emitted from the cylindrical roller bearing, the following results are summarized :

(1) Main frequency components of squeal occur over a wide range. However, quite unlike the previous papers, they are not equal to the natural frequencies of the in-plane vibrations of only the outer race in a bearing under two testing conditions, a single bearing and a bearing fitted into an annular housing. Also, they do not depend on the radial load nor on cage material.

(2) Considering the frictional force between the

roller in the loading zone and the outer race and assuming that the roller is stationary yields the self-excited frequencies in the coupled vibrations between the roller and the outer race or in the housing including the outer race. These self-excited frequencies which are very close to the natural frequencies of only a roller with a spring coincide with the main frequency components of squeal. In other words, we can conclude that the main frequency components of squeal are approximately equal to the self-excited frequencies of the coupled system between the roller and the outer race and that their frequencies are very close to the natural frequencies of only a roller with a spring.

Acknowledgment

The authors would like to express their appreciation to Mr. Toru Takahashi of Koyo Co. Ltd. for his helpful suggestions.

References

- (1) Iida, K., The Study of Noise in Cylindrical Roller Bearings, J. Jpn. Soc. Lubr. Eng., (in Japanese), Vol. 4, No. 3(1959), p. 114.
- (2) Ito, S., Extreme Noise in Cylindrical Roller Bearing Attributed to Lubricant Resistance, J. Jpn. Soc. Lubr. Eng., (in Japanese), Vol. 4, No. 4 (1959), p. 178.
- (3) Kobayashi, M. and Kurita, M., Lubrication Effects on Noise in Cylindrical Roller Bearings, J. Jpn. Soc. Lubr. Eng., (in Japanese), Vol. 17, No. 9 (1972), p. 554.
- (4) Igarashi, A., Noise of Ball Bearing (3rd Report, Case of Fitted Ball Bearing), Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 27, No. 178(1961), p. 963.
- (5) Nagamatsu, A. and Fukuda, M., A Study on Sound Noise Generated by Ball Bearing in High Speed Rotation, Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 43, No. 376(1972), p. 4661.
- (6) Mirsky, I. and Herrmann, G., Nonaxially Symmetric Motions of Cylindrical Shells, J. Acoust. Soc. Am., Vol. 29, No. 10(1957), p. 1116.
- (7) Bickford, W.B. and Maganty, S. P., On the Out-of-Plane Vibrations of Thick Rings, J. Sound & Vibr., Vol. 108, No. 3(1986), p. 503.