Influence of Surface Roughness on Sliding Characteristics of Rubber Seals

K. YAMAMOTO　D. OZAKI　T. NAKAGAWA

The influence of surface roughness on frictional vibration of a bearing seal has been studied to establish the method for identifying generation mechanism of frictional vibration. As a result, the degree of influence on frictional vibration could be clarified by the power spectrum arising from the roughness of sliding surface.

Stability of frictional vibration was examined by using viscous coefficient obtained from FEM analysis, contact load and μ - V curve of rubber material. The result was approximately consistent with actual measurement results.

***Key Words**: seal, frictional vibration, surface roughness, power spectrum*

1. Introduction

Rubber seals, including bearing seals, are liable to generate abnormal noise due to frictional vibration caused by stick-slip (hereinafter referred to as "S-S") when seals slide under starved lubrication conditions.

Generally, rubber materials tend to generate S-S because its coefficient of friction declines with the increase of sliding speed. And noise is generated when frictional vibration due to S-S is unstable (self-excited vibration), and is influenced by the change of frictional coefficient relative to the speed including the effect of surface roughness, viscosity of the material and contact load. There have been many papers reported concerning the analysis of this mechanism of frictional vibration generation[1].

However, prediction of noise generation due to frictional vibration of rubber products based on mechanical and frictional properties of rubber materials was yet impossible.

In this study, therefore, influence of the sliding surface roughness on the frictional vibration of bearing seals in dry condition was clarified and generation of frictional vibration was studied in relation to frictional and vibration characteristics of rubber materials.

2. Test Method

2. 1 Test Specimen

Rubber seals of 6 200 deep groove ball bearing as shown in **Fig. 1** were chosen for testing. Four different nitrile rubber materials (hereinafter referred to as NBR), as summarized in **Table 1**, were used.

2. 2 Effect of Sliding Surface Roughness

Sliding surface roughness (unevenness) was prepared by molding seals with each cavity treated with mat finish or sand blasting, respectively.

Surface roughness was measured with a laser microscope, and sliding test was performed with seals under dry condition.

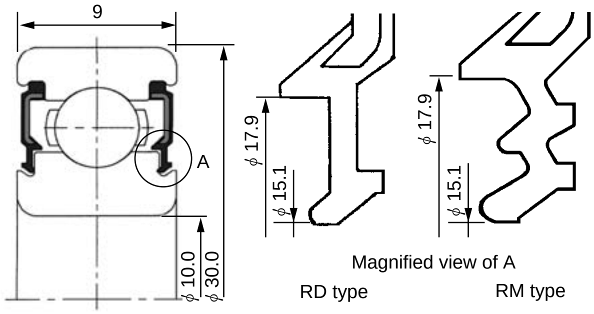

Fig. 1 Bearing seal

Table 1 Property of rubber material

	NBR1	NBR2	NBR3	NBR4
Hardness, HA	64	70	67	69
Tensile strength, MPa	12.7	13.9	20.2	18.7
Elongation, %	580	520	630	590
50% modulus, MPa	1.4	1.9	2.3	2.2
tan – δ (RT, 10 Hz)	0.18	0.19	0.19	0.20

2. 3 Distinction of Frictional Vibration

Schematic sliding part of the seal was shown in **Fig. 2** and sliding-motion model used was shown in **Fig. 3**.

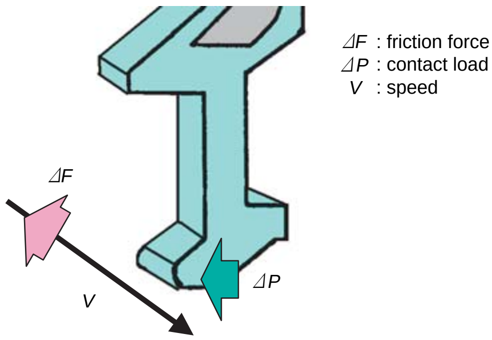

Fig. 2 Schematic of sliding part

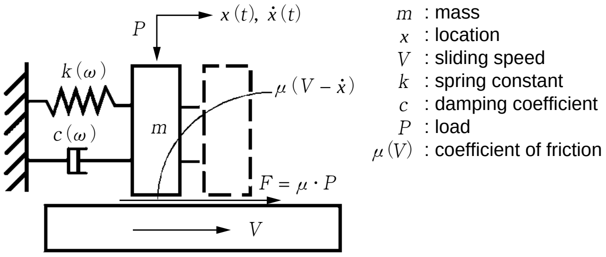

Fig. 3 Sliding motion model

The motion equation is expressed as follows:

$$m\frac{d^2x}{dt^2} = -k(x-x_0) - c\frac{d}{dt}(x-x_0) + \mu\left[V - \frac{d}{dt}(x-x_0)\right]\cdot P \qquad (1)$$

Then, by Taylor expansion,

$$\mu(V-\dot{x}) = \mu(V) - \dot{x}\frac{d}{dV}\mu(V) + \frac{1}{2}\dot{x}^2\frac{d^2}{dV^2}\mu(V)\cdots$$
$$\approx \mu(V) - \dot{x}\frac{d}{dV}\mu(V)$$

Supposing $X = x - \dfrac{\mu(V)\cdot P}{k}$, $\omega^2 = \dfrac{k}{m}$,

$$\ddot{X} + \frac{P}{m}\left\{\frac{c}{P} + \frac{d}{dV}\mu(V)\right\}\dot{X} + \omega^2 X = 0 \qquad (2)$$

Thus, stable or unstable frictional vibration can be distinguished as follows[2]:

$$\frac{c}{P} + \frac{d}{dV}\mu(V) > 0 \text{ -----Stable (no abnormal noise)}$$
$$\frac{c}{P} + \frac{d}{dV}\mu(V) \leq 0 \text{ ----- Unstable (abnormal noise generated)} \qquad (3)$$

In order to distinguish stable from unstable frictional vibration generated on the seals, the μ-V curve for each rubber material as well as the damping coefficient, c, and imposed load, P, were determined by FEM analysis and were compared with the test result.

The μ-V curve was obtained by the ring-on-disc friction tester as shown in **Fig. 4**. This test was conducted under the average contact pressure condition based on the load, P that was obtained from the seal sliding velocity and static FEM analysis.

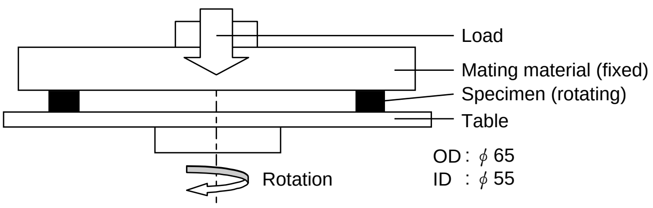

Fig. 4 Ring-on-disk type test method

As the damping coefficient, c, is expressed as $c = 4\pi f_0 m\zeta$ (f_0: natural frequency, m: mass, ζ : damping ratio), f_0 and m were calculated by the natural frequency analysis, and ζ by the transmissibility analysis.

The environment for the analysis was as follows:

Hardware: Octane/SI R10000/195MHz by SGI
Solver: ABAQUS ver.6.3-1 by ABAQUS, Inc.
Pre-post: I-DEAS Master Series ver. 9 by SDRC

3. Test Result

3. 1 Influence of Sliding Surface Roughness on Frictional Vibration

The test results on the conventional seals without any roughing on the sliding surface are shown in **Fig. 5** for frictional torque and **Fig. 6** for noise generating speed range. In both figures, the test plot and the speed range at which noise was generated are shaded dark. The result of FFT analysis when noise generates is shown in **Fig. 7**.

Frequency of the noise due to frictional vibration was around 11 kHz. The noise was generated in the speed range where the frictional torque markedly decreased. Furthermore, the noise generating speed range depended on the rubber material and interference. NBR1 rubber showed wider noise generating range and further wider with larger interference.

A measurement example of frictional torques with the NBR1 seals with rough sliding surface that showed wide noise generating range in the previous tests was shown in **Fig. 8**, wherein dark shaded plotting showed noise generation.

As seen in **Fig. 8**, seals with the sand blasted surface indicated lower friction torque and no noise generation. Although observations of two types of sliding surfaces were different as shown in **Fig. 9**, roughness values of them did not show any significant difference as seen in **Table 2**.

However, when these surfaces were compared by the power spectrum as shown in **Fig. 10**, significant difference was distinguished. Specifically, the sand blasted surface had larger short wave component and smaller long wave component compared with mat finish surface.

As a result, it was found that providing the sliding surface with fine and deep roughness was effective to prevent generation of noise due to frictional vibration.

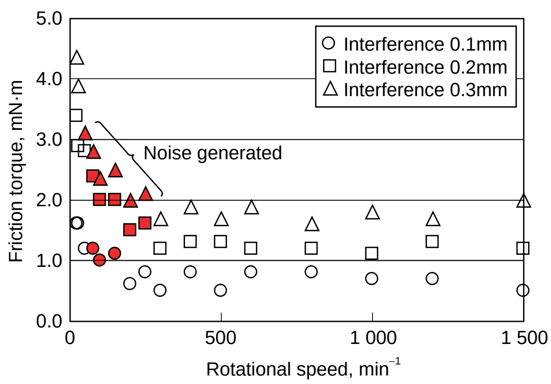

Fig. 5 Measurement example of friction torque (NBR1, type RD)

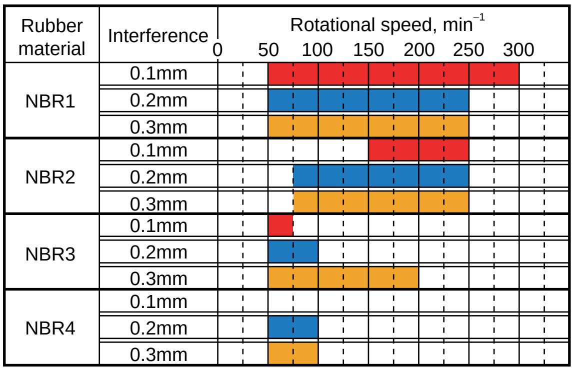

Fig. 6 Noise generating region caused by frictional vibration (type RD)

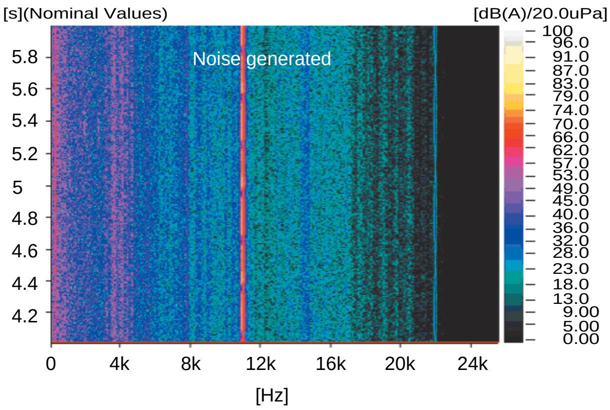

Fig. 7 FFT analysis result on noise

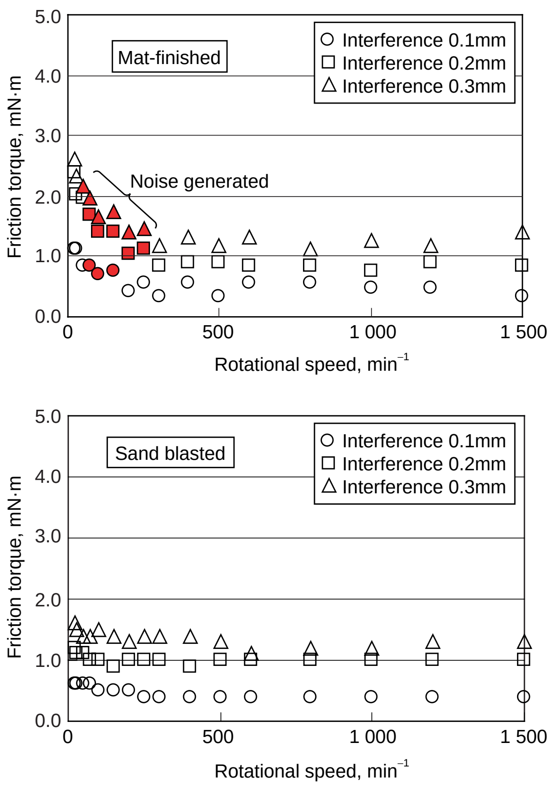

Fig. 8 Measurement example of friction torque (NBR1, type RD)

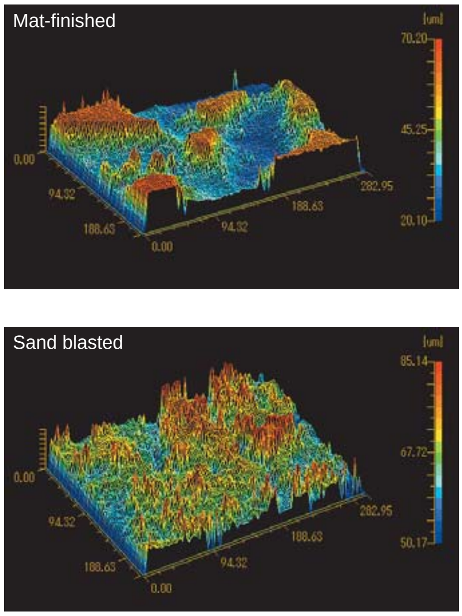

Fig. 9 Example of sliding surface observation (NBR1)

Table 2 Roughness of sliding surface

	Conventional (non-treated)	Mat-finished	Sand blasted
Ra	0.1~0.5	3~8	2~4
Ry	2~10	30~50	25~60
Rz	1~4	25~40	30~55
Sm	40~65	30~55	25~40

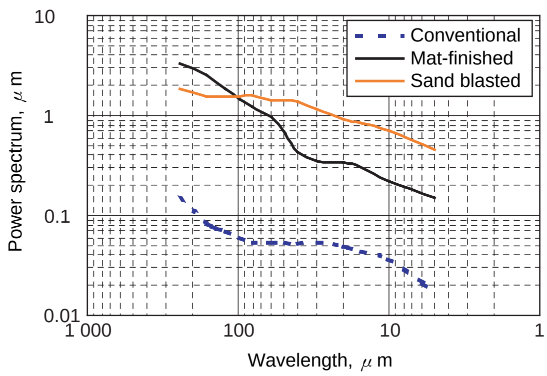

Fig. 10 Variation of power spectra on surface roughness

3. 2 Distinction of Frictional Vibration

To obtain damping ratio, ζ, vibration at the end of the seal lip when forced vibration was given on the metal ring was simulated by use of the model shown in **Fig. 11**. For each frequency, damping ratio ζ was determined by the relation among the vibration frequency, the ratio of amplitude at the metal ring and that at the lip end.

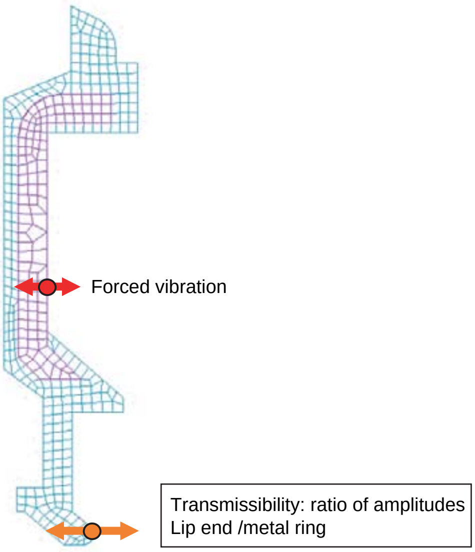

Fig. 11 FEM analysis model

In this analysis, Rayleigh's damping coefficient, α, (provided β = 0), obtained by comparison between vibration analysis and experimental results on a strip specimen (5 × 50 × 2) of each rubber material were used. Also, ζ was calculated by way of regression of transmissibility dependence on the frequency using the following expression:

$$\frac{\sqrt{1+\left(2\zeta\dfrac{f}{f_0}\right)^2}}{\sqrt{\left\{1-\left(\dfrac{f}{f_0}\right)^2\right\}^2+\left(2\zeta\dfrac{f}{f_0}\right)^2}} \qquad (4)$$

where, f : frequency

f_0: natural frequency

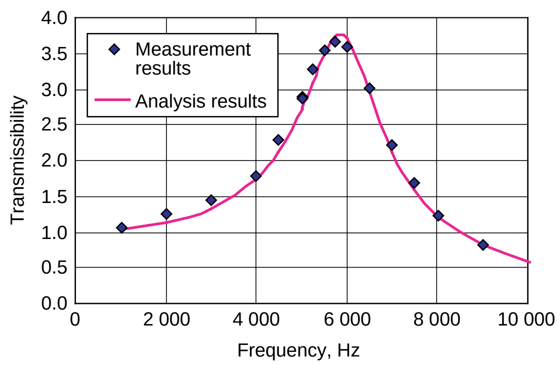

Fig. 12 Example of analysis on transmissibility (NBR1, type RD)

Analysis examples of transmissibility for each frequency were shown in **Fig. 12**.

In any seal type, as the transmissibility obtained by the analysis was almost consistent with that obtained from the experiment, the effectiveness of the analysis method was confirmed.

Based on the analysis of transmissibility, etc. the values, c and P, for RD type seal with each material was calculated and summarized in **Table 3**.

The damping ratio, c, is found to depend significantly on the rubber material, but not much on the seal design from comparison of c values for RM and RD type seals. There was no correlation between damping ratio and physical properties of rubber materials shown in **Table 1**.

Friction characteristics of NBR1 seal with roughened surface are shown in **Fig. 13**. Also, **Fig. 14** shows result of analysis to distinguish the frictional vibration.

In this study, the interference was 0.3mm.

The result for the seal with the mat finish surface was that the frictional vibration appeared in the same speed ranges as that for conventional non-treated seals.

On the other hand, the sand blasted surface seal did not show any frictional vibration even at the lowest speed, coinciding with the experimental finding.

In addition, **Fig. 15** shows comparison between the analysis of frictional vibration and the noise generation in the test on RD type seals. Though the test result did not agree with the analysis at low speed range because of difficulty in recognizing noise, these results were consistent on the high-speed side.

Therefore, the values obtained by the formula of discrimination based on the damping ratio, contact pressure and μ-V curve of each seal material was proved effective for estimation of frictional vibration range for seals.

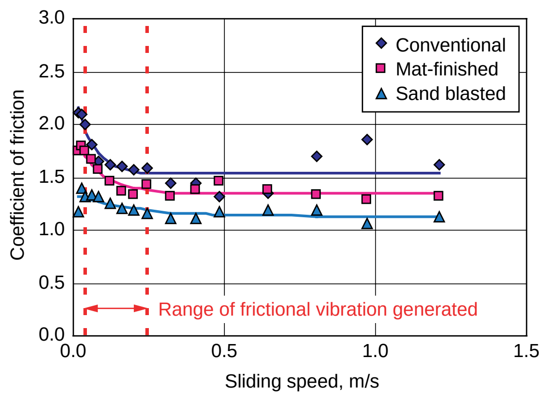

Fig. 13 Friction characteristics of nitrile rubber 1

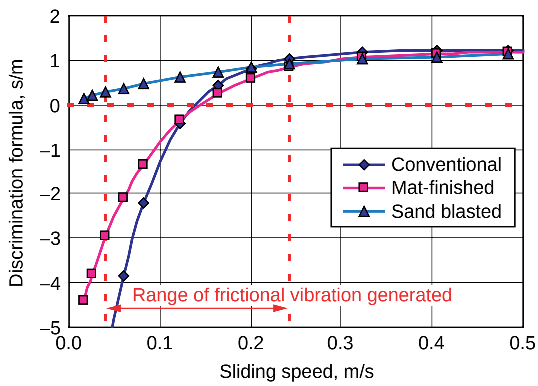

Fig. 14 Analysis on frictional vibration for each treatment

Table 3 Results of FEM calculation for c and P

Rubber material	Natural frequency analysis		Transmissibility analysis	Damping coefficient	Interference	Load	
	f_0	m	ζ	c		P	c/P
	Hz (= sec^{-1})	kg		kg/s = N·s/m	mm	N	s/m
RD type NBR1	5 950	1.33E–05	0.141	1.38E–01	0.1	4.05E–02	3.41
					0.2	7.78E–02	1.77
					0.3	1.14E–01	1.21
RD type NBR2	7 796	9.10E–06	0.189	1.67E–01	0.1	3.00E–02	5.56
					···	···	···
RD type NBR3	8 370	8.85E–06	0.235	2.17E–01	0.1	3.57E–02	6.08
					···	···	···
RD type NBR4	4 794	1.22E–05	0.103	7.47E–02	0.1	3.62E–02	2.06
					···	···	···
RM type NBR1	6 080	1.36E–05	0.135	1.35E–01	0.1	3.62E–02	3.74
					···	···	···

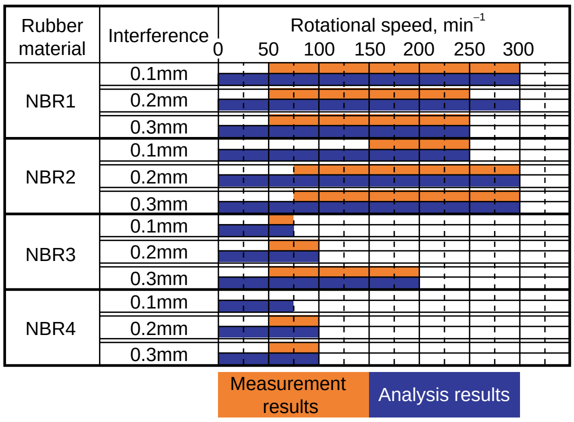

Fig. 15 Comparison of frictional vibration between analyzed and actual measurement

4. Conclusions

The effect of roughness of seal sliding surface on the frictional vibration of bearing seals was examined and the method to discriminate the condition that causes frictional vibration was established. The results are summarized as follows:

1) As a method to prevent noise due to frictional vibration, it is effective to roughen the seal sliding surface by applying sand blasting on the molding cavity.

2) It was also found that the magnitude of the effect of seal roughness on frictional vibration could be identified by power spectrum.
Also, increasing the short wave component and reducing the long wave component can minimize frictional vibration.

3) The results of discrimination analysis for frictional vibration based on viscosity ratio, contact load pressure obtained by FEM analysis, and μ -V curve of rubber material were well consistent with the test results.
This method was found effective to estimate frictional vibration range for seals.

References

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K. YAMAMOTO* D. OZAKI** T. NAKAGAWA*

* *Core Technology Research & Development Department, Research & Development Center*

** *Tennessee Koyo Steering Systems Company*