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Experimental Study on the Dynamic Response of Oil Seals

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Abstract: This paper deals with an experimental study on the dynamic response of an elastomeric oil seal when the interferences between shaft and lip as well as the dynamic eccentricities are present. The dynamic response of seal lip in oil seals was observed with the aid of an image processing apparatus. The temperature of the seal lip edge, friction torque and the dynamic sealing gap between shaft and lip are measured at different conditions of the initial interference and the shaft eccentricity. The data were simultaneously measured under dynamic conditions. Experimental results show that, as the shaft speed is increased, S/e_d has nearly reached a constant asymptotic value for a certain range of shaft speeds. The results indicate that the gap separation between shaft and lip is provided due to the shaft eccentricity because the seal lip cannot follow quickly the radial displacements with increasing shaft speed.

Key words : oil seals, dynamic response, frictional torque, shaft eccentricity

Introduction

Oil seals are widely used in machinery to seal rotating shafts at low oil pressure. Proper seal function depends upon the maintenance of the contact force arising from the interference between lip and shaft. Fig. 1 shows a typical oil seal as used in the tests. Variations in the operating parameters which include the interference, shaft dynamic runout and shaft speed will change the friction forces which will result in seal performance changes. Friction generates heat and torque that result in power losses and a reduction in service life.

In practice, dynamic excitation of the seal lip always occurs due to out-of-roundness of the shaft or oscillating shafts. The dynamic eccentricity may leak the lubricating oil because of viscoelastic behaviors. The deformed geometry of rubber lip is quickly unable to follow the radial displacement of the shaft as the shaft speed increases. The dynamic response of the seal lip to the shaft has been observed by Ishiwata [1] and Prati [2,3]. Ishiwata and Hirano have reported the followable limit of seal lip with the shaft eccentricity and interference. However they did not maintain the oil sump temperature constant during the test. Prati indicated that the gap clearance between shaft and seal lip may occur under certain conditions, but the film of lubricating oil is ignored in his experiment. Many researchers have proposed several lubricating mechanism of the oil seals [4-7]. Nakamura and Kawahara [8] analyzed the contact pattern of the rubber lip on a glass shaft and confirmed the existence of microasperity contacts.

This paper deals with an experimental study of the dynamic response of oil seals of both the shaft eccentricity and the shaft interference. The seal lip temperature and friction torque were measured with a thermocouple and transformer torque sensor, respectively. The gap separation between seal lip and shaft was detected using the image processing apparatus.

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Experiment

Tester

Fig. 2 shows the testing apparatus utilizing measurement system. A servo controller was used for various speed control. The total static misalignment of the test rig is 10 μm TIR or less between test seal housing and shaft. The temperature of the seal lip measured by Chromel-Alumel thermocouple (Omega K-type wire diameter 0.127 mm) which is inserted in the lip edge. The friction torque was measured with a torque

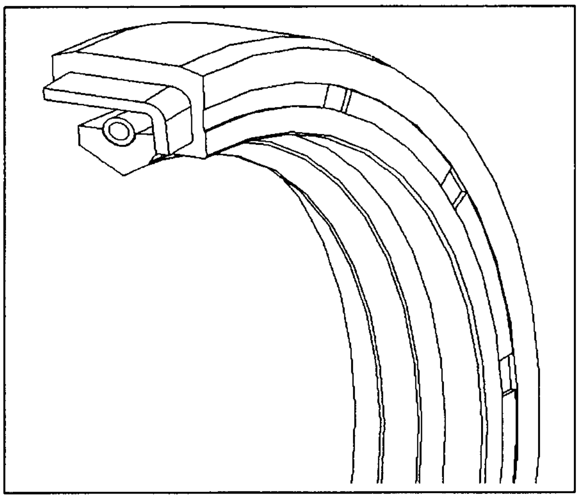

Fig. 1. Typical oil seal.

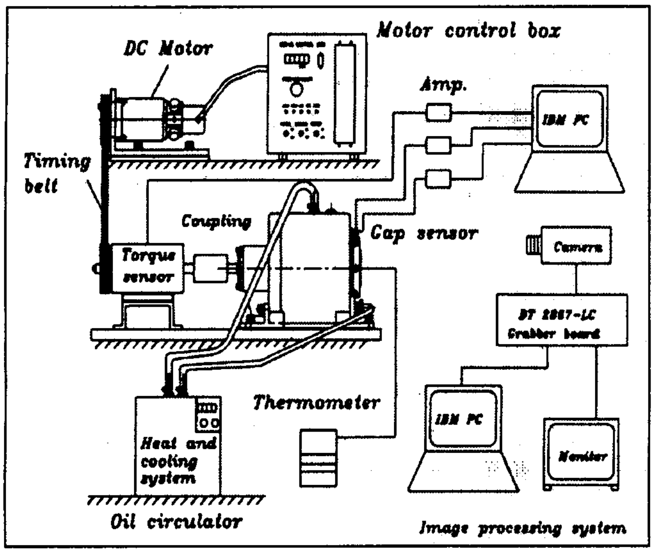

Fig. 2. Schematic representation of the testing device for an oil seal performance.

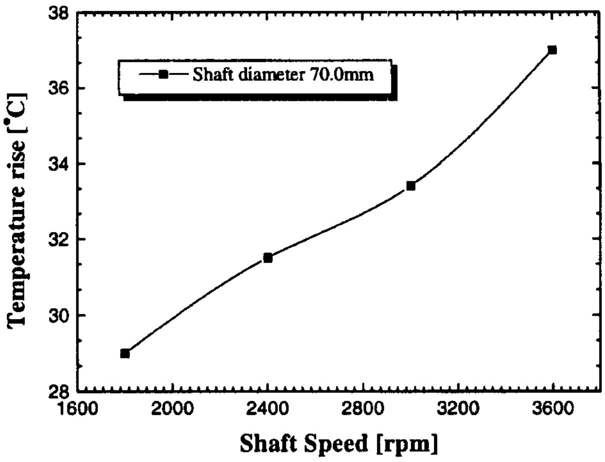

Fig. 3. Seal lip temperature rise as a function of the shaft speed.

sensor. For the actual measurement of the seal torque, the friction torque of the test rig itself was compensated without the test seal. The radial displacement of the shaft surface was measured by a non-contact gap sensor with fast response. And the gap separation between the shaft surface and lip edge was observed with an image processing system.

During the test runs the oil temperature is kept at 30℃ by heating and cooling system. Lubricant used is a base oil(viscosity 20.07cSt at 40℃, 4.10cSt at 100℃) and oil level is usually kept at the center of the rotating shaft.

Test specimen

Commercially available nitrile elastomer NBR (hardness 72 Shore A) and a 70×95×13 seal (nominal shaft diameter = 70 mm; external seal diameter = 95 mm; seal width = 13 mm; unloaded internal seal diameter = 67.6 mm) are used as oil seal samples. The spring constant of the garter spring is 76.3N/m, wire diameter is 0.2 mm, coil diameter is 3.0 mm. The alloy steel shaft is hardened to Rockwell C 32~38, finished to 0.8S (0.2a) and shaft roundness 5 m TIR or less. Various dynamic eccentricities (shifted center) ed are used for the test shaft, which are 0.0, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7 and 0.8 mm with the shaft diameter of 69.0, 69.5 and 70.0 mm, respectively.

Test Results and Discussion

Lip temperature

It is well known that excessive lip temperature is a prime cause of the seal failure. Lip temperature rise between lip and ambient fluid is a function of the viscosity of the sealed fluid and heat transfer from the contact zone.

Lip seals should not run without lubrication for any prolonged period of time. When operating condition is dry, lip temperature climbed to an excessively temperature rise of 141℃ after a 400 seconds run and went into such violent stick-slip friction that it destroyed itself within 200 seconds, at a constant shaft speed 1800 rpm and shaft diameter 70.0 mm. High temperature of the seal lip may quickly degrade the material properties of an elastomeric rubber. This may lead to become hard, brittle, and be unable to follow shaft surfaces. This is strongly related to the gap separation which is given by dynamic eccentricities.

The steady-state lip temperature was reached after a 900 seconds run at a constant shaft speed 1800 rpm. When the shaft diameter is 70.0 mm, seal lip temperature rise was from 29 to 37℃ with increasing the shaft speed from 1800 to 3600 rpm, as shown Fig. 3. The temperature profile of the figure is similar to that of Horve's experiments [9].

Friction torque

Steady-state friction torque was measured after a 900 seconds run as a function of the shaft speed. Friction torque of the seal increases as the shaft speed increases. Figs. 4 and 5 show the effect of shaft interference. At the speed of 3000 rpm, torque changes from 0.47 to 0.60 N · m as shaft diameter increases from 69.0 to 70.0 mm, and the power consumption increases by approximately 40.82 W.

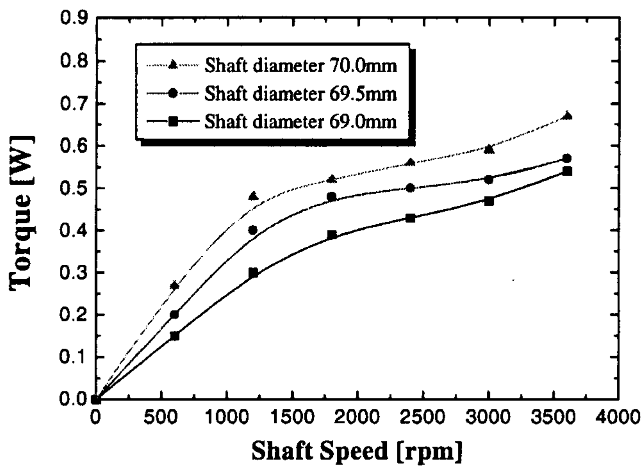

Fig. 4. Friction torque as a function of the shaft speed for various shaft interferences.

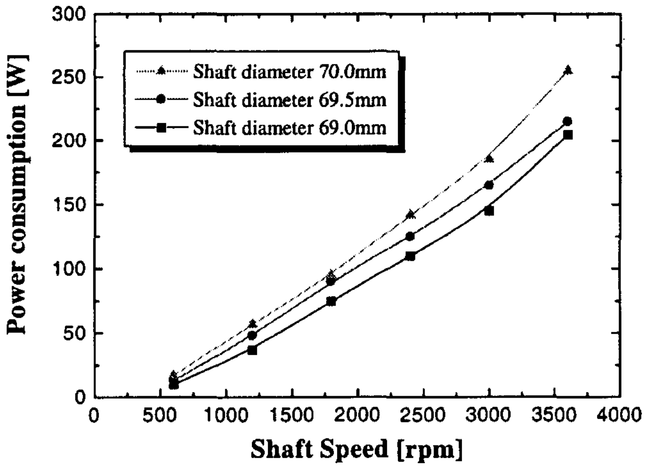

Fig. 5. Power consumption as a function of the shaft speed for various shaft interferences.

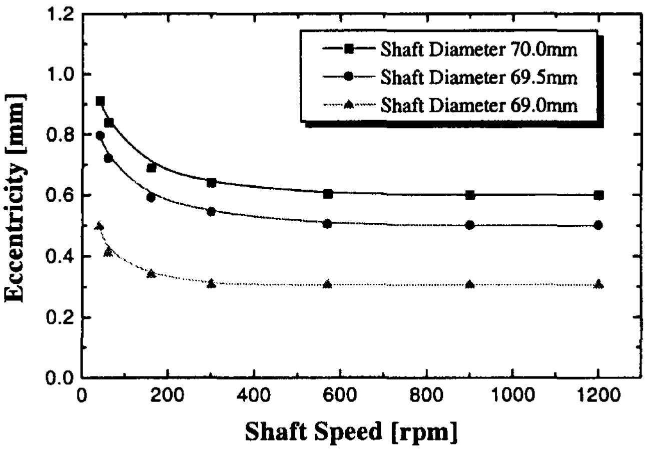

Fig. 6. Dynamic eccentricity as a function of the shaft speed for lip-shaft loss contact conditions.

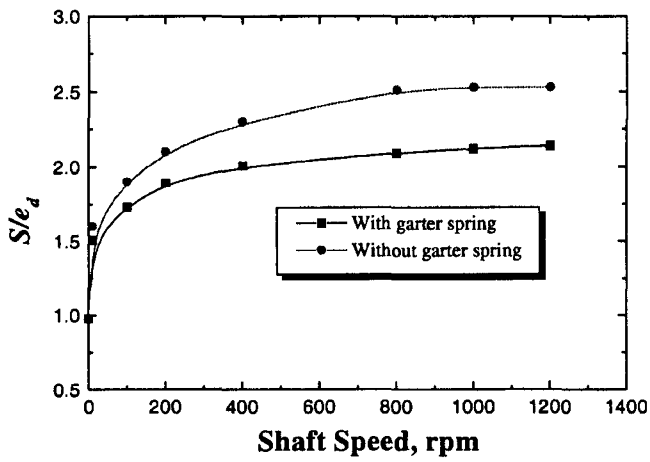

Fig. 7. S/e_d as a function of the shaft speed for lip-shaft gap separation contact conditions.

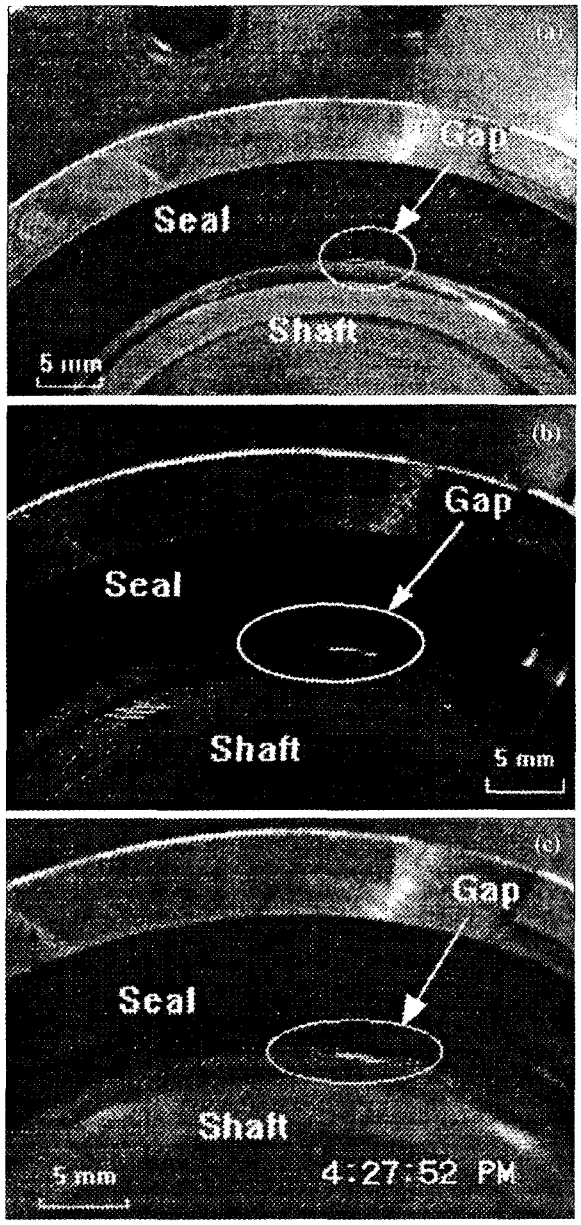

Fig. 8. Gap separation of contact between lip and shaft. (a) Eccentricity = 0.6 mm; speed = 570 rpm; shaft diameter = 70.0 mm. (b) Eccentricity = 0.5 mm; speed = 593 rpm; shaft diameter = 69.5 mm. (c) Eccentricity = 0.3 mm; speed = 520 rpm; shaft diameter = 69.0 mm

Lip-shaft contact behavior

The lip-shaft contact conditions were determined for various values of the shaft interference, dynamic eccentricity and shaft speed. The loss of contact is evidenced by the light passing through the narrow gap between seal lip and shaft surface. This is demonstrated by visually observing light passing under the seal lip during rotation. Experimental results suggest that the gap separation of contact was highly dependent on the shaft interference and the level of dynamic eccentricity. The shaft diameters of 70.0, 69.5, 69.0 mm produce the gap separation of contact for shaft eccentricities 0.6, 0.5, 0.3 mm, respectively, as given in Fig. 6.

Shaft interference δ is defined as a difference in outer diameter d_1 of the shaft and diameter d_2 of the seal, i.e. $\delta = d_1 - d_2$. Static interference S is given by $S = e_s + \delta/2$. It is assumed to be e_s = zero, $S = \delta/2$.

Fig. 7 shows that as the shaft speed increases, it can be observed that the contact limit value S/e_d of the gap separation approaches an asymptotic value of 2.0 at relatively low shaft speed 600 rpm with garter spring. But S/e_d is about 2.7 without garter spring. This result indicates that the gap separation of the seal with the increased stiffness of seal materials may be occurred at low speed.

As an example, Fig. 8 illustrates that the gap separation of the seal between lip and shaft is made evident by the light that crosses the gap.

Conclusions

An experimental study of the dynamic response of oil seals for rotating shafts has been presented for both the shaft interference and the dynamic eccentricities.

The steady-state seal lip temperature was reached after a 900 seconds run. When the shaft speed rose to 3600 rpm, the lip temperature also rose to 37°C. The friction torque of the seal was increased with increasing the seal-shaft interference and the shaft speed. The gap separation of contact was highly dependent on the shaft interference and on the level of dynamic eccentricity. The experimental results show that as the shaft speed is increased, the asymptotic value at relatively low speed for the stiff materials.

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