KSTLE International Journal, Vol. 1, No. 1, June 2000, pp. 43-47

Numerical Study on the Dynamic Response in Elastomeric Oil Seals

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Abstract : Oil seals will experience a small amplitude dynamic excitation due to the shaft eccentricity as well as out-of-roundness of the shaft. The direct integration method is selected to analyze the time domain response of the seal lip-shaft contact. The physical properties of rubber seal materials are experimentally analyzed. Effects of both frequency and temperature on the material stiffness behavior are investigated for the linear viscoelastic materials of the seal. Using the nonlinear transient model, a finite element analysis of the lip-shaft contact behaviors under dynamic conditions is presented as a function of the shaft eccentricity, the shaft interference and the garter spring stiffness. The FEM results based on the experimental data indicate that the increased rotating speed may produce the separation conditions. These results will be very useful in predicting the leakage of oil seals under dynamic conditions.

Key words : oil seals, FEM, dynamic response, elastomeric rubber

Introduction

An elastomeric oil seal is a key element used for sealing non-pressurized fluids throughout industry to contain lubricants and exclude contaminants. Oil seals for rotating shafts usually consist of a metal case, a flexible rubber seal, a sealing lip loaded by a garter spring. To be effective, an elastomeric oil seal must maintain an adequate contact force between the rotating shaft and the seal. Though the contact force of elastomeric oil seals prevents the leakage of the lubricating oil, the increased friction force can degrade the sealing performance of the rotating dynamics of the shaft. Fig. 1 shows typical oil seal for a pump, compressor, etc.

In practice, a dynamic excitation of the seal lip always occurs due to out-of-roundness of the shaft or motions of the shaft center. This misalignment is called the dynamic eccentricity. During shaft rotation, the dynamic eccentricity is able to produce a gap between lip and shaft because of viscoelastic behavior; thus, this produces a gap through which the sealed fluid leaks. Gawlinski [1] suggested that the radial lip sealing mechanism depends upon an axial scrubbing action that results from shaft dynamic eccentricity. Gawlinski [2] also studied on seal-shaft contact problems with dry conditions using the finite element method treated to linear viscoelastic model. Ishiwata and Hirano [3] and Prati [4] experimentally observed the viscoelastic behaviors of the web-lip-spring system for the seal, and then a three-parameter solid viscoelastic model was employed. But this model is available only for given seal specimen. Concerning the viscoelastic material behavior of rubber it is important to make a distinction between the rubber stiffness under static load and dynamic load. The contact force of the preload of the seal due to interference between seal and shaft was achieved by using a FEM under static conditions [5].

In the present work, employing a FEM cylindrical coordinate solid model, seal response on a small-amplitude harmonic displacement input was obtained in time domain, superposed on a nonlinear preload of the seal which analyzed in static condition with garter spring. The numerical study investigates a limit value under certain operating conditions of oil seals. First, the physical properties of rubber seal materials are experimentally analyzed, and are used in FEM analysis. Next, the dynamic response of oil seals is parametrically studied including shaft interference, dynamic eccentricity, the spring rate of the garter spring and physical properties of rubber materials.

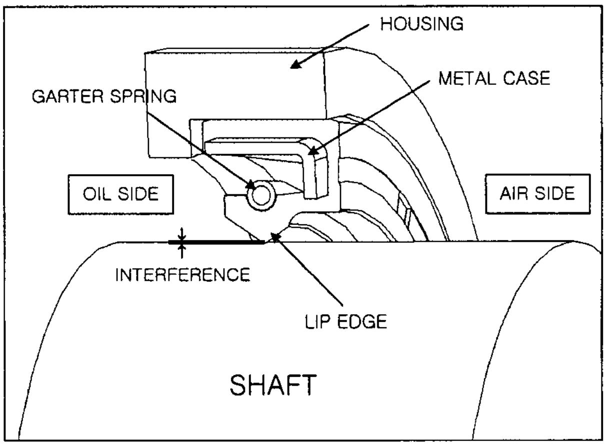

Fig. 1. Typical oil seals.

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Table 1. Test conditions for temperature-frequency dependence.

Frequency [Hz]	Temp. [°C]	Specimen [mm]	Dynamic strain [%]
0.01~16.7	-50~120	- width : 11.1 - Thickness : 2.33 - Length : 48.1	0.2

Material Property of a Nitrile Rubber

The dynamic mechanical properties of rubber depend on a number of environmental factors, such as temperature, frequency, dynamic load, and static preload. The most important factors are temperature and frequency. In this paper, it is assumed that for small amplitude oscillating displacements the influence of the amplitude of the load on the material stiffness can be neglected, then the material behaves linearly thermo-viscoelastic. The linear dynamic stiffness of rubber is represented as a function of time-temperature superposition principle [6]. By this principle, measurements of the mechanical properties as a function of frequency made at different temperatures can be collapsed on one master graph, if the appropriate temperature shift factor is used. In this experiment, the dynamic stiffness is obtained by temperature-frequency torsional sweep test using the RMS (Rheometrics Mechanical Spectrometer) under test conditions summarized in Table 1.

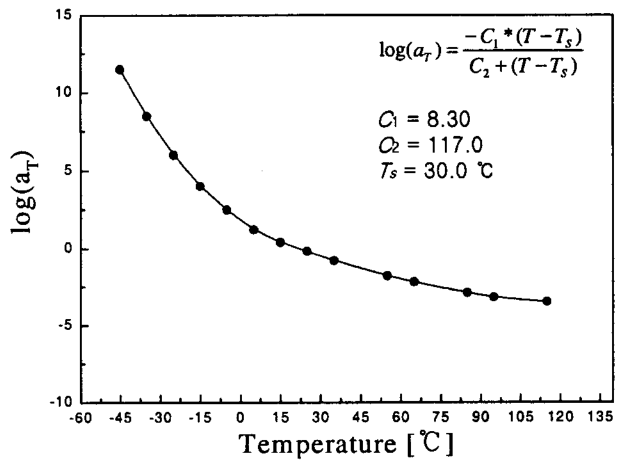

Fig. 2. Temperature shift factor $\log(\alpha_T)$ plotted against temperature.

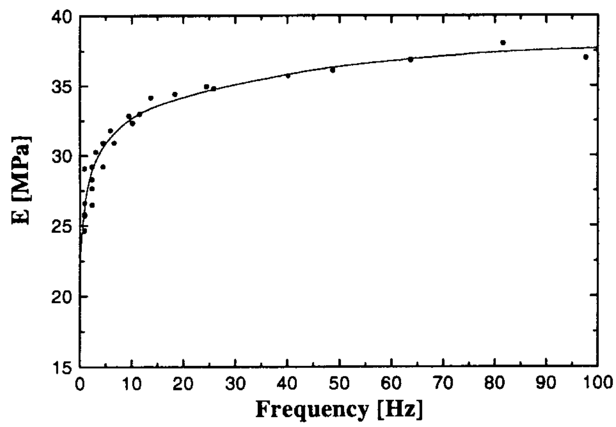

Fig. 3. Variation of the elastic modulus with frequency, 0.01~100Hz.

Temperature shift factor of $\log(\alpha_T)$ for the seal rubber, based on 30°C as reference temperature, plotted against temperature is presented in Fig. 2.

The dynamic stiffness of rubber at $T = 30°C$ versus frequency as given in Fig. 3 was used. Variation of the dynamic stiffness is from 17.0 MPa to 37.5 MPa for 0.01~100 Hz. This will be accomplished by a seal response in time domain, based on a steady-state dynamic FE analysis.

FE Analysis

A conventional oil seal was modeled using the MSC/NASTRAN FEM software [7]. The seal-shaft contact problem was assumed to be axisymmetric and isothermal, $T = 30°C$. The FEM model consists of 266 8-node solid elements for cylindrical coordinate with 0.2 degree. The nodes of the solid elements restrain the circumferential displacement, considering only the radial displacement. The symmetric nodes are also considered having the radial displacement only. The garter spring was modeled by one spring element connected to the ground at one end, and a spring element was connected to seven rigid beam elements at the other end as indicated in Fig. 4. The beam elements restrain the displacements but not the rotations and provide a pinned rigid link between the nodes on the seal and a node on the end of the spring element. Using the gap elements the seal-shaft contact forces act on when the seal lip lifts up, and are free when the shaft comes downward. Seal dynamic equation is expressed as following

$$M\ddot{x} + C\dot{x} + Kx = P(t) \tag{1}$$

Seal deformation can be divided into static deformation components xs due to shaft interference and a sinusoidal deformation components xd due to dynamic eccentricity, which is superposed on xs. The total radial displacement x(t) is given by

$$x(t) = x_s + x_d(t) \tag{2}$$

Fig. 5 presents the circular profile of the lip-shaft contact; "O" denotes the theoretical center of the oil seal before the shaft is inserted, and "Q", the geometrical center of the shaft rotating around point "O" with angular speed ω. The radial displacement $x_d(t)$ of the lip points is in contact with the shaft which is evaluate with respect to the circumference of center "O", and dynamic motion is

$$R + x_d(t) = e_d \cos \omega t + (R^2 - e_d^2 \sin^2 \omega t)^{1/2} \tag{3}$$

If dynamic eccentricity $e_d << R$, $x_d(t)$ is given by

$$x_d(t) = e_d \cos \omega t \tag{4}$$

It is therefore that seal displacement $x(t)$ is expressed as following.

$$x(t) = x_s + e_d \cos \omega t \tag{5}$$
$$x_s = S = e_s + \delta/2 \tag{6}$$

If static eccentricity $e_s = 0$, S is given by

$$S = \delta/2 \tag{7}$$

It is assumed that a node of the seal lip edge will be last to remain in contact with the seal. Therefore an output displacement $x(t)$ is the seal lip edge node. NASTRAN provides an input displacement $x(t)$ by large mass method. Employing a direct integration method with numerical integration Newmark-Beta obtained an output radial displacement $x(t)$.

FEM model also takes into account both the inertia of the lumped garter spring mass and the rubber mass. The physical properties used in the dynamic FEM model is given by

- Garter spring
  - Mass [kg] $m = 5.54 \times 10^{-3}$
  - Initial length [m] $l_0 = 232 \times 10^{-3}$
  - Initial spring force [mN] $F_0 = 37.023$
  - Spring constant for 0.2 deg. [N/m] $k_s = 3.31$
- Seal material (NBR)
  - Specific mass [kg/m^3] $\rho = 1.22 \times 10^3$
- Inner diameter of the oil seal
  - With garter spring [m] $d_2 = 67.6 \times 10^{-3}$
  - Without garter spring [m] $d_2 = 68.3 \times 10^{-3}$
- Nominal shaft diameter [m] $d_1 = 70.0 \times 10^{-3}$
- Shaft interference [m] $S = 1.2 \times 10^{-3}$

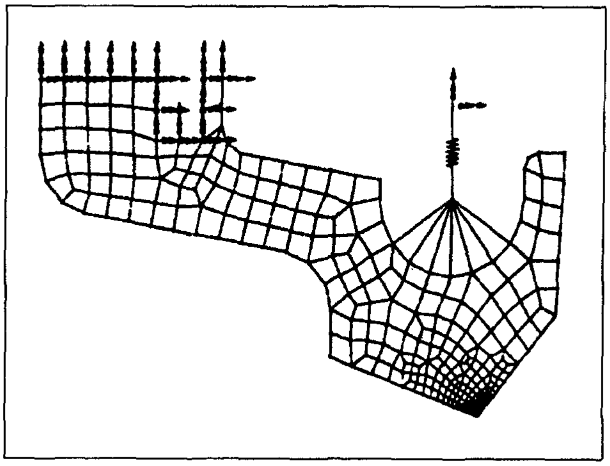

Fig. 4. FEM solid model of the dynamic seal responses for cylindrical coordinate with 0.2 degree.

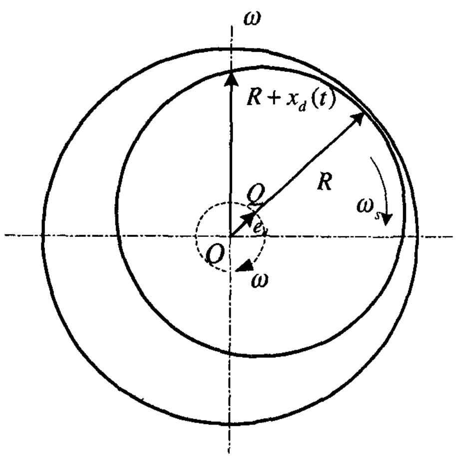

Fig. 5. Displacement $x(t)$ of the seal lip in contact with the rotating shaft.

Discussions

Effects of the shaft interference

The calculated and measured values of the radial direction contact force for a lip seal for different shaft diameters at a constant temperature $T = 30°C$ after $t = 12$ hrs are given in Fig. 6. The contact force was measured on a split-shaft measuring device [5]. FEM program NASTRAN is used here for initial force in calculating the steady-state seal response.

The seal behaves under the following operating conditions: $e_d = 0.3$ mm, $n = 600$ rpm, 1200 rpm, $S(\delta/2) = 0.7$ mm, 0.95 mm, 1.2 mm. The lip-shaft loss of contact produced at shaft interference 0.7 mm and the gap increased with increasing the shaft speed, see Fig. 7.

Fig. 8 shows the radial displacement with increasing the shaft speed. In case of $S = 1.2$ mm and $e_d = 0.6$ mm, lip-shaft loss of contact did not produce even at high angular velocities. Fig. 9 shows that loss of contact appeared at dynamic eccentricity $e_d = 0.6$ mm. The lip-shaft loss of contact conditions is expressed by S/e_d as a function of the shaft speed. Lip-shaft loss of contact S/e_d is approaching a certain value as the shaft speed increases. The asymptotic values of S/e_d approaches already at 600 rpm as shown in Fig. 10.

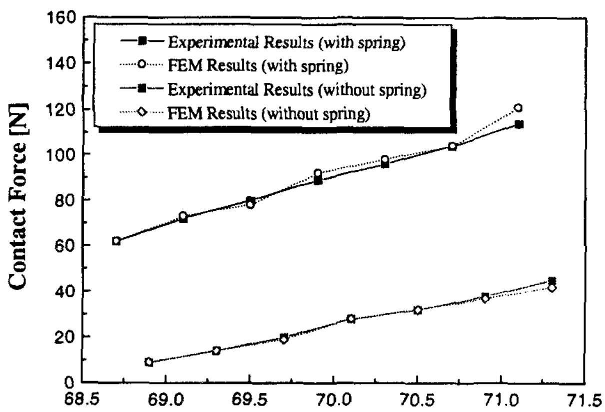

Fig. 6. Contact force, calculated with the FEM and measured with a split-shaft device, for a radial lip seal, with and without a garter spring.

Effects of the garter spring force

Radial lip-shaft loss of contact conditions is strongly influenced by garter spring's initial force. Fig. 11 shows that S/e_d decreases with increasing the initial force of the garter spring. According to the FE analysis, the garter spring constant

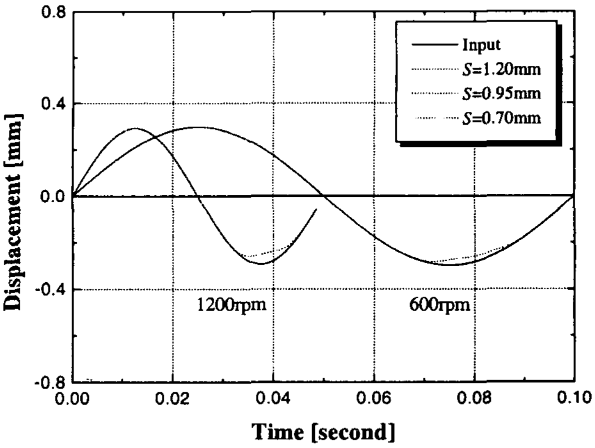

Fig. 7. Seal lip response on the shaft interference level.

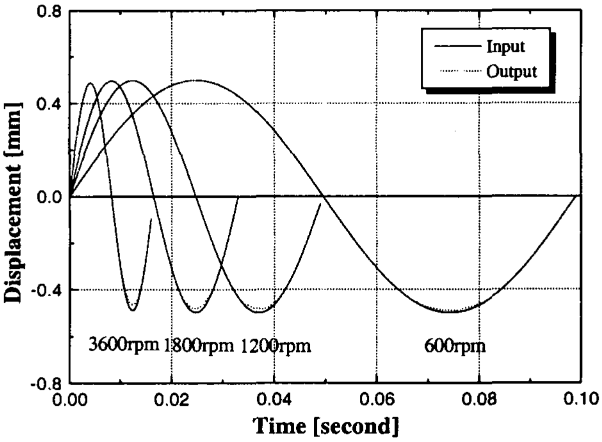

Fig. 8. Seal lip response on various shaft speed.

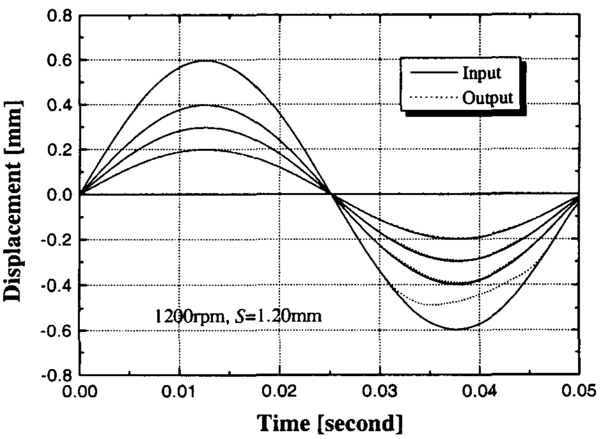

Fig. 9. Seal lip response on various dynamic eccentricity.

(in case of k_s = 0.8 N/m, 1.6 N/m, 3.3 N/m) yields only a minor effect on loss of contact, because seal displacement is very small compare to the seal diameter. Therefore a good understanding of such effects, both initial force and spring constant, on the seal design is necessary.

Effect of the material stiffness

Fig. 12 shows that the effect of the material dynamic stiffness.

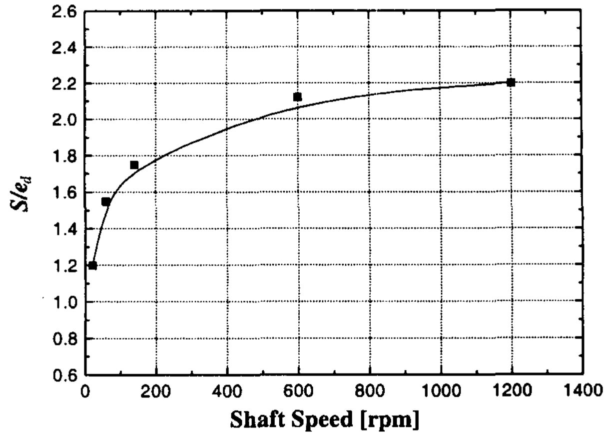

Fig. 10. S/e_d as a function of the shaft speed for lip-shaft loss contact conditions.

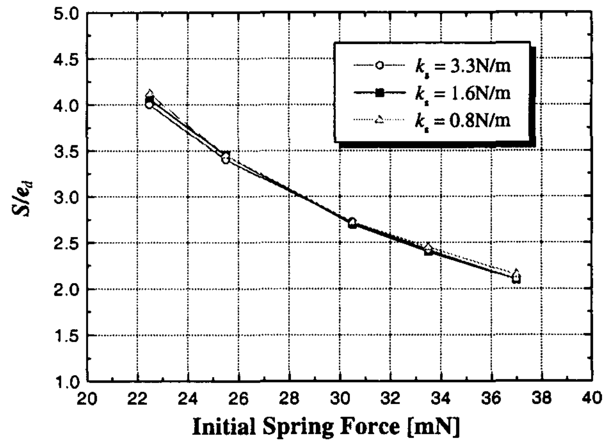

Fig. 11. Effect of garter spring force for lip-shaft loss of contact.

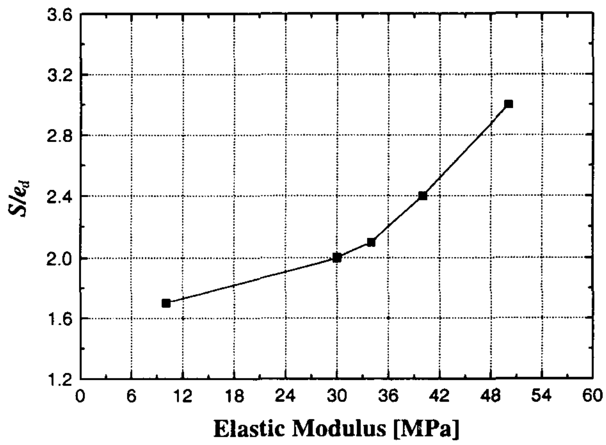

Fig. 12. Effect of rubber material stiffness for lip-shaft loss of contact.

Lip-shaft loss of contact highly depends on rubber material stiffness. S/e_d is increased from 1.7 to 3.0 as an elastic modulus is 10 MPa to 30 MPa. This results represent that the higher elastic modulus value, the more resistant the material is to being stretched. Thus highly flexible seal designs may be required if excessive dynamic eccentricity is expected.

Conclusions

Employing a direct integration method, harmonic response characteristics of oil seals are modeled and simulated for small amplitude dynamics as a function of the shaft interference and shaft eccentricity as well as a garter spring and material stiffness. Through the FEM analysis, the shaft eccentricity clearly produces the gap between shaft and lip which is unable to follow the radial displacements of shaft as the shaft speed increases. Results of the FEM analysis are summarized as follows:

(1) Lip-shaft loss of contact strongly depends on both the shaft interference and dynamic eccentricity. The gap was produced at certain values of dynamic eccentricity, but this gap was not produced even at high shaft speeds.

(2) Lip-shaft loss of contact S/e_d is approaching a certain value as shaft speed increases. The asymptotic values of S/e_d are approached relatively at low speed.

(3) Dynamic response of the seal is strongly influenced by garter spring's initial force, and spring constant of the garter spring only yields a minor effect on loss of contact.

(4) Also, seal dynamic behavior highly depends on rubber material stiffness. Thus under the dynamic conditions, it is not allowed that assuming a static stiffness of the rubber material, because a dynamic stiffness can be orders of magnitude higher. If excessive dynamic eccentricity is expected, highly flexible seal designs may be required.

Nomenclatures

$\boldsymbol{C}$ = Damping matrix
C_1, C_2 = WLF parameter
e_d = Dynamic eccentricity
e_s = Static eccentricity
$\boldsymbol{K}$ = Stiffness matrix
k_s = Spring constant of the garter spring
$\boldsymbol{M}$ = Mass matrix
$P(t)$ = Contact force in time domain
S = Static deformation of the lip (=$\delta/2$)
T_s = Reference temperature
x_s = Static deformation
$x(t)$ = Small-amplitude harmonic displacement
δ = Interference (= d_1 - d_2)
ω = Angular frequency
ω_s = Shaft angular frequency

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