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Upstream Pumping of Radial Lip Seals by Tangentially Deforming, Rough Seal Surfaces

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This paper aims at a theoretical explanation of the following two experimental observations of radial lip seals: fluid film formation and upstream pumping action. The origins of these observations are still poorly understood. A hydrodynamic analysis is presented for the fully flooded contact zone of a smooth shaft surface running against a rough seal surface. Four roughness structures are considered: one-dimensional sinusoidal and two-dimensional sinusoidal seal roughness structures both without and with a tangential seal surface displacement. The tangential seal surface displacement is chosen asymmetric with the axial position in the contact. It follows that the tangential deformation of the one- and two-dimensional seal roughness structures used in this paper can explain the aforementioned experimental observations and simulate the observed order of magnitude of load-carrying capacity, pump rate, and friction torque found in the literature.

1 Introduction

Radial lip seals are used to prevent oil leakage from a reservoir to the environment and entrance of contaminants into this reservoir. Figure 1 shows the assembly of a shaft, housing and radial lip seal. The seal consists of a metal case with a molded elastomeric lip attached to it. Generally a metal garter spring is added. Radial lip seals have been applied in large numbers for more than 50 years. In spite of this an experimentally verified basic theory explaining the operating principle is still lacking. In this quest for the explanation of the lubrication, friction, and sealing behavior of radial lip seals, two experimental observations are considered important:

(1) Radial lip seals generate a thin, load-carrying fluid film in the contact zone when running against a rotating shaft. The existence of this oil film has remained undisputed over the years (Jagger, 1957b; Schouten, 1978; Poll and Gabelli, 1992; Poll et al., 1992). This fluid film formation is remarkable since conditions to establish full film lubrication are absent at first sight.

(2) Radial lip seals show an upstream pumping action (Johnston, 1973; Kawahara and Hirabayashi, 1979; Kawahara et al., 1980; Kammüller, 1986; Horve, 1987; Horve, 1991). This action denotes the ability of a fully flooded radial lip seal to generate a net axial fluid flow from the air side to the oil side of the seal against a static pressure head.

The physical process underlying these observations is poorly understood at this moment. A basic model of a radial lip seal should provide an explanation for both experimental observations.

An explanation of the fluid film formation should concentrate on the exploration of the presence of imperfections in the nominal parallel contact zone of seal and shaft. These imperfections introduce deviations from nominal parallelism which enable fluid film formation by entrainment action.

An example of a macroscopic deviation from nominal parallelism to explain the fluid film formation was proposed by Blok (1957, 1965). He studied the effect of a nonaxisymmetrical

contact pressure distribution caused by e.g., a static eccentricity between the seal and the shaft. Another macroscopic deviation from nominal parallelism was introduced by Van Leeuwen and Stakenborg (1990, 1991) who studied fluid film formation due to a delayed seal surface rebound caused by the viscoelastic and inertial seal behavior when excited periodically by shaft runout.

Fluid film formation due to microscopic imperfections has also been suggested. Microhydrodynamic lubrication of a rough shaft and a smooth seal was reported by Hirano and Ishiwata (1965). This smooth seal assumption was criticized, however, by Halliday (1965), Jagger and Walker (1966), and Johnston (1978). The latter claimed that a rough seal and a smooth shaft assumption is far more realistic when comparing the roughness heights of a run-in seal and shaft surface. Jagger and Walker (1966) continued along these lines and analyzed the elastohydrodynamic lubrication of an individual hemispherical seal asperity. The coefficient of friction predicted theoretically could be brought into agreement with the experimental observations. This approach was extended by Gabelli (1989) and Gabelli and Poll (1992) to account for a random seal asperity height.

Explaining the sealing action of radial lip seals while maintaining a liquid lubricated contact zone represents the major challenge. Early work of Jagger (1957a) and Jagger and Wallace (1973) claims that the surface tension of the lubricant of the oil-air interface which exists at the air side of a normally installed lip seal is responsible for the sealing action. However, the upstream pumping action cannot be explained. Later it was concluded by Ott (1983) that Taylor vortices develop at the oil side provided that a sufficient amount of oil is available at the reservoir. It should explain the sealing behavior by lowering the static pressure head and by generating a reservoir directed velocity flow field. More recent experimental and theoretical work concentrates on the explanation of the sealing action by upstream pumping. The origin of this pumping action is not well understood and has not been firmly established yet.

Seal practice suggests that the following facts are considered important in achieving a properly operating radial lip seal:

(1) Lip seals where the macroscopic axial tip geometry of the seal is chosen such that the oil side angle α is larger than the air side angle β , see Fig. 1, tend to seal (Hirano et al., 1961; Kammüller, 1986).

(2) Lip seals with smooth seal surfaces tend to have difficulty in dynamic sealing whereas rough seal surfaces tend to

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Fig. 1 Assembly of a shaft, housing and radial lip seal

have greater reliability in dynamic sealing, caused by an increased pump rate (Nakamura and Kawahara, 1984; Kammüller, 1986; Nakamura, 1987; Horve, 1991).

(3) In addition, seal research has revealed that lip seal surfaces deform tangentially due to the friction forces exerted on the seal surface. This displacement is distributed asymmetrically with the axial position in the contact zone (Kawahara et al., 1980; Kammüller, 1986).

Note that it is not fully understood why these facts are important. They form important clues to seal analysts in the development of a basic theory.

Seal roughness studies show that the following structures were observed on run-in seal surfaces:

(1) Axially aligned seal surface roughness grooves (Ott, 1983; Kammüller, 1986).

(2) Hemispherical seal surface asperities (Jagger and Walker, 1966; Nakamura and Kawahara, 1984; Nakamura, 1987; Müller, 1987).

Possible concepts to explain the pumping action where seal roughness is crucial were proposed but hardly rigorously analyzed from a hydrodynamic point of view. So their potential and value to explain radial lip seal behavior have not been established.

Kammüller (1986) proposed the so-called visco-seal concept, using the axially aligned roughness structure in combination

with a tangential seal surface displacement that is asymmetrical with the axial position, resulting in a herringbone like (Muijderman, 1966) seal surface. This concept has been subjected to a rigorous hydrodynamic analysis by Salant (1992a) and to an elastohydrodynamic analysis by Salant and Flaherty (1994). Müller (1987) proposed his so-called cascade pumping side flow concept. In this case, the pumping action is assumed to be caused by the flow interaction of hemispherical seal asperities in combination with the asymmetrical macroscopic axial geometry of the tip of the seal.

Sponagel et al. (1987) proposed that an individual hemispherical seal asperity in the contact zone becomes oriented when the seal surface is subjected to a tangential seal surface displacement that is asymmetrical with the axial position. Each asperity acts like an individual microscopic pump.

This paper presents a hydrodynamic analysis of a fully flooded contact zone of a running, smooth shaft and the rough tip of a radial lip seal. Four roughness structures are analyzed:

(1) A one-dimensional sinusoidal (axially aligned) seal roughness structure and

(2) A one-dimensional sinusoidal (axially aligned) seal roughness structure including a tangential seal surface displacement,

(3) A two-dimensional sinusoidal seal roughness structure and

(4) A two-dimensional sinusoidal seal roughness structure including a tangential seal surface displacement.

The objective is to test the capabilities of these four cases to predict radial lip seal behavior by comparing the computed pump rate, load-carrying capacity, and shaft friction torque with measurements found in the literature.

2 Theory and Analysis

This section presents the mathematical model used to analyze the four cases mentioned previously. For this purpose the fluid flow behavior is studied by performing a hydrodynamic analysis

Nomenclature

b = contact width, m	p = absolute pressure, Pa	$u(y)$ = tangential seal surface displacement, m
$F(X, Y)$ = cavitation index, Eq. (4)	p_c = cavitation pressure (absolute value), Pa	U = shaft surface speed, m/s
G = amplitude of the macroscopic axial geometry of the oil film, m	p_a = atmospheric pressure (absolute value), Pa	$U(Y)$ = dimensionless tangential seal surface displacement, $U = u/\lambda_x$
$h(x, y)$ = film thickness, m	p_{av} = average hydrodynamic pressure (absolute value), Pa	x = tangential coordinate, m
h_0 = film thickness constant, m	p_s = reference pressure (absolute value), Pa	X = dimensionless x -coordinate, $X = x/\lambda_x$
$h_a(y)$ = macroscopic axial geometry of the oil film, m	P_1 = dimensionless number, $P_1 = (\lambda_x/b)^2$	y = axial coordinate, m
$h_r(x, y)$ = seal surface roughness geometry, m	P_2 = dimensionless number, $P_2 = 6\eta U \lambda_x / ((p_s - p_c) G^2)$	Y = dimensionless y -coordinate, $Y = y/b$
$H(X, Y)$ = dimensionless film thickness, $H = h/G$	P_3 = dimensionless number, $P_3 = h_0/G$	α = oil side tip angle
$H_a(Y)$ = dimensionless macroscopic axial geometry of the oil film, h_a/G	P_4 = dimensionless number, $P_4 = t/b$	β = air side tip angle
$H_r(X, Y)$ = dimensionless seal surface roughness geometry, h_r/G	P_5 = dimensionless number, $P_5 = L/2G$	λ_x = seal roughness wavelength in x -direction, m
K = maximum tangential seal surface displacement, m	P_6 = dimensionless number, $P_6 = M/4G$	λ_y = seal roughness wavelength in y -direction, m
$\frac{1}{2}L$ = amplitude of one-dimensional seal roughness geometry, m	P_7 = dimensionless number, $P_7 = b/\lambda_y$	$\phi(X, Y)$ = dependent variable, Eqs. (2) and (3)
$\frac{1}{2}M$ = amplitude of two-dimensional seal roughness geometry, m	P_8 = dimensionless number, $P_8 = K/\lambda_x$	η = dynamic viscosity, Pa.s
	P_9 = dimensionless number, $P_9 = u(y=0)/K$	τ_{av} = average hydrodynamic shear stress, Pa
	Q_y = net axial fluid flow, positive to the air side, mm^3/rev	ρ = density of the medium in a cavitation area, kg/m^3
	t = axial position in the contact width, m	ρ_c = density of the lubricant at cavitation pressure, kg/m^3
	T = shaft friction torque, $\text{N} \cdot \text{m}$

Fig. 2 Solution domain

of the liquid lubricated, fully flooded contact zone of a rotating, perfectly smooth shaft and a rough elastomeric tip of a radial lip seal.

The dimensionless Reynolds equation is solved on the dimensionless rectangular solution domain Ω depicted in Fig. 2. The axial boundaries $Y = 0$ and $Y = 1$ represent the oil and air side of the seal respectively.

2.1 Governing Equations

2.1.1 Universal Reynolds Equation. In the hydrodynamic analysis cavitation is treated while accounting for mass conservation by employing the so-called universal Reynolds equation, as developed by Payvar and Salant (1992). This equation is valid in both the full film and the cavitation region. For an isoviscous lubricant with a dynamic viscosity η it reads

$$\frac{\partial}{\partial x} \left(\frac{h^3}{12\eta} \frac{\partial(F\phi)}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\eta} \frac{\partial(F\phi)}{\partial y} \right) = \frac{U}{2(p_s - p_c)} \frac{\partial}{\partial x} ([1 + (1 - F)\phi]h) \quad (1)$$

where the x and y coordinate represent the unwrapped circumferential direction and the axial direction, respectively. The film thickness is represented by $h(x, y)$ and U equals the shaft surface speed. The meaning of the cavitation index F and the variable ϕ in the full film area differs from that in the cavitation area. They are defined according to

$$\frac{p - p_c}{p_s - p_c} = F\phi \quad (2)$$

where p_s represents a reference pressure and p_c equals the cavitation pressure.

$$\frac{\rho}{\rho_c} = 1 + (1 - F)\phi \quad (3)$$

where ρ represents the density of the medium in the cavitation area and ρ_c equals the density of the lubricant at the cavitation pressure.

$$\begin{cases} F(x, y) = 1, & \text{if } \phi(x, y) \geq 0 \\ F(x, y) = 0, & \text{if } \phi(x, y) < 0 \end{cases} \quad (4)$$

In a full film area the cavitation index F equals $F = 1$ by definition. The variable ϕ then equals a dimensionless pressure; its value is unknown in advance. In a cavitation area the cavitation index F equals $F = 0$ by definition. The quantity $(1 + \phi)$ then equals the fractional film content; again the value of ϕ is unknown in advance. Note that treatment of cavitation by this universal Reynolds equation implements the film rupture boundary condition of Swift (1932) and Stieber (1933) together with the cavitation treatment and the film reformation boundary condition after Jakobsson and Floberg (1957), enabling a mass conserving cavitation simulation. For further details see Payvar and Salant (1992). Substitution of the dimensionless unwrapped circumferential coordinate X , the dimensionless axial

coordinate Y , and the dimensionless film thickness $H(X, Y)$ according to

$$X = \frac{x}{\lambda_x}, \quad Y = \frac{y}{b}, \quad H(X, Y) = \frac{h(x, y)}{G} \quad (5)$$

in the universal Reynolds equation (1) yields the dimensionless universal Reynolds equation, which reads

$$\frac{\partial}{\partial X} \left(H^3 \frac{\partial(F\phi)}{\partial X} \right) + P_1 \frac{\partial}{\partial Y} \left(H^3 \frac{\partial(F\phi)}{\partial Y} \right) = P_2 \frac{\partial}{\partial X} ([1 + (1 - F)\phi]H) \quad (6)$$

where P_1 and P_2 are dimensionless numbers, defined according to

$$P_1 = \left(\frac{\lambda_x}{b} \right)^2 \quad \text{and} \quad P_2 = \frac{6\eta U}{(p_s - p_c)} \frac{\lambda_x}{G^2} \quad (7)$$

The reference pressure p_s , which value is arbitrary, will be chosen equal to the atmospheric pressure p_a .

Atmospheric oil pressures ($p = p_a$) under fully flooded conditions ($F = 1$) are enforced at the two axial boundaries $Y = 0$ and $Y = 1$ by applying the following boundary conditions.

$$\begin{aligned} F(X, Y = 0) &= 1, \quad 0 \leq X \leq 1 \\ F(X, Y = 1) &= 1, \quad 0 \leq X \leq 1 \\ \phi(X, Y = 0) &= 1, \quad 0 \leq X \leq 1 \\ \phi(X, Y = 1) &= 1, \quad 0 \leq X \leq 1 \end{aligned} \quad (8)$$

The film thickness $H(X, Y)$ will be chosen periodic in the unwrapped circumferential (X) direction. Then it suffices to analyze one such period in the unwrapped circumferential (X) direction when periodicity boundary conditions are applied at the boundaries $X = 0$ and $X = 1$, meaning

$$\begin{aligned} \phi(X = 0, Y) &= \phi(X = 1, Y), \quad 0 \leq Y \leq 1 \\ F(X = 0, Y) &= F(X = 1, Y), \quad 0 \leq Y \leq 1 \end{aligned} \quad (9)$$

2.1.2 Film Thickness Equation. The present theory assumes that the film thickness is nominally parallel in the circumferential (X) direction. Figure 3 depicts the macroscopic axial geometry of the film gap. In the present theory the fluid film is established by micro-hydrodynamic lubrication due to a running, smooth shaft surface and a rough seal surface.

In those cases where a tangential seal surface displacement is not considered the dimensionless film thickness equation $H(X, Y)$ is written as, see Fig. 3,

Fig. 3 Dimensionless film thickness

$$H(X, Y) = P_3 + H_a(Y) + H_r(X, Y) \quad (10)$$

where P_3 is a dimensionless constant, $H_a(Y)$ represents the dimensionless macroscopic axial geometry of the oil film and $H_r(X, Y)$ represents the dimensionless seal roughness geometry.

The dimensionless macroscopic axial geometry of the oil film $H_a(Y)$ equals, see Fig. 3,

$$H_a(Y) = \frac{h_a(y)}{G} = \begin{cases} 1 - \cos \left[\frac{\pi}{2P_4} (Y - P_4) \right], & 0 \leq Y \leq P_4 \\ 1 - \cos \left[\frac{\pi}{2(1 - P_4)} (Y - P_4) \right], & P_4 \leq Y \leq 1 \end{cases} \quad (11)$$

The seal tip will be flattened by elastic deformation and wear due to running-in. It is assumed that the film thickness profile (11) reflects the axial pressure distribution of the seal tip. Early work (Hirano et al., 1961) claims that the axial pressure distribution of the seal tip is asymmetric with the axial position in the contact zone whereas recent work (Gabelli et al., 1992) claims that the axial pressure distribution is more uniform due to wear. Using Eq. (11), both situations can be treated by varying the input parameter G .

In those cases where a one-dimensional seal roughness structure is used the following dimensionless one-dimensional sinusoidal (axially aligned) seal roughness geometry $H_r(X, Y)$ is applied

$$H_r(X, Y) = \frac{h_r(x, y)}{G} = -P_5(1 - \cos(2\pi X)), \quad 0 \leq X \leq 1 \quad (12)$$

A two-dimensional seal roughness structure is simulated by using the dimensionless two-dimensional sinusoidal seal roughness geometry $H_r(X, Y)$ according to

$$H_r(X, Y) = \frac{h_r(x, y)}{G} = -P_6(1 - \cos(2\pi X))(1 - \cos(2\pi P_7 Y)), \quad 0 \leq X \leq 1, \quad 0 \leq Y \leq 1 \quad (13)$$

It can be expected that an analysis using the film thickness equation (10) will show a net pumping action because it has asymmetrical geometrical properties in the axial direction. The net pump direction and net pump rate are unknown however.

In the cases where a tangential seal surface displacement is included in the analysis the previously used dimensionless film thickness equation (10) changes to

$$H(X, Y) = P_3 + H_a(Y) + H_r(X - U(Y), Y) \quad (14)$$

where $U(Y)$ equals the dimensionless tangential seal surface displacement.

A radial lip seal surface shows a displacement in the circumferential direction. This is caused by the friction forces exerted on the seal in the contact zone by a rotating shaft. These displacements have been reported by Kawahara et al. (1980), measured by Kammüller (1986), and studied theoretically by Salant and Flaherty (1994). In all these studies it appears that the tangential seal surface displacement varies with the axial coordinate as shown in Fig. 4. All authors conclude that the maximum variation of this tangential displacement in the contact zone equals tens of micrometers and that the position of the maximum displacement is located towards the oil side of the seal. Furthermore, it was found that the seal surface displace-

ment at the oil side ($Y = 0$) is larger than at the air side ($Y = 1$) of the contact zone. In this work the following distribution of the dimensionless tangential seal surface displacement $U(Y)$ with the dimensionless axial coordinate Y is used. It reflects the characteristics just mentioned, see Fig. 4

$$U(Y) = \frac{u(y)}{\lambda_x} = \begin{cases} P_8 \cos \left(\frac{-1}{P_4} (Y - P_4) \arccos(P_9) \right), & 0 \leq Y \leq P_4 \\ P_8 \cos \left(\frac{\pi}{2(1 - P_4)} (Y - P_4) \right), & P_4 \leq Y \leq 1 \end{cases} \quad (15)$$

Note that $U(Y)$ is used to describe the differences in the tangential displacement in the contact zone and not the absolute value of the tangential displacement. The dimensionless number P_8 controls the maximum tangential seal surface displacement relative to the roughness wavelength in x -direction. The dimensionless number P_9 controls the tangential displacement of the oil side relative to the maximum tangential displacement.

$$P_8 = \frac{K}{\lambda_x}, \quad P_9 = \frac{u(y=0)}{K} \quad (16)$$

The present analysis concentrates on solving the dimensionless Reynolds Eq. (6) with the boundary conditions (8) and (9) combined with the dimensionless film thickness equation (10) when the tangential seal surface displacement is absent and with the dimensionless film thickness equation (14) when the tangential seal surface displacement is included in the analysis. For a one-dimensional seal roughness description Eq. (12) applies, and for a two-dimensional seal roughness Eq. (13) is used. In all cases the macroscopic dimensionless axial geometry of the oil film is described by Eq. (11).

2.2 Solution Method. The dimensionless universal Reynolds Eq. (6) was discretized according to the finite volume method, see Patankar (1980). Commonly used discretizations were employed, see Payvar and Salant (1992) or Patankar (1980). Since both $\phi(X, Y)$ and $F(X, Y)$ are unknown in Eq. (6) some iterative procedure has to be followed. To start the iterative process full film conditions, i.e., $F(X, Y) = 1$, are assumed in the entire solution domain. The resulting linear system of equations was solved for a given cavitation index $F(X, Y)$ using banded Gauss elimination. Next the cavitation index was updated according to Eq. (4). Subsequently a new linear system of equations was formulated and solved using the updated cavitation index. This procedure was continued until no

Fig. 4 Dimensionless tangential seal surface displacement

Table 1 Numerical input data of the operating conditions

dynamic viscosity of the oil η , [Pa · s]	$1 \cdot 10^{-2}$
shaft surface speed U , [m/s]	4
atmospheric pressure p_a , [Pa]	$1 \cdot 10^5$
cavitation pressure p_c , [Pa]	$0.7 \cdot 10^5$
contact width b , [m]	$100 \cdot 10^{-6}$
axial position seal tip t , [m]	$30 \cdot 10^{-6}$
axial amplitude G , [m]	$0.5 \cdot 10^{-6}$
maximum seal surface displacement K [m]	0 to $40 \cdot 10^{-6}$

differences in the cavitation index $F(X, Y)$ and $\phi(X, Y)$ were found.

3 Results and Discussion

The model developed in the previous section is used to investigate the lubrication, friction and pumping action of a fully flooded radial lip seal involving a one-dimensional seal roughness structure with and without a tangential seal surface displacement. This is described in Section 3.1. In addition a two-dimensional seal roughness structure with and without a tangential seal surface displacement is analyzed in Section 3.2. Table 1 lists the parameters that are used in these two sections.

The numerical results will be compared to measurements found in the literature which are typical for radial lip seals. The following data are used as guidelines. The range of the average contact pressure p_{av} equals $3 \cdot 10^5 \leq p_{av} \leq 10 \cdot 10^5$ Pa (Ott, 1983; Gabelli et al., 1992). The net axial fluid flow Q_y of a fully flooded radial lip seal with both sides at atmospheric pressure and a 70 mm shaft running at 1000 rev/min with an oil reservoir temperature of 90°C is of the order of $Q_y = 0.025$ mm³/rev (Horve, 1987). The shaft friction torque T of a fully flooded radial lip seal of a 70 mm shaft running at 1000 rev/min and an oil reservoir temperature of 70°C is of the order of $0.15 \leq T \leq 0.2$ N.m (Ott, 1983; Kammüller, 1986).

The authors are aware of the fact that these guideline values as well as the numerical input data which will be mentioned in the sequel should preferably be derived from the same experiment and sufficiently averaged. Unfortunately, these data are currently not available in the public literature.

3.1 One-Dimensional Seal Roughness Structure. In this section first the simulation results involving a one-dimensional sinusoidal seal roughness structure without a tangential seal surface displacement are discussed. Next the results involving the same seal roughness structure but with a tangential seal surface displacement are shown.

Table 2 lists the parameters that are used in this section only. The choice of the seal roughness wavelength λ_x is based on the work of Kammüller (1986) and the choice of $u(y = 0)$ on Kammüller (1986) and Salant and Flaherty (1994). The choice of the seal roughness amplitude L is based on Jagger and Walker (1966) and Nakamura and Kawahara (1984). In each case two seals are analyzed denoted with seal R and seal S where the former represents a rougher surface compared to the latter. The results of the mathematical analysis are determined entirely by

Table 2 Numerical input data of the one-dimensional seal roughness structure

	Seal R	Seal S
wavelength in x direction λ_x , [m]	$10 \cdot 10^{-6}$	$10 \cdot 10^{-6}$
seal roughness amplitude L , [m]	$2 \cdot 10^{-6}$	$0.5 \cdot 10^{-6}$
film thickness constant h_0 , [m]	$2.15 \cdot 10^{-6}$	$0.65 \cdot 10^{-6}$
tangential displacement at oil side $u(y = 0)$, [m]	0.25 K	0.25 K

Table 3 Numerical input data of the dimensionless numbers

dimensionless number	Seal R	Seal S
P_1 [—]	0.01	0.01
P_2 [—]	320	320
P_3 [—]	4.3	1.3
P_4 [—]	0.3	0.3
P_5 [—]	2	0.5
P_8 [—]	0 to 4	0 to 4
P_9 [—]	0.25	0.25

the dimensionless numbers listed in Table 3. The numerical data of this table follow from the data listed in Tables 1 and 2. Note that in the case where a tangential seal surface displacement is absent the dimensionless numbers P_8 and P_9 of Table 3 are not relevant.

Figures 5(a) and 6(a) present the numerically obtained distribution of $\phi(X, Y)$ and $F(X, Y)$ of seal S belonging to the general film thickness equation without a tangential seal surface displacement Eq. (10) and the one-dimensional roughness Eq. (12). Comparable figures can be obtained in the case of seal R. It appears that the maximum of the hydrodynamic pressure distribution in the axial (Y) direction is located more to the oil than to the air side of the seal, both in case of seal R and seal S.

Table 4 shows some numbers of practical importance that can be calculated rather straightforward once the solutions shown previously have been obtained. The following quantities are shown:

- The pump rate Q_y of a fully flooded seal with no pressure difference across the seal.

This pump rate is expressed by the net axial fluid flow Q_y at an axial boundary. Its value is computed by assuming that the entire seal surface is covered with the periodic surface roughness structure under consideration. Note that Q_y is positive in the positive y -direction, i.e. to the air side.

- The load-carrying capacity of the generated oil film pressures which is expressed by the average hydrodynamic pressure p_{av} .

Fig. 5 The solution $\phi(X, Y)$ for seal S, (a) without a tangential seal surface displacement, $P_8 = 0$, (b) with a tangential seal surface displacement, $P_8 = 1$

Fig. 6 The full film ($F = 1$) and the cavitation area ($F = 0$) for seal S, (a) without tangential seal surface displacement, $P_8 = 0$, (b) with tangential seal surface displacement, $P_8 = 1$, (c) with tangential seal surface displacement, $P_8 = 4$

- The viscous friction force acting on the shaft surface which is expressed by the average hydrodynamic shear stress acting on the shaft surface τ_{av} .
- The friction torque T exerted on the rotating shaft by an entire seal according to $T = \pi/2 d^2 b \tau_{av}$, assuming a shaft diameter d of 70 mm.

According to Table 4 the net axial fluid flow appears to be directed to the air side of the seal for both seal R and S. This contradicts the experimental observations of the oil side directed net axial oil flow at fully flooded radial lip seals. Besides this, the order of magnitude is too small by a factor of about 35 to 500 compared to the experimentally observed values mentioned previously. Note that the net axial fluid flow at the two axial boundaries $Y = 0$ and $Y = 1$ will be equal due to the mass conserving cavitation simulation. The average hydrodynamic pressure generated is too low when compared to the experimentally observed values. The discrepancy is a factor of about 3 to 4. This also applies to the shaft friction torque. Here the discrepancy is a factor of about 3.

Now the results of the same seal roughness structure are shown but with a tangential seal surface displacement included in the analysis.

Table 4 Numerical results without a tangential displacement, $P_8 = 0$

	Seal R	Seal S
net axial fluid flow Q_y , [mm ³ /rev]	0.00068	0.000046
average pressure p_{av} , [Pa]	$1.8 \cdot 10^5$	$1.5 \cdot 10^5$
average shaft shear stress τ_{av} , [Pa]	$6.0 \cdot 10^4$	$6.9 \cdot 10^4$
shaft friction torque T , [N·m]	0.046	0.053

Table 5 Numerical results with a tangential displacement, $P_8 = 1$

	Seal R	Seal S
net axial fluid flow Q_y , [mm ³ /rev]	-0.49	-0.064
average pressure p_{av} , [Pa]	$3.3 \cdot 10^5$	$5.8 \cdot 10^5$
average shaft shear stress τ_{av} , [Pa]	$7.8 \cdot 10^4$	$1.0 \cdot 10^5$
shaft friction torque T , [N·m]	0.06	0.079

Figures 5(b) and 6(b) present the numerically obtained distribution of $\phi(X, Y)$ and $F(X, Y)$ of seal S belonging to the general film thickness profile with a tangential seal surface displacement Eq. (14) and the one-dimensional roughness Eq. (12) using $P_8 = 1$. Comparable figures can be obtained in the case of seal R. Comparison with Figs. 5(a) and 6(a) shows the increased maximum pressure and the reduction of the extent of the cavitation area. These two observations suggest an increased load-carrying capacity, as will be confirmed numerically in the sequel.

Table 5 shows the net axial fluid flow, average pressure, average shear stress and friction torque. The net axial fluid flow is directed to the oil side of the seal now. This direction of the net axial oil flow is in agreement with experimental observations of fully flooded radial lip seals. The net axial oil flow is more in agreement with the experimentally observed values, however a factor of about 3 to 20 too high now.

The average hydrodynamic pressure is a factor 2 to 4 higher compared to the values of Table 4. These values now agree with the lower limit of the experimentally observed average contact pressure mentioned previously.

The shaft friction torque values have increased by a factor of 1.4 compared to the values of Table 4 but are still too low by a factor of about 2.

Figure 6 shows the extent of the different cavitation areas of seal S using $P_8 = 0$ (where a tangential seal surface displacement is absent) and $P_8 = 1$ and $P_8 = 4$, respectively (where the tangential displacement gradually increases). It appears that the extent of the cavitation area reduces from one relatively large area which extends over the full axial width to two relatively small cavitation areas: one at each axial boundary.

The dimensionless number P_8 controls the magnitude of the maximum tangential seal surface displacement relative to the seal roughness wavelength in the X-direction. The range of P_8 values considered is restricted to $0 \leq P_8 \leq 4$. The upper limit of this range is based on the maximum differences of the tangential seal surface displacement within the contact zone of 20 to 40 μm , as measured by Kammüller (1986), in combination with a seal roughness wavelength of $\lambda_x = 10 \mu\text{m}$.

Figure 7 shows the effect of a variation of the dimensionless number P_8 on the net axial fluid flow Q_y , the average pressure p_{av} and the shaft friction torque T . The net axial fluid flow and the average pressure increase with respect to the values of P_8 considered. The experimentally observed order of magnitude of the net axial fluid flow and average pressure level can be obtained rather easily. The shaft friction torque remains relatively constant in the range of P_8 considered and remains below the experimentally observed values by a factor 2 to 3.

3.2 Two-Dimensional Seal Roughness Structure. In

this section the results involving a two-dimensional sinusoidal seal roughness structure without a tangential seal surface displacement are first discussed. Next the results involving the same seal roughness structure but with a tangential seal surface displacement are shown.

Table 6 lists the parameters that are used in this section only. The results of the mathematical formulation are determined entirely by the dimensionless numbers listed in Table 7. The numerical data of this table follow from the data listed in Tables 1 and 6. Note that in the case where a tangential seal surface

(a)

(b)

(c)

Fig. 7 (a) The net axial fluid flow Q_y , (b) average pressure p_{av} , (c) shaft friction torque T of seal R (solid curve) and S (dashed curve) versus the dimensionless number P_8

displacement is absent the dimensionless numbers P_8 and P_9 of Table 7 are not relevant.

Figures 8(a) and 9(a) present the numerically obtained distribution of $\phi(X, Y)$ and $F(X, Y)$ of seal S belonging to the general film thickness equation without a tangential seal surface displacement Eq. (10) and the two-dimensional roughness geometry Eq. (13). Comparable figures can be obtained in the case of seal R.

It appears that the maximum of the envelope of the hydrodynamic pressure distribution in the axial (Y) direction is located

Table 6 Numerical input data of the two-dimensional seal roughness structure

	Seal R	Seal S
wavelength in x direction λ_x , [m]	$10 \cdot 10^{-6}$	$10 \cdot 10^{-6}$
wavelength in y direction λ_y , [m]	$10 \cdot 10^{-6}$	$10 \cdot 10^{-6}$
seal roughness amplitude M , [m]	$2 \cdot 10^{-6}$	$0.5 \cdot 10^{-6}$
film thickness constant h_0 , [m]	$2.15 \cdot 10^{-6}$	$0.65 \cdot 10^{-6}$
tangential displacement at oil side $u(y=0)$, [m]	0.25 K	0.25 K

more to the oil than to the air side of the seal, both in case of seal R and seal S. Despite flow interaction between the individual pressure distributions of the asperities, the pressure level between the asperities drops to values equal to the pressure level at the two axial boundaries. This indicates that there is hardly any cooperation present between the various asperities in generating the axial pressure distribution.

Table 8 shows some numerical results of practical importance for seal R and S. The net axial fluid flow appears to be directed to the air side of the seal in both cases of seal R and seal S. This is in contrast with experimental observations of fully flooded radial lip seals. Besides this, the order of magnitude of the computed net axial oil flow is too low by a factor of about 100 to 650.

The average hydrodynamic pressure generated is too low by a factor of about 5 to 6 compared to values observed experimentally. The same applies to the shaft friction torque which is too low by a factor of 3 to 7.

Now the results of the same seal roughness structure but with a tangential seal surface displacement included, are shown.

Figures 8(b) and 9(b) present the numerically obtained distribution of $\phi(X, Y)$ and $F(X, Y)$ of seal S belonging to the general film thickness equation with a tangential seal surface displacement Eq. (14) and the two-dimensional roughness geometry Eq. (13) using $P_8 = 1$. Comparable figures can be obtained in the case of seal R. Since the pressure between the various asperities reaches levels beyond the pressure level at the two axial boundaries it appears that the various asperities

(a)

(b)

Fig. 8 The solution of $\phi(X, Y)$ for seal S, (a) without a tangential seal surface displacement, $P_8 = 0$, (b) with a tangential seal surface displacement, $P_8 = 1$

Table 7 Numerical input data of the dimensionless numbers

Dimensionless number	Seal R	Seal S
P_1 [-]	0.01	0.01
P_2 [-]	320	320
P_3 [-]	4.3	1.3
P_4 [-]	0.3	0.3
P_6 [-]	1	0.25
P_7 [-]	10	10
P_8 [-]	0 to 4	0 to 4
P_9 [-]	0.25	0.25

are more cooperative in generating the total pressure distribution than in the case where a tangential seal displacements is absent.

Comparison with Figs. 8(a) and 9(a) shows the increased maximum pressure and the reduction of the cavitation area. These two observations suggest an increased load-carrying capacity, as will be confirmed numerically in the sequel.

Table 9 presents some numerical results of practical interest. The net axial fluid flow is directed to the oil side of the seal. This is in agreement with experimental observations of fully flooded radial lip seals. The order of magnitude of the net axial oil flow that can be obtained are relatively close to the experimental values found in the literature, a factor 5 too high for seal R and a factor 2 too low for seal S. The average hydrodynamic pressure is a factor of 1.3 to 1.6 times higher compared to the values of Table 8. These values are now more close to the experimentally observed values, but a factor 3 to 4 times too low. The friction torque values have increased only slightly compared to the values of Table 8, and are too low by a factor of about 3 to 6.

Figure 9 shows the extent of the cavitation areas of seal S in the case of $P_8 = 0$, $P_8 = 1$ and $P_8 = 4$. It appears that both the number of cavitation areas and the extent of a cavitation area reduce with increasing values of P_8 . Micro-cavitation areas in lubricated radial lip seals have been reported by Jagger (1965), Nakamura and Kawahara (1984) and Nakamura (1987). The existence of large cavitation areas is considered to be a detrimental factor in the performance of liquid lubricated seals. Air ingestion and the stability of the oil-air axial free boundary which develops at the air side of a radial lip seal are responsible to this as reported by Bootsma (1975) in the case of grooved bearings. The reduction of the extent of the cavitation area as present in Figs. 6 and 9 is likely to be considered as beneficial in achieving reliably operating seals.

Figure 10 shows the effect of the tangential seal surface displacement through a variation of the dimensionless number P_8 on the net axial fluid flow, the average pressure and the shaft friction torque. The range of P_8 values is again restricted to $0 \leq P_8 \leq 4$ for reasons given previously. The net axial fluid flow and the average pressure increase with respect to the values of P_8 considered. The experimentally observed values of the net axial fluid flow and the lower limit of the average pressure level can be obtained. The shaft friction torque remains relatively constant and remains below the experimentally observed values.

In this paper fully flooded conditions at the oil and at the air side of the seal were used. The leakage-free state of the liquid lubricated contact zone of a radial lip seal can be explained by

Fig. 9 The full film ($F = 1$) and the cavitation area ($F = 0$) for seal S, (a) without tangential seal surface displacement, $P_8 = 0$, (b) with tangential seal surface displacement, $P_8 = 1$, (c) with tangential seal surface displacement, $P_8 = 4$

adding the existence of the oil-air axial free boundary at the air side of the seal to the upstream pumping model described here, see e.g., Bootsma (1975) or Salant (1992b).

4 Conclusions

From the hydrodynamic analysis of a coaxially running, smooth shaft surface and a rough, radial lip seal under fully flooded conditions it follows that:

- The tangentially nondeforming, one- and two-dimensional seal roughness structures used in this work yield a pumping direction that disagrees with the pumping direction observed experimentally;
- The one- and two-dimensional seal roughness structures used in this work where a tangential seal surface displacement is included in the analysis yield a pumping direction that agrees with the experimentally observed pumping direction; the order of magnitude of the experimentally observed pump rate, load-carrying capacity and friction torque found in the literature can be simulated;
- A comparison of the results of tangentially nondeforming and tangentially deforming seal surfaces shows that for both the one- and the two-dimensional seal roughness structures used in this work the tangentially deforming cases show, compared with the tangentially non-deforming cases, a reversal of the net axial pumping direction, an increase of the net pump rate, an

Table 8 Numerical results without a tangential displacement, $P_8 = 0$

	Seal R	Seal S
net axial fluid flow Q_y , [mm ³ /rev]	0.00026	0.000038
average pressure p_{av} , [Pa]	$1.08 \cdot 10^5$	$1.12 \cdot 10^5$
average shaft shear stress τ_{av} , [Pa]	$2.92 \cdot 10^4$	$5.79 \cdot 10^4$
shaft friction torque T , [N·m]	0.022	0.045

Table 9 Numerical results with a tangential displacement, $P_8 = 1$

	Seal R	Seal S
net axial fluid flow Q_y , [mm ³ /rev]	-0.14	-0.013
average pressure p_{av} , [Pa]	$1.45 \cdot 10^5$	$1.8 \cdot 10^5$
average shaft shear stress τ_{av} , [Pa]	$3.12 \cdot 10^4$	$6.29 \cdot 10^4$
shaft friction torque T , [N·m]	0.024	0.048

Fig. 10 (a) The net axial fluid flow Q_y , (b) average pressure p_{av} , (c) shaft friction torque T of seal R (solid curve) and S (dashed curve) versus the dimensionless number P_8

increase of the load-carrying capacity and a reduction of the number and extent of the cavitation areas;

—A relatively simple model is able to predict a behavior similar to the basic operation of a radial lip seal. This does not preclude the presence of other mechanisms. A definitive answer requires a consistent set of experiments.

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