A new concept in rotary shaft seal lubrication : viscoelastohydrodynamic (VEHD) lubrication

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A New Concept in Rotary Shaft Seal Lubrication: Viscoelastohydrodynamic (VEHD) Lubrication

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In practice rotary shaft seals will experience a small-amplitude dynamic excitation. It is shown that under periodic excitation circumferentially nonuniform clearances develop due to viscous seal material behaviour. The nearby fluid will fill these gaps, so entrainment and squeeze effects can develop fluid pressures which are sufficiently high to overcome the radial preload. Viscous seal properties are essential in this type of EHD lubrication. Hence it is designated *visco*elastohydrodynamic (VEHD) lubrication. At present, this lubrication concept is the only macrohydrodynamic theory that explains the existence of a consistent circumferentially nonuniform film geometry of appreciable dimensions. Moreover, calculated values of film thickness and friction are in agreement with experimental data.

1 INTRODUCTION

Rotary shaft seals are widely employed in industry to prevent (a) fluid leakage and (b) dirt and dust penetration into the sealed fluid, at a rotating shaft/seal interface. This paper discusses the lubrication mechanism of a standard lip seal, without supplemental sealing devices, see Figure 1. The study of rotary shaft seals is not a mere academic one. As seal life, friction and leakage have to keep pace with the ever increasing power densities and speeds of machinery, and with environmental regulations, the sealing and lubrication principles should be well understood.

Until now, rotary shaft seal research has been focussed mainly on the *sealing mechanism*. Müller [1] and Stakenborg [2] provide a survey of the most recent theoretical sealing concepts. Stakenborg distinguishes passive and active sealing models. The former indicate a stationary condition, whereas the latter denote a dynamic pumping effect. An example of the first category is the surface tension concept of Jagger [4]. The most important active sealing models are (a) Taylor vortices in the thick film, resulting in a suction effect, (b) micro-grooves formation due to tangential deformation in the contact, yielding a pumping effect, (c) reciprocating lip motion due to shaft eccentricity or seal tilt, and (d) oscillatory squeeze motions in the fluid film, creating a pumping action through fluid film inertia ([1], [2], and [3]).

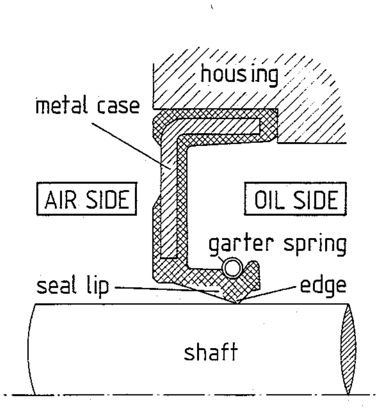

Fig.1 Rotary shaft seal lay out.

Since 1956, when Jagger published his papers on the sealing [4] and lubrication [5] mechanism of rotary shaft seals, it is generally accepted that a hydrodynamic fluid film exists between the seal lip and the shaft. Jagger [5] finds that vibration is conducive to oil-film formation. The measured oil-film thickness increases considerably when the out-of-squareness of the counterface is enlarged. It is striking to learn that this finding has not received much attention in the past 35 years. Only a small number of reports on the generation of a fluid film exists. If there were no fluid film, no leakage would occur. Hence it is worth the effort to study fluid film formation.

The literature provides only a small number of theoretical studies into the *lubrication mechanism*. A major problem, also perceived by Jagger, is the apparently parallel film. One candidate mechanism, put forward by, a.o., Jagger and Walker [6], is the micro-asperity lubrication concept, based on the surface roughness of the shaft and seal. These authors included the elastic flattening of the seal asperities due to local pressure distributions. However, seal deformations will not be *local*, at the top of the asperities, but rather be *global*, at the entire asperity, leading to a different surface texture under operating conditions. This is evidenced by an experimental/theoretical study on reciprocating seals by Kanters [7], who concluded that at thin film conditions lip friction and leakage are better described by smooth surface lubrication than by a flow factor method for rough full film lubricated surfaces.

These concepts are all rotatory symmetric (except the reciprocating lip model, see below). Consequently they all yield, if at all, a circumferentially uniform film. However, the origin of fluid film formation remains undisputed, or remains ambiguous.

It is questionable if a circumferentially uniform film will exist. The few experimental results which are reported all show a nonuniform film, see under section 2. Nonuniform films can be attributed to (1) seal misalignment (case (c) mentioned above), and (2) shaft vibration or shaft out-of-roundness. The former ends up in unrealistic thin films, as is shown in the appendix to [8], while the latter two result in a *dynamic* excitation of the seal rubber.

In practice *dynamic* conditions at the seal-fluid interface will prevail. If the seal elastomer had (non)linear *elastic* properties only (and no inertia), the seal surface would simply follow the shaft surface motions. Consequently, no fluid film would develop. As rubbers always show *viscous* properties (and will have some inertia), resulting in a poorer "followability" of the seal, separation may occur. Hence, fluid pressures could be generated through the entrainment effect. This paper attempts to treat fluid film formation as a macrohydrodynamic phenomenon (without surface irregularities), due to dynamic excitation of a viscoelastic (and inertial) solid. Its objective is to investigate whether a (nonuniform) fluid film of appreciable thickness is possible or not. Isothermal conditions (at 30^{0}C) were assumed. The seal in this study has dimensions 70x100x10/9.5 mm, an inner diameter of Ø 68 mm and is made of Nitrile. The geometry is for a virgin seal. The shaft has a diameter of Ø 70 mm.

1.1 Notation

b	seal/shaft contact width	[m]
e	eccentricity	[m]
E*	complex valued dynamic stiffness	[N/m^2]
E_0	stress relaxation function constant	[N/m^2]
E_1	stress relaxation function constant	[N/m^2]
f	frequency	[s^{-1}]
g	compliance function in time domain	[m/Pa.s]
h	local film thickness	[m]
H	transfer function in frequency domain	[Pa/m]
j	complex number	[-]
p	fluid film pressure	[Pa]
t	time	[s]
T	temperature	[^{0}K]
x	displacement in radial direction	[m]
y	axial contact coordinate	[m]
δ	radial lip displacement	[m]
η	dynamic viscosity	[Pa.s]
ϑ	circumferential coordinate, along seal	[-]
τ	time integration variable	[t]
τ_1	time constant	[t]
ϕ	circumferential coordinate	[-]
ω	angular velocity	[s^{-1}]

indices:

cav	cavitation
d	dynamic
ref	reference
s	static
se	seal
sh	shaft
t	total
w	whirl
3	at maximum static pressure

2 STATE OF THE ART

At present, it would suffice to cite Hirano et al. [9] to describe the state of the art. Even after 30 years of research, ..."the mechanism of the generation of the hydrodynamic oil film between the lip and the shaft has been little explained" ([9], p.1)! A rigorous theoretical treatment of this problem is still lacking. This is not surprising, since a rotary shaft seal is an extremely complex machine element, with many design and application variables, as [10] demonstrates.

As mentioned in the introduction, the apparently parallel fluid film has been a puzzling issue. Therefore, researchers resorted to micro-asperity lubrication mechanisms, see also Hirano and Ishiwata [11], who consider the shaft surface roughness, and, more recently, Gabelli [12], who took both the shaft and seal irregularities into account.

Lebeck [13] states, in his analysis of the most important load support creating mechanisms in contacting mechanical face seals, that all of the parallel sliding lubrication mechanisms considered are too weak to account for the observed friction behaviour, *except a deviation from the parallel.* In a later contribution to this problem Lebeck [14] shows that a mixed friction parallel face model (without any hydrodynamic entrainment action) serves as an *upper* bound on measured seal friction, while a mixed friction wavy seal model (including hydrodynamic effects) appears to yield a *lower* bound on it. Hence, deviations from parallelism are most likely to occur. This is even more true for highly compliant rotary shaft seals: the surface roughness texture will be flattened by the high fluid film pressures.

The literature on rotary shaft seals shows a few *experimental studies* of the lubrication mechanism. Iny and Cameron [15] performed film thickness measurements at an axial seal test rig. They found that seal/shaft gap fluctuations in sliding direction generate an oil film which can carry a considerable load. Ishiwata and Hirano [9] recorded lip motion by lighting up embedded glass particles at the lip end face. They observed elliptical lip motions, and reasoned that the lip did not follow the shaft motions due to viscous rubber properties. Symons [16] measured the motions of a spot on the lip by a stereo (grammophone) cartridge transducer. He found that increased shaft out-of-roundness and lobing resulted in reduced tangential motion, indicating that a thicker (full) film had been established, but did not provide an explanation. Arai [17] investigated the pumping action, but also noticed that a marked lift-off occurred even at low speeds under dry running conditions, which apparently disappeared when oil lubrication was used. Gawlinski and Schouten [18] measured film thicknesses in a radial lip seal. One of their results is reproduced in Figure 2. Although they did not mention explicitly the fluctuation of the film geometry in circumferential direction, this is a remarkable result. Papers [15] and [18] employed a capacitance measurement technique. Prati [19] also observed separation of the rubber seal from a shaft with dynamic runout, at very low frequencies. In addition, he provides a three parameter solid model with viscoelastic and mass effects for the radial behaviour of the entire seal.

In a very recent paper Pohl and Gabelli [20] use a magnetic fluid and a magnetic flux sensor (like in a tape recorder) to determine the fluid film thickness in the seal/shaft interface. Just as in [18], the film profile is close to parallel in axial direction under realistic preloads. Unfortunately, no results in circumferential direction are reported.

All experimental papers report full film lubrication conditions in a wide range of operating

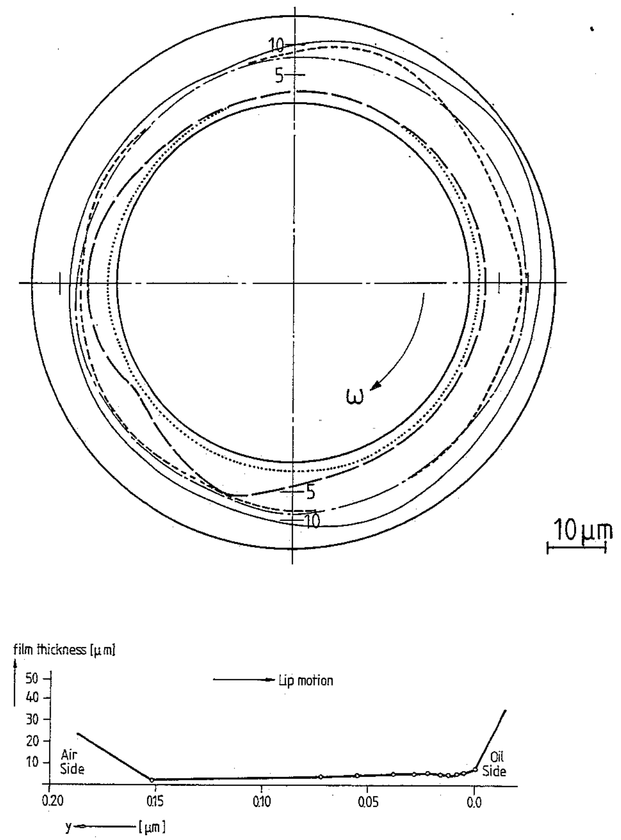

Fig.2 Radial (top) and axial (below) local film thickness measurements from [18] at 5 Hz.

conditions, film thicknesses typically being of the order of 1 μm. From local film thickness measurements in [15] and [18] and lip motion studies [9] and [16] it can be concluded that the film shape is nonuniform. The nonparallel film can be created by (a) viscous, and/or (b) inertial behaviour of the rubber solid. This will be discussed in the next two sections.

3 DYNAMIC BEHAVIOUR OF RUBBER

In practice always some dynamic excitation exists, but the amplitude is low under normal conditions. If a nonuniform film is going to occur, it will be a necessity for the solid surfaces to separate. The problem now is, that static preloads and strains are very high, in order to maintain a good "followability" of the seal, even at elevated temperatures. In this way, a low-amplitude excitation is never going to yield separation. However, the dynamic mechanical behaviour of seal rubbers lends a helping hand in this matter.

The seal is usually made of an elastomer. The mechanical behaviour of these rubbers is thermo-viscoelastic, i.e. the elastic modulus is dependent on temperature and frequency (or time). See Figure 3, which provides a schematic for this behaviour. At low frequencies (at the rubbery state), the equilibrium value for the dynamic stiffness E* is low, while at high frequencies (at the glassy state) the stiffness is much higher. An increasing temperature will reduce the stiffness, except at very low or high frequencies. The transition in the dynamic stiffness, which is called the glass transition, can be rather pronounced, of the order of two orders of magnitude and more. This

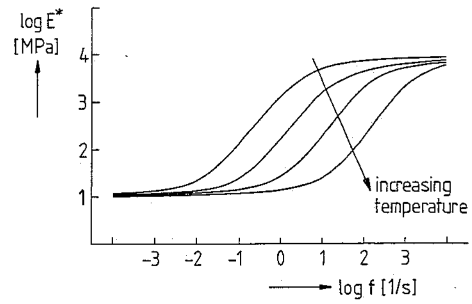

Fig.3 Schematic of the dynamic stiffness E* of nitrile rubber as function of the excitation frequency at different temperatures.

glass transition is also temperature dependent. The purpose of this study is to ascertain whether a fluid film can at all be established or not. Therefore, it is not of primary importance *where* the glass transition exactly turns up, but *that* it happens. Hence, this transition can strongly affect the contact stress distribution.

Under separation conditions the dynamic stresses will cancel out the (high) static stresses. Under low-amplitude excitation this is possible only if the stiffness increases dramatically with respect to the static case. Owing to the observed mechanical behaviour, this might be the case: a marked transition from the rubbery state (*low* stiffness) to the glassy state (*high* stiffness) could yield separation. In plain words, if the dynamic excitation is of the order of 1% of the static strain, an increase in stiffness will be required of the order of a factor 100.

Figure 4a shows a simple spring-dashpot (Maxwell/Voigt) model, which is frequently employed to describe the linear viscoelastic behaviour of a solid. If the following data are used (for more details, see also [21] and [22]):

$$E_0 = 10^1 \text{ MPa};\ E_1 = 10^4 \text{ MPa};\ \tau_1 = 0.003 \text{ s}$$
$$(\text{or } \eta_1 = \tau_1 E_1 = 30 \text{ MPa.s})$$

The effective (dynamic) modulus of elasticity, which has a complex value, can be described by its magnitude $|E^*|$ and its argument arg(E*), see Figure 4b. Note that due to the viscous action the stress is running *ahead* of the strain.

In this study it was assumed that the influence of the dynamic strain amplitude on the material stiffness may be neglected. The small-amplitude oscillations will be superposed on the nonlinear static state, see the next section.

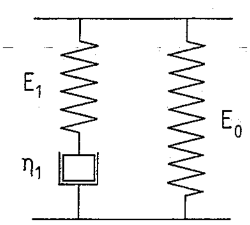

Fig.4a Three parameter viscoelastic solid model.

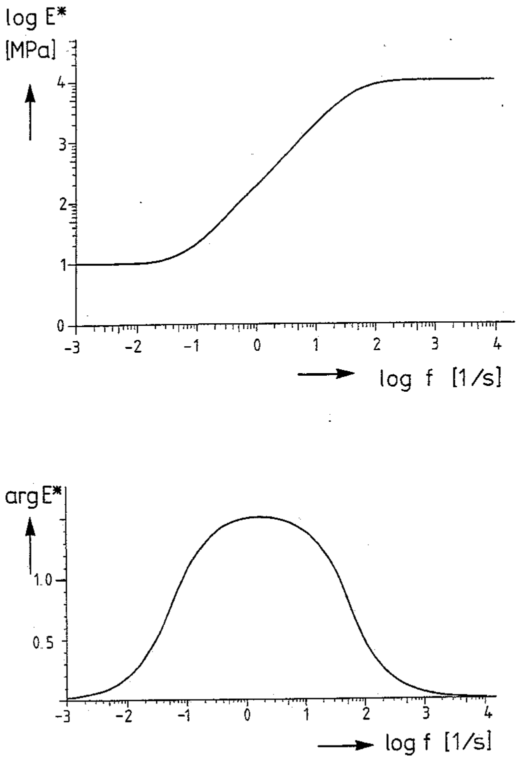

Fig.4b The complex valued dynamic stiffness E* for the solid model of Figure 4a.

In order to be able to explore the fluid film formation, the frequency dependent stiffness of the contact zone must be defined. Therefore it is needed to investigate the dynamic behaviour of the entire seal.

4 DYNAMIC BEHAVIOUR OF A (DRY) ROTARY SHAFT SEAL

It is assumed that the seal geometry and deformation are both axisymmetric. This assumption is not far off reality, since the maximum dynamic radial deformation is always much smaller than the axisymmetric preloaded deformation (2 orders of magnitude or less). The theory developed by Morman and Nagtegaal [23], as implemented in the nonlinear FEM program MARC [24], is used to model the rubber behaviour. A distinction is made between:

- -a *static* (nonlinear) component: the static deformation yields a static contact stress distribution;
- -a *dynamic* (linearized) component: the dynamic deformation yields a dynamic contact stress component; in total:
- -the superposition principle holds for small-amplitude vibrations.

This is illustrated by Figure 5. Under static conditions a high contact stress exists. Under low frequency excitation (low stiffness) the variations in the *total* stress are small. At very high frequencies (high stiffness) the departures from the static stress ditribution become enormous, and the total stress may drop to zero or negative values. As it is impossible to create normal tensile stresses in the contact, separation must occur. It is not

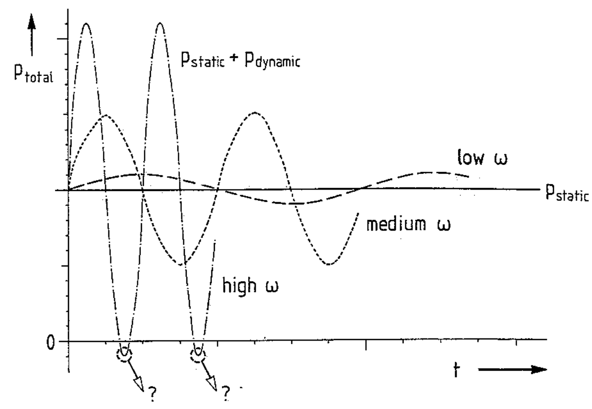

Fig.5 Total contact stress response as a function of excitation frequency.

important *how* the gap geometry looks like, but *that separation occurs*. The Fourier spectrum of a fluid pressure distribution (see the next sections) will be composed of many components. As a fluid can not sustain a negative stress (at least not for any duration of time), the total pressure can not drop below the fluid cavitation pressure. Of course several Fourier components of the pressure may on their own become negative, as long as the total pressure is equal to or larger than the cavitation pressure.

The static contact stress component is calculated using the Mooney-Rivlin incompressible material model, see [21]. I-DEAS software was used as a pre- and postprocessor ([22], [25]). The dynamic contact stress component is calculated by employing a linearized viscoelastic model, like in Figure 4. In this case more springs and dashpots were added (8 resp. 7, see [22]). The approach can be described by:

$$\text{input:} \quad x_t(t) = x_s + x_d(t) = x_s + |x_d| \cdot \cos(\omega t) \tag{1.1}$$

$$\text{output:} \quad p_t(t,y) = p_s(y) + p_d(t,y) = p_s(y) + |p_d(y)| \cdot \cos(\omega t + \psi) \tag{1.2}$$

where the *static* deformation x_s corresponds to a *static* stress distribution $p_s(y)$, and the *dynamic* deformation $x_d(t)$ correlates with a *dynamic* stress distribution $p_d(t,y)$, and ψ is the phase angle. The excitation is harmonic.

Figure 6 shows the FEM model of the seal. Only the lip part has been modeled, because the metal case makes the upper part very stiff.

The result of the static calculations is given by Figure 7. It can be concluded that for a brand new seal the maximum stress is attained at the third node (at y=0.018 mm) and is as high as 4.7 MPa, and that the contact width b is as narrow as 0.075 mm. It can be expected that this third node will be the last to remain in contact with the seal.

As the system is linear as far as it concerns its dynamic behaviour, the *dynamic* deformation $\delta(t,y)$ can now be described by a convolution integral in the time domain:

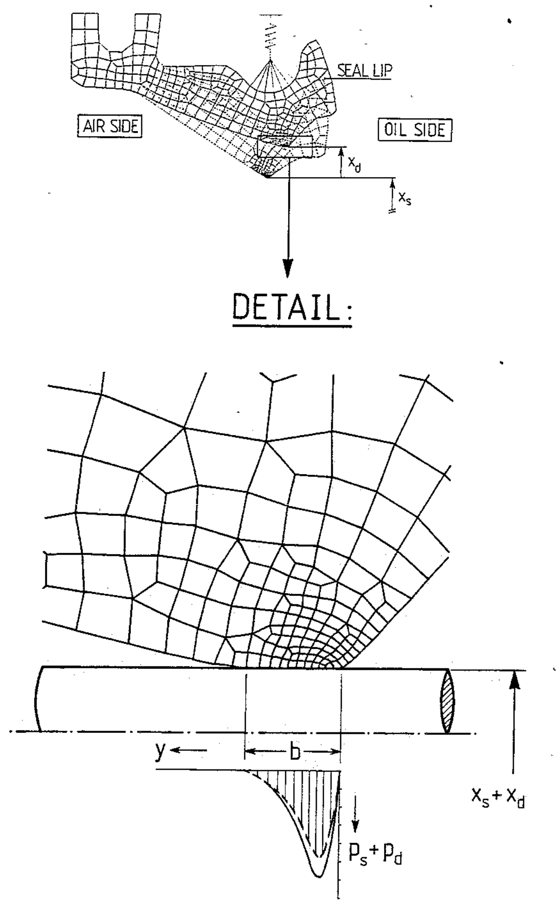

Fig.6 FEM seal model. See text for explanation.

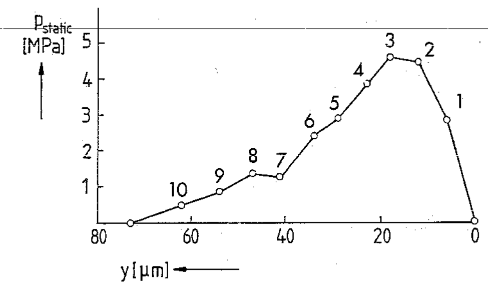

Fig.7 Static contact stress distribution.

$$\delta(t,y)= \int_{\tau=-\infty}^{\tau=\infty} g(t-\tau,y).p_d(\tau,y).d\tau \qquad (2.1)$$

where

$$g(t,y)= \int_{f=-\infty}^{f=\infty} \{1/H(f,y)\}.\exp(j2\pi ft).df \qquad (2.2)$$

and H(f,y) is a (complex) transfer function or dynamic stiffness in the frequency domain, g(t,y) a compliance function in the time domain, and $p_d(t,y)$ the dynamic stress in the contact.

It was assumed that x_d=0.01 mm, a realistic value for normal roller bearings, and a little bit low for sliding bearings for a 70 mm dia. shaft. Figure 8 shows the stiffness magnitudes (transfer function magnitudes) for four contact nodes. The curve identification numbers correspond to the nodal

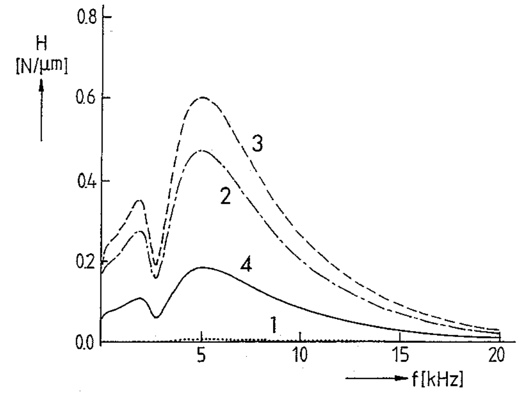

Fig.8 Dynamic stiffness (transfer function) for several contact nodes, numbers conform Fig.7.

point numbers of Figure 7. Inertia of the rubber and garter spring is included; see [21] for the behaviour without inertia. The phase angle is positive at low frequencies, but is running behind at frequencies beyond 2 Hz due to inertia effects. Figure 8 reveals that the node numbered 3 also has the highest stiffness, as might be expected.

When dynamic and static stresses just balance at a certain moment, loss of contact will occur. This eventuates at 1.8 Hz, which seems very low. But in the literature even lower experimental values have been reported, see e.g. [17] and [19]. Hence it may be concluded that under dry contact conditions *separation may occur in many cases*. This implies a circumferentially nonuniform clearance geometry.

5 FLUID-STRUCTURE INTERACTION

If the shaft rotational speed is sufficiently high (a condition that it very easily met), clearances will develop. As the contact width b is very narrow, fluid flow resistance will be low at separated parts of the shaft/seal interface. In addition, various pumping mechanisms like in [1] may help to fill these clearances with fluid. Due to shaft vibrations or imperfections the seal will be forced to perform small-amplitude oscillations around a prestressed state, because the lip rebound is impeded by viscous and inertial rubber behaviour. As the clearance geometry is no longer parallel, entrainment and squeeze action may create a load carrying film.

Figure 9 shows two ways of fluid film formation due to dynamic excitation. If shaft and seal are *coaxial*, shaft irregularities like out-of-roundness (lobes) can cause separation. At very high speeds the radial motions of the lip will be minimal, and a circular seal shape will result. If shaft and seal centre do *not coincide*, dynamic shaft eccentricity (radial runout) will create a delay in the seal response, again causing separation. At very high speeds the radial seal motions will be minimal again.

Fluid film formation through entrainment action is depicted in Figure 10. A rigid sinusoidal indenter is forced into the viscoelastic solid, resulting in a high prestress. When it starts to move, it will impose a harmonic excitation on the seal. At a sufficiently large speed the entrainment action will boost the fluid pressure so that it can overcome the high prestress. Note that the lubrication mechanism is *macro*hydrodynamic.

To demonstrate the feasibility of viscoelasticity in rotary seal lubrication, a case

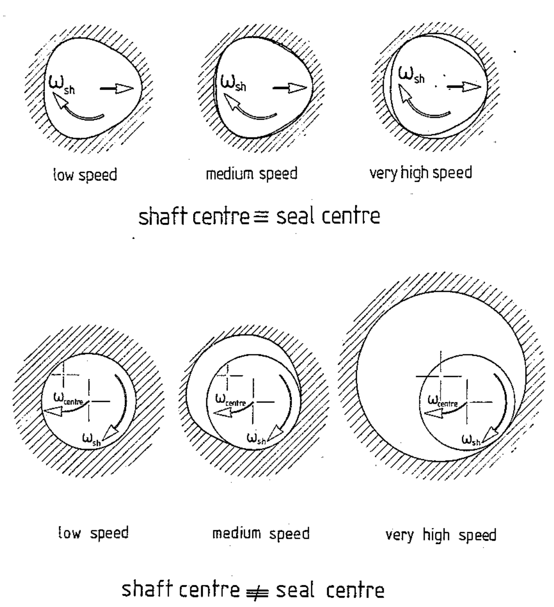

Fig.9 Effect of shaft out-of-roundness (top) and radial shaft runout (below) on lip geometry at several speeds.

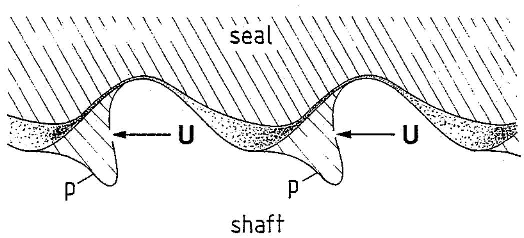

Fig.10 Pressure development in a viscoelastic/rigid contact through harmonic excitation.

of periodic excitation through radial shaft runout was chosen. It is assumed that whirl speed and shaft speed are equal, see Figure 11. Hence, squeeze effects can be eliminated by introducing an observer who rotates with shaft speed. Therefore, steady-state conditions remain. The eccentricity in this study is e=0.01 mm. The fluid is a Newtonian isoviscous fluid, and fully flooded lubrication prevails. The influence of fluid shear stresses on the seal deformation is omitted.

As the seal has an aspect ratio b/R of the order of 10^{-3}, it is the best example of a "short bearing" that is available. In this case, pressure and geometry are related by two equations, viz. the deformation equation (2) and the Reynolds equation, which reads as:

$$\frac{\partial}{\partial y}\left(\frac{h^3}{12\,\eta}\,\frac{\partial p}{\partial y}\right) = \frac{1}{2}\left(\omega_{sh} - 2\omega_{ref}\right)\frac{\partial h}{\partial \phi} \qquad (3.1)$$

where $\phi = \vartheta + \omega_{ref}.t$

$p = p_t$

$p(\phi,y{=}o) = p(\phi,y{=}b) = 0$

$p = p_{cav}$ where $p \leq p_{cav}$

$\omega_{ref} = \omega_{sh},\ \omega_{se} = 0$

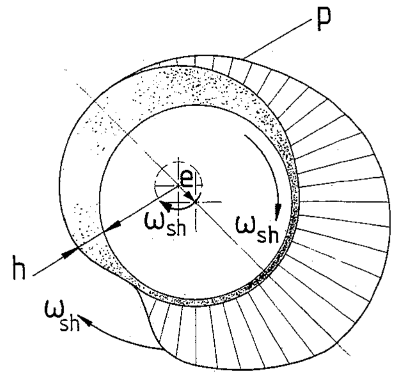

Fig.11 Case study of a shaft having radial runout.

The compatibility equation, tying together equations (2) and (3.1), is:

$$h(y,t) = e.\left\{ 1 + \cos(2\pi f_{sh}.t) \right\} + \delta(t,y) \qquad (3.2)$$

A cavitation pressure of 10 kPa below ambient was arbitrarily assumed. The viscosity in this study is constant, η = 0.1 Pa.s. Figure 9 suggests that for sufficiently high speeds the seal geometry resembles that of a plain journal bearing. Hence, it was tried in an earlier stage of this project to use a short bearing solution for Reynolds' equation, instead of equation (3.1). The problem then boils down to: given the eccentricity, find the unknown bearing clearance (and the film thickness). The solution turned out to be incompatible with equation (3.2), and large deviations from the circular geometry occurred at the minimum film thickness. As fluid pressures are high in this area, this idea was rejected.

It is appropriate to presume a parallel film in axial direction, see Figure 2, so $h \neq h(y)$. Hence, the fluid pressure distribution is parabolic over the contact width, which simplifies Reynolds' equation further. For the moment this is not to be considered as a serious drawback, since the aim is to show the *existence* of a fluid film. Because the profile in axial direction is known, the location where the film formation conditions are most severe was chosen, i.e. in the third node (see Figures 7 and 8). So the transfer function at $y{=}y_3$ has to be used in equation (2.2).

Now that two equations for unknowns h and p have been established, a solution simultaneously satisfying both equations is requested. From experience with elastohydrodynamics, parallel film regions can be anticipated, making Reynolds' equation very stiff. After careful scaling of the variables and discretization, a Newton-Raphson scheme was utilized to find solutions to equations (2) and (3). The method of solution is decribed in more detail in [8].

6 RESULTS AND DISCUSSION

For frequencies larger than 5 Hz full film lubrication conditions were only found if $p_s \leq 0.2$ MPa, whereas for a virgin seal $p_s(y_3)$ = 4.7 MPa. (see Figure 7). Three explanations can be offered:

(a) The lubrication mode is mixed film for a brand new seal.
(b) It is known that during running the contact width increases considerably, see a.o. [17] and [18]. This will dramatically lower the static stress component. Regretfully no geometry data on a run in seal were available.
(c) The problem is of numerical nature, and the solution scheme is not sufficiently robust. From elementary hydrodynamic lubrication it is known that, as long as the newtonian fluid behaviour does not break down (i.e. the viscosity remains constant whatever happens), the film thickness will always stay finite under finite loads. In consequence a solution should exist, and mixed film lubrication will never occur. Therefore the non-newtonian fluid behaviour, as suggested by, e.g., Jacobson [26], should be incorporated if film thicknesses are very small.

Under the conditions outlined above film profiles and corresponding pressure distributions were obtained as shown in Figure 12. Because full film conditions existed, the viscous shear induced friction in the fluid film could easily be calculated, see Figure 13. This Figure also presents some experimental results from [2]. At low frequency values (7.5 and 10 Hz) the numerical results are close to the experimental ones, but at 30 Hz and higher a clear discrepancy exists. This can be attributed to a temperature increase, which lowers the viscosity and changes the viscoelastic seal response. Table 1 summarizes some numerical data.

It can be noted that at low frequencies the minimum film thickness becomes extremely thin, and that the maximum value is much larger and typically of the order of 1÷10 μm.

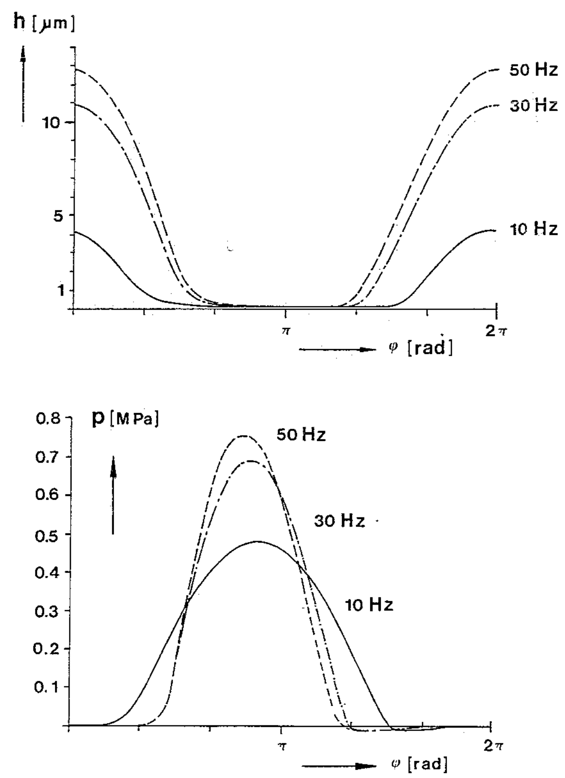

Fig.12 Film thickness and corresponding pressure distributions at $y=y_3$ for constant viscosity η = 0.1 Pa.s and variable frequency.

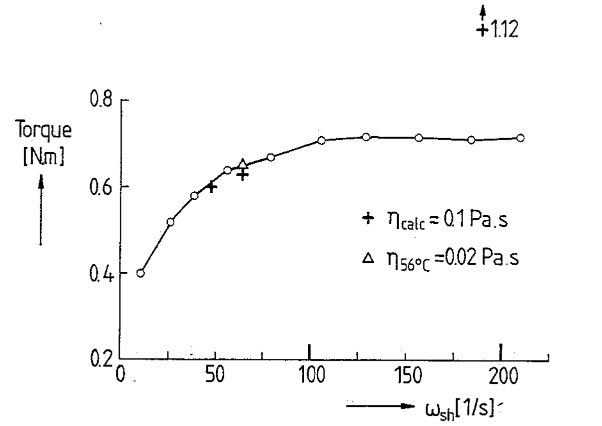

Fig.13 Frictional torque as a function of shaft frequency; o = measured, + = calculated, Δ = temperature-corrected value.

The minimum thickness values are on the low side, the maximum values are in agreement with experiments from the literature. A more realistic (higher) preload will reduce the minimum film thickness furthermore, although this should not be overemphasized, see e.g. [27]. Note that the fluid viscosity is high when compared to, e.g., automotive applications.

η = 0.1 Pa.s				
f [Hz]	h_{min} [μm]	h_{max} [μm]	p_{max} [MPa]	T [N.m]
7.5	0.077	2.40	0.434	0.601
10	0.088	4.18	0.479	0.637
30	0.121	11.0	0.690	1.12
50	0.147	12.8	0.762	1.40
100	0.224	14.3	0.806	1.70

Tabel 1: some numerical results for minimum and maximum film thickness, maximum fluid pressure, and frictional torque, at variable frequency.

In this study the VEHD lubrication mechanism is the only source of fluid film formation. At this moment it is the *only* concept that acknowledges the circumferentially nonuniform film, and it encompasses many seal design and application variables. However, VEHD theory presumably can not model active pumping action ([1], [2]). Probably a synergism of lubrication and sealing mechanisms occurs. Incorporation of effects as surface micro-irregularities (deformed by fluid pressure, [7]) and tangential surface deformation (through viscous fluid shear, [7] and [16]) should be considered.

As the viscoelastic (and inertial) seal response is essential for fluid film formation, this type of full film lubrication shall be called *viscoelastohydrodynamic (VEHD) lubrication.*

7 CONCLUSIONS

(1) Dynamic excitation can generate separation.
(2) Under dry frictionless contact conditions, the seal may separate from the shaft at low frequencies.
(3) Essential is a large increase in dynamic stiffness with frequency of the rubber.
(4) Separation introduces fluid filled clearances.
(5) This film geometry is circumferentially non-uniform, which is consistent with experimental findings in the literature.
(6) Entrainment effects at the inlet create a load carrying fluid film. Viscous rubber behaviour is essential, hence the lubrication concept is designated viscoelastohydrodynamic (VEHD) lubrication.
(7) Film thickness and friction results are in agreement with literature data.
(8) Because the mechanical behaviour of the rubber is of paramount importance, elastomer changes will affect fluid film formation. Consequently friction, wear and leakage are affected by the choice of the rubber.

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