Discussion of ASME paper No. 90-Trib-64 by A. Gabelli and G. Poll, entitled "Formation of lubricant film in rotary sealing contacts : Part 1 : lubricant film modelling"

DISCUSSION

H. van Leeuwen[1]

To explain the load-carrying capacity of the apparently parallel film in rotary shaft seals several lubrication mechanisms have been suggested. Surface roughness has been considered a primary source of fluid film formation since more than 25 years. The authors elaborate on this concept, by ingeniously tying together up to four models for dry/lubricated, and smooth/rough, and undeformed/deformed contact, see below. In addition, they have investigated the influence of solid viscoelasticity by studying the pressure response in a one degree-of-freedom model under periodic excitation. Fluid film formation in radial lip seals is indeed very complex. The authors have attempted to present a unified approach to this problem, and this is very much appreciated.

The main theme of this paper is surface roughness effects. We find this a very interesting contribution, because we have studied the macrohydrodynamic lubrication by viscoelastic effects, neglecting surface roughness, see Van Leeuwen and Stakenborg (1990).

What follows is first a commentary on the model employed by the authors, in order to verify the correct interpretation of the paper, and next a few questions.

As mentioned above, several existing models have been combined:

(*a*) The statistical Greenwood-Tripp (1970) model for dry rough contacting surfaces. This model for two rough surfaces is transformed into a contact between a smooth flat and a rough surface, having a cosine-like pattern in two dimensions, which is mainly described by a mean separation, an amplitude and 2 wavelengths. This implies a transformation from a stochastic model to a deterministic roughness model. It is understood that this is used to map the statistical data from the real surface texture into the deterministic model for lubrication studies.

(*b*) The deterministic Chittenden et al. (1986) model for isoviscous EHD lubrication of elliptical contacts. As the Greenwood-Tripp model yields data like mean contact area, and the mean value of the contact pressure, it is possible to calculate the elastohydrodynamic (EHD) minimum and central film thickness through the Chittenden et al. (1986) formulae by equating the mean contact force and the mean fluid force. This ingenious idea essentially is a Grubin/Ertel type of approach. The Chittenden et al. model allows for elastic deformations on a scale which is local to an asperity's summit.

(*c*) A model for the hydrodynamic (HD) entrainment action. It consists of a separate solution to the Reynolds equation for the deterministic roughness model. Continuity in the EHD (*b*) and HD (*c*) film thickness is more or less taken into account.

(*d*) A model for parallel smooth films under normal approach. Squeeze effects are superposed by adding the resulting pressures to the HD or EHD pressures.

(*e*) A model for a smooth viscoelastic flat, which is excited by a periodic displacement formed by a series of cosines. This simulates the viscoelastic behavior.

This discusser would like to touch upon the following issues:

(1) In the discusser's opinion it is not clear whether the experiments reported in section 2 suggest that a complete fluid film exists or not. From the model it can be concluded that

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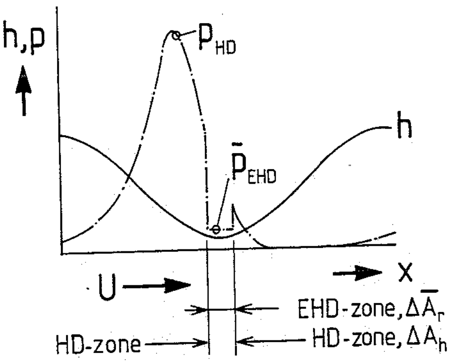

Fig. 9 Film thickness and corresponding fluid film pressure for one asperity. See text for explanation.

full film lubrication is assumed in the entire film. Do the authors have evidence to support this point of view?

(2) As equation (4.1) does not containt $\partial h/\partial t$, squeeze action cannot be simulated. Hence, surface roughness can only be attribued to the nonmoving surface. From a hydrodynamic point of view a moving rough surface yields a different pressure distribution than a stationary rough one. As the experimental results from Fig. 8 are obtained by rotating the seal (see Poll and Gabelli (1990)), which is considered the rough surface, this probably will not match the numerical results.

(3) A fluid film model that acknowledges the statistical nature of the surface texture has been defined by, a.o., Patir and Cheng (1978, 1979). Their model does not allow for asperity deformations. Asperity deformations are taken into acount in the paper discussed here. It therefore is a welome addition to the understanding of the lip seal behavior. These deformations are limited to the immediate neighborhood of the real contact area, since EHD (and Hertz) theory assumes local deformations. Hence, the deformation zone is much smaller than the dimensions of the deformed body (the asperity). Why should the deformations be restricted to a very small local area, under a global fluid film pressure?

The authors' model suggests that, in general, the pressures in the HD part and in the EHD part (consistent with the dry rough contact pressure) will show a discontinuity. This interpretation is depicted by Fig. 9, where $\bar{p}_{EHD}$, $\Delta\bar{A}_r$, and ΔA_h represent the average pressure, the average real contact area (both for one contact spot), and the area of hydrodynamic lubrication (for one asperity), respectively. Equation (6.3*b*) and Fig. 5 suggest that micro-HD pressures exceed micro-contact (or EHD) pressures by at least one order of magnitude. This would imply that the deformation zone encompasses the entire asperity. In conclusion, the lubrication of the asperities should be considered as EHD for the entire asperity.

(4) How have data to evaluate equation (6.2) and to find Fig. 6 been obtained? What are the operating conditions in Fig. 6? It seems to me that the excitation frequency is rather low, which implies a tremendous viscous action at realistic frequencies encountered in practice. Are the results from Fig. 6 fully converged? They do not seem to be cyclic.

(5) From extensive discussions in the 60's on bearing dynamics it is known that it is not allowed to apply the superposition principle for the pressure, if a Reynolds or Jakobssen-Floberg type of boundary condition is employed. This is a nonlinear condition. Pressure inducing effects as represented in equation (6.3) should then be treated *simultaneously*. In addition, the EHD (*b*) model and the HD (*c*) model have different boundary conditions. The Chittenden et al. boundaries are very close to the real contact, making the contribution of the micro-contact part in the load carrying capacity of minor

importance. Consequently, EHD pressures are much lower than HD pressures. They should be continuous.

(6) As the authors have found solutions for the film thickness, it should be relatively easy to evaluate equation (6.4) to obtain the global (total) friction force. This would be another check for the validation of the authors' model. It is suggested that the authors perform this evaluation.

(7) Which film thickness is meant in Figs. 7 and 8? I believe that in the case of Fig. 8B it is the minimum nominal thickness in time, $h_{0,\,min}$.

Would the authors please comment on these queries? As the authors state in their conclusion, the next step is the inclusion of macro-viscoelastic effects. It would be nice to combine the effects of viscoelasticity and surface roughness. I am looking forward to see more work in the future.

Additional References

van Leeuwen, H. J., and Stakenborg, M. J. L., 1990, "Visco-elastohydrodynamic (VEHD) Lubrication in Radial Lip Seals: Part 2—Fluid Film Formation," ASME JOURNAL OF TRIBOLOGY, Vol. 112, No. 4, pp. 584-592.

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A. O. Lebeck[2]

The authors have provided a detailed and valuable analysis of the role of microasperity hydrodynamic lubrication in lip seals. In my work referenced by the authors it is concluded that in mechanical seals such lubrication effects are not strong enough to provide significant load support in most cases. This is because the Youngs modulus is relatively high (carbon-graphite) and contact stresses (with high friction) occur at the low film thicknesses required to obtain significant load support (unit loads are higher than in lip seals as well).

Would the authors explain how the $h/\sigma = 2$ their hydrodynamic analysis is applied. This film thickness would suggest contact, so how is the film thickness computed to take account of the elastic deformation?

Can one conclude from the authors results that with new and very smooth rubber at the interface one would not develop load support due to such microhydrodynamic bearings and therefore have touching? Have such observations been made experimentally?

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Authors' Closure

The authors wish to thank Dr. Alan Lebeck and Mr. H. van Leeuwen for their careful reading of the paper and interesting discussion and would like to make the following remarks:

Further experimental observation and measurement of the lubricant film of lip seal contacts, see G. Poll et al. (1992), show that the lip seal works in mixed lubrication. In addition, it provides further evidence about the physical hypothesis of the lubrication model presented in the paper, i.e., stationary roughness.

Figure 9 of the discussion is hardly an adequate interpretation of the pressure shown in Fig. 5 of the paper. Indeed, the microcontact pressure of Fig. 5 is the load-carrying capacity due to contacting asperities in the sealing area, i.e., total microcontact load divided by the nominal area of the sealing contact. On the other hand, the average pressure of a contact spot is given by Eq. (5.1), and is the total microcontact load divided by the area of real contact of the seal. Furthermore the microcontact pressure should have been superposed onto the hydrodynamic pressure which is seen by the asperities as a local ambient over-pressure.

The data used in Eq. (6.2) were experimentally measured on a rubber lip seal. The results of Fig. 6 show the transient to reach the steady state condition.

The present study takes account of the squeeze film effects by superposing a squeeze film term onto the sum of the hydrodynamic and apparent microcontact pressure, see Eq. 6.3. However, in the presence of cavitation, squeeze film effects would alter the cavitation pattern and influence the hydrodynamic pressure distribution. On the other hand, due to the average nature of this lubrication model and the periodic character of the pressure fluctuations, the load support terms are thought to be sufficiently decoupled, at least in their mean value, so that superposition of the different terms would apply.

Figures 7 and 8 show the minimum nominal film thickness of a time cycle. Comparison of measured and calculated contact pressure distribution and friction of a lip seal contact can be found in A. Gabelli et al. (1992) and G. Poll et al. (1992).

Finally the authors would like to answer Dr. Alan Lebeck's comments. The hydrodynamic analysis applied in this paper is based on a gap with rigid surfaces and morphological characteristics equivalent to those of the sealing contact. This has proven to be a reasonable assumption for the present model. As the elastic deformation is restricted to the tips of the highest asperities, it will affect the average shape of the sealing gap only in a limited way. However, this assumption can easily be relaxed and accommodated into the model.

Concerning the relationship between roughness and film formation of lip seals it is the authors' practical experience that very smooth, mirror-like, surfaces are detrimental to the ability of lip seals to form a lubricant film. However, this does not preclude that in such circumstances other phenomena will arise and play a significant role in the lubrication of the contact.

Additional References

Gabelli, A., Ponson, F., and Poll, G., 1992, "Computation and Measurement of the Sealing Contact Stress and Its Role in Rotary Lip Seal Design," 13th International Conference on Fluid Sealing. Brugge, Belgium.

Poll, G., Gabelli, A., Binnington, P. G., and Qu, J., 1992, "Dynamic Mapping of Rotary Lip Seal Lubricant Film by Fluorescent Image Processing," 13th International Conference on Fluid Sealing, Brugge, Belgium.