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.

Patir, N., and Cheng, H. S., 1978, "An Average Flow Model for Determining Effects of Three-Dimensional Roughness on Partial Hydrodynamic Lubrication," ASME JOURNAL OF LUBRICATION TECHNOLOGY, Vol. 100, No. 1, pp. 12–17.

Patir, N., and Cheng, H. S., 1979, "Effect on Surface Roughness Orientation on the Central Film Thickness in EHD Contacts," *Elastohydrodynamics and Related Topics*, Mechanical Engineering Publications, London, pp. 15–21.

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?

[2] Mechanical Seal Technology, Inc., Albuquerque, NM 87109.

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.