ratio, except that for pure rolling the amplitude is somewhat smaller. But the present results, exemplified by Figs. 7 to 11, seem to demolish our theory completely. As we said above, we are dismayed.

Additional References

A Greenwood, J. A., and Morales-Espejel, G. E., "The Behavior of Transverse Roughness in EHL Contacts," to appear in *I.Mech.E. Journal of Engineering Tribology*, Apr. 1994.

B Ai, X., and Cheng, H. S., "A Fast Model for Pressure Profile in Rough EHL Line Contacts," ASME JOURNAL OF TRIBOLOGY, Vol. 115, 1993, pp. 460–465.

C Venner, C. H., and Lubrecht, A. A., 1992, "Transient Analysis of Surface Features in an EHL Line Contact in the Case of Sliding," submitted to ASME JOURNAL OF TRIBOLOGY.

D Venner, C. H., "Multilevel Solution of the EHL Line and Point Contact Problems," Ph.D. thesis, University of Twente, Netherlands, 1991.

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The authors present transient results of EHL between a perfectly smooth surface and a rough surface of random roughness. This is a significant step forward from earlier steady-state analyses with a moving smooth surface and a stationary rough surface. Figures 7 to 11 show that abrupt pressure rippling is generated (except in pure rolling), and the magnitudes of the pressure ripples increase with sliding. This discusser speculates that different micro-EHL results might have been obtained had the rheological model included the shear-thinning characteristics of the lubricant. A brief analysis follows.

Consider two locations inside the Hertzian region, x_1 and x_2 which, along with the two surface segments in between, define a control volume. Neglect lubricant compressibility, flow continuity in this volume gives:

$$\left(uh-\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)\bigg|_{x_1}-\left(uh-\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)\bigg|_{x_2}=\int_{x_1}^{x_2}\frac{\partial h}{\partial t}\,dx \quad \text{(D1)}$$

The first term on the left is the rate of lubricant inflow and the second term outflow. The right hand side is the rate of accumulation of the lubricant in the control volume. Rearrange Eq. (D1) as:

$$u(h_{x_1}-h_{x_2})-\left(\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)\bigg|_{x_1}-\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)\bigg|_{x_2}\right)=\int_{x_2}^{x_2}\frac{\partial h}{\partial t}\,dx \quad \text{(D2)}$$

Let h_{x_1} be a local maximum and h_{x_2} the adjacent local minimum downstream. If the rough surface is stationary while the smooth surface in motion (i.e., steady-state sliding, SR = −2), the right-hand side of Eq. (D2) is zero. Then the second term on the left-hand side must be large enough to balance the first term. Since h^3 is small and η large in the Hertzian region, $|\partial p/\partial x|$ needs to be large to maintain flow continuity, generating sharp pressure ripples. The large pressure ripples deform the roughness in such a way to reduce the difference between h_{x_1} and h_{x_2}. In the limit, $h_{x_1}\approx h_{x_2}$, or the roughness is flattened out (Fig. 7). Consider next the case of SR = −1 (Fig. 8) where the rough surface is also in motion but moves more slowly than the smooth surface. The right-hand side of Eq. (D2) is positive but can be shown to be smaller than the first term on the left-hand side. A smaller $|\partial p/\partial x|$ is needed in this case than in the zero-right-hand-side case to maintain flow continuity. In the case of pure rolling (Fig. 9), the right-hand side of Eq. (D2) is larger than the previous case of SR = −1 and is about equal to the first term on the left-hand side (authors' results suggest exact equal). Consequently, there is little need to generate pressure ripples to maintain flow continuity. As SR further increases, the right hand side of Eq. (D2) becomes greater than the first term on the left, pressure ripples are then generated to balance the difference between these two terms, the larger the difference, the larger the pressure ripples.

If the shear-thinning behavior of the lubricant is modeled, another competing factor enters the system which may significantly change the pressure-ripple generation. Consider again the case of steady-state sliding (Fig. 7). With SR = −2 (or SR = 2), the lubricant exhibits the strongest shear thinning, which substantially reduces the effective viscosity of the lubricant. For the given problem (i.e., Fig. 7), the effective viscosity with shear thinning (by Eyring viscous law) is about two to three orders of magnitude smaller than the (two-slope-law) viscosity. Therefore, a much smaller $|\partial p/\partial x|$ is needed to generate significant pressure-induced flow of lubricant to maintain flow continuity. Smaller $|\partial p/\partial x|$ means smaller pressure ripples and thus smaller roughness deformation. The roughness does not have to be flattened out and flow continuity can still be satisfied. Next, consider the case of SR = −1 again (Fig. 8). The shear thinning is weaker and therefore the effective viscosity is larger in this case than in the case of SR = −2. Whether larger or smaller pressure ripples will be generated depends on the changes in the following two competing mechanisms as SR varies. One is the change in the difference between the first term on the left-hand side of Eq. (D2) and the right-hand side. The other is the change in the effective viscosity due to shear thinning. The maximum pressure rippling may be generated at a slide-to-roll ratio at which the shear thinning is weak while the different between the first term on the left-hand side of Eq. (D2) and the right-hand side is sufficiently large. Most important of all is that, in any case, the magnitudes of pressure ripples seem to be limited by one of these two competing mechanisms.

Since EHL lubricants can exhibit strong shear-thinning behavior which may substantially affect micro-EHL conditions, it is important to incorporate this behavior into the rheological model in micro-EHL analyses.

Authors' Closure

The authors are grateful for the constructive discussions by Drs. Greenwood and Morales-Espejel and Dr. Chang. We would like to respond to the discussions respectively.

Response to Drs. Greenwood and Morales-Espejel

As mentioned by the discusser, the pressure prediction models developed by the two groups are not in a very good agreement, at least for Eqs. (D-1) and (D-2) in their appearances; the former is strictly linear in A, the amplitude of roughness component, while the latter shows some nonlinearity in A. However, the difference could be well understood by examining the basis upon which the models are developed.

We all agree that for stationary roughness under heavily loaded conditions the roughness amplitude is greatly diminished, particularly for those low frequency components, and that the Reynolds equation shows a great deal of linearity which indicates the possible applicability of superposition. However, it by no means implies that the EHL is a strict linear system. Since Eq. (D-2) is regressed directly from numerical simulation results for a relatively wide range of operation conditions, the nonlinearity in A is expected. It reflects the nonlinearity of EHL system.

Obviously, the exponent of A in Eq. (D-2) depends on the range of load used in the regression. For Eq. (D-2) the regression covered 41 sets of simulation results with the load ranges form $W=4.33\times10^{-4}$ to 2.17×10^{-5}. If only high load ($W=4.33\times10^{-4}$) results are used, regression yields

$$\Delta P=C\bullet A^{0.955}L^{-1.01}$$

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