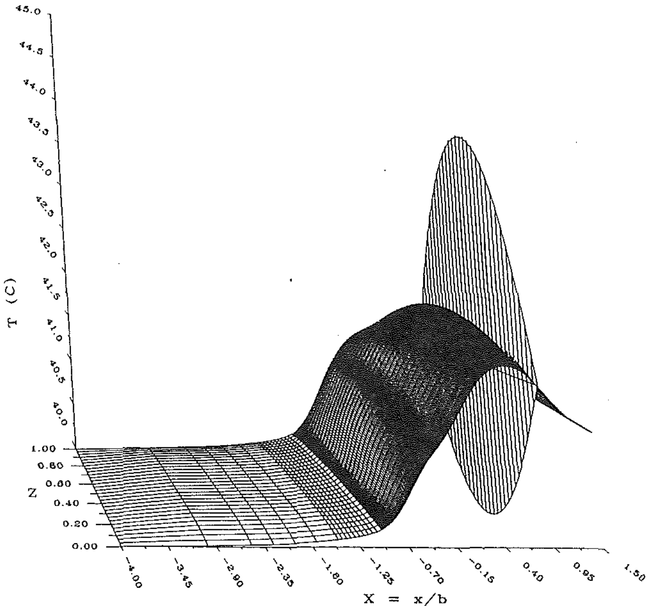

Fig. 7(*a*) Temperature distribution for oil-1 at A = 0.1

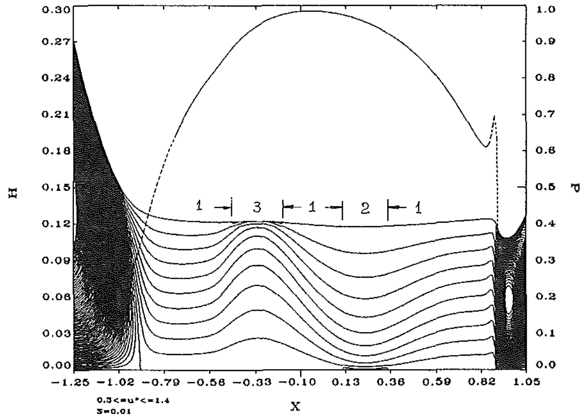

Fig. 7(*b*) Velocity contours indicating flow-zone distribution for oil-1 at A = 0.1 (1 designates flow zone with velocity boundaries, 2 designates flow zone with a stress boundary at $Z=0$, 3 designates flow zone with a stress boundary at $Z=1$)

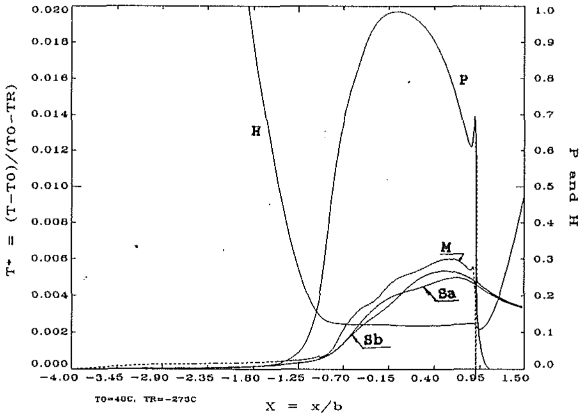

Fig. 7(*c*) Surface and midfilm temperatures with film shape and pressure profile for oil-1 at A = 0.1. (Sa designates surface temperatures at $Z=0$, M designates midfilm temperatures at $Z=0.5$, Sb designates surface temperatures at $Z=1$)

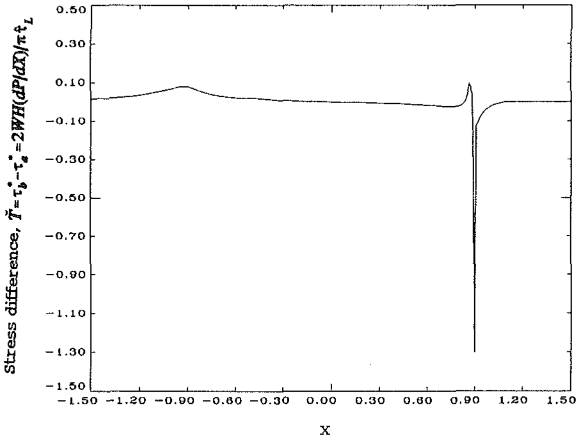

Fig. 7(*d*) Stress-difference profile for oil-1 at A = 0.1

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DISCUSSION

L. Chang[1]

The authors present a new method to calculate the surface temperatures in an EHL conjunction. This method appears to be applicable to problems with both high and low surface velocities and is actually used by the authors in the simulation of thermal EHL under simple sliding conditions. This discusser would appreciate the authors' comments to the following questions and/or statements:

(1) The surface temperature is calculated by carrying the time integral in Eq. (42) through a finite duration of $t=(x_{N_{\max}}-x_1)/|u_a|$ or $t=(x_{N_{\max}}-x_1)/|u_b|$, whichever is shorter. Is there any physical explanation behind the selection of this time duration for the integration?

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(2) For a given entraining velocity, the integration time determined above is shortest when the slide-to-roll ratio is equal to 2.0, at which one of the surfaces is stationary. It seems to this discusser that the integration needs to be carried out for a longer time for the slower moving surface and it should be the longest when the surface is stationary.

(3) If the method is valid and if the integration time is sufficiently long, the surface temperature calculated using this method should be consistent with those calculated using the conventional flash-temperature method for problems where the conventional method is applicable. Have the authors quantitatively evaluated this new method?

Authors' Closure

The authors wish to thank Dr. Chang for his interesting questions and valuable comments.

The authors agree that the time integral in Eq. (42) needs to be carried out for a longer period of time for the slower moving solid surface if the "pseudo-stationary state solutions" are to be attempted. The "pseudo-stationary state" is referred to as the state at which the repetition of the cyclic phenomenon in the EHL conjunction becomes unchanged. The "quasi-stationary state" mentioned before was used to describe the state when the time integral in Eq. (42) was evaluated at infinity. It seems to the authors that carrying out the time integrals in Eq. (42) to $t_a = (x_{N_{\max}} - x_1)/|u_a|$ and $t_b = (x_{N_{\max}})/|u_b|$ for the slower and the faster surfaces respectively might give a good approximation for the pseudo-stationary state solution when the surface cooling effects are sufficiently significant. The pseudo-stationary state solutions are interesting and will be presented in later papers.

The same amount of integration time used for both surfaces in this paper was to compare the thermal effects for various slide-to-roll ratios upon completion of the initial transient cycle of the faster moving surface. No other physical considerations were made.

The authors agree that the new model and the conventional model should agree closely with each other when the entraining velocity is sufficiently high, the slide-to-roll ratio is low enough, and the pseudo-stationary state solutions are of interest. Although no results are available for direct comparison between these two models yet, some interface temperature results obtained by using the conventional interface temperature model for Ree-Eyring fluids show good agreement with that obtained by using the new model for circular non-Newtonian lubricants in high velocity and pure rolling conditions. For example, in Fig. A the surface temperature rise provided by Salehizadeh and Saka (1991) agrees very well with the results for oil 2 at $p_H = 1$ GPa, $\bar{u}_{ab} = 10$m/s, $T_0 = 60°$C, and $A = 0$ obtained by methods proposed in this paper. The results of the conventional model with a higher peak surface temperature at the beginning of Hertzian contact zone and lower surface temperatures at both inlet and Hertzian zones are mainly due to neglecting heat diffusion in the direction of entraining motion.

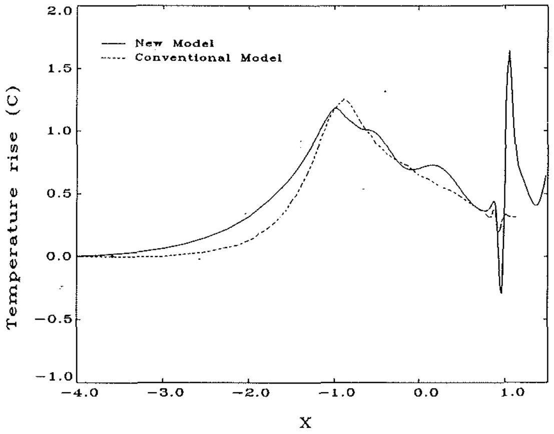

Fig. A Comparing surface temperature rises for oil 2 at $p_H = 1$ GPa, $\bar{u}_{ab} = 10$ m/s, $T_0 = 60°$C, and $A = 0$ obtained by using the new interface temperature model with the results obtained by using the conventional model from Salehizadeh and Saka (1991)

Additional Reference

Salehizadeh H., and N. Saka, 1991, "Thermal Non-Newtonian Elastohydrodynamic Lubrication of Rolling Line Contacts." ASME Journal of Tribology, Vol. 113, pp. 481-491.