NASA TECHNICAL NOTE

NASA TN D-7825

NASA TN D-7825

CORRELATION OF ELASTOHYDRODYNAMIC FILM THICKNESS MEASUREMENTS FOR FLUOROCARBON, TYPE II ESTER, AND POLYPHENYL ETHER LUBRICANTS

by Stuart H. Loewenthal, Richard J. Parker, and Erwin V. Zaretsky

Lewis Research Center and
U.S. Army Air Mobility R&D Laboratory
Cleveland, Ohio 44135

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • NOVEMBER 1974

1. Report No. NASA TN D-7825	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle CORRELATION OF ELASTOHYDRODYNAMIC FILM THICKNESS MEASUREMENTS FOR FLUOROCARBON, TYPE II ESTER, AND POLYPHENYL ETHER LUBRICANTS		5. Report Date November 1974 6. Performing Organization Code
7. Author(s) Stuart H. Loewenthal, Richard J. Parker, and Erwin V. Zaretsky		8. Performing Organization Report No. E-7954 10. Work Unit No. 501-24
9. Performing Organization Name and Address NASA Lewis Research Center and U.S. Army Air Mobility R&D Laboratory Cleveland, Ohio 44135		11. Contract or Grant No. 13. Type of Report and Period Covered Technical Note
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

A minimum films thickness correlation applicable to heavily loaded elastohydrodynamic (EHD) contacts was formulated from experimental data obtained by an X-ray transmission technique. The correlation, based on data generated with fluorocarbon, type II ester, and polyphenyl ether lubricants, extends a previous analysis developed from data for a synthetic paraffinic oil. The resulting correlation represents the data of the four lubricants reasonably well over a large range of operating conditions. Contained within the derived relation is a factor to account for the high-load dependence displayed by the measurements beyond that which is provided for by theory. Thermal corrections applied to a commonly used film thickness formula showed little improvement to the general disagreement that exists between theory and test. Choice of contact geometry and material are judged to have a relatively mild influence on the form of the semi-empirical model.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Bearings; Lubrication; Elastohydrodynamic lubrication; Rolling-element bearings; Ball bearings; High temperature lubrication; Film thickness; Synthetic lubricants	Unclassified - unlimited Category 15

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	28	$3.25

* For sale by the National Technical Information Service, Springfield, Virginia 22151

CORRELATION OF ELASTOHYDRODYNAMIC FILM THICKNESS MEASUREMENTS FOR FLUOROCARBON, TYPE II ESTER, AND POLYPHENYL ETHER LUBRICANTS

by Stuart H. Loewenthal, Richard J. Parker, and Erwin V. Zaretsky

Lewis Research Center and

U. S. Army Air Mobility R&D Laboratory

SUMMARY

A minimum film thickness correlation for heavily loaded elastohydrodynamic (EHD) contacts was formulated from X-ray film thickness measurements made with fluorocarbon, type II ester, polyphenyl ether, and synthetic paraffinic lubricants. The semi-empirical model was developed from test data which covered a wide and practical range of operating conditions. Maximum Hertz stresses ranged from 1.04×10^9 to 2.42×10^9 newtons per square meter (150 000 to 350 000 psi), disk temperatures from 339 to 589 K (150° to 600° F), and mean surface rolling speeds from 9.4 to 37.6 meters per second (370 to 1480 in./sec).

A comparison of the predicted values of the minimum film thickness from the deduced EHD formula with the X-ray results revealed that the film thickness correlation, despite its simplified form, represented the measured data reasonably well throughout the full range of test conditions.

At elevated temperatures the fluorocarbon appeared to be the best film former followed by the synthetic paraffinic, polyphenyl ether, and type II ester fluids in diminishing order of film forming capability.

Comparisons were made with a representative isothermal EHD analysis with and without thermal corrections for shear heating at the inlet of the contact zone. This comparison reconfirmed that theoretical consideration of heating effects would not appreciably influence the weak relation of isothermal film thickness to contact stress nor significantly diminish the magnitude discrepancy between theory and the X-ray data except at the more severe heating conditions.

The semi-empirical film thickness equation contains a high-contact stress factor φ_s to correct classical film thickness formulas for the higher sensitivity of minimum film thickness to contact stress exhibited by the X-ray test data. This factor indicates that this observed higher than theoretically anticipated sensitivity is responsible for the predicted film thicknesses being up to twice as large as those actually measured.

It is anticipated that the semi-empirical film thickness formula can be used with reasonable certainty for many rolling-element bearing and gear systems. These systems include principally those composed of steel, employing the lubricants studied herein, and those whose contact geometry approximates that upon which the model is based.

INTRODUCTION

The importance of maintaining a sufficient elastohydrodynamic (EHD) film thickness between dynamically contacting machine elements has in recent years been more fully appreciated. The prediction of EHD film thickness has been the focal point of many theoretical and experimental investigations; it has been summarized well in references 1 and 2.

The ratio of EHD minimum film thickness to composite surface roughness of the mating contact surfaces has become an acceptable indicator of the effectiveness of the lubricant film within the rolling-element contact zone. It has been shown experimentally that this ratio influences the fatigue life of rolling-element bearings (refs. 3 and 4). Predetermination of this lubricant parameter with an accurate prediction of minimum film thickness is of value to the designer in obtaining more realistic estimates of rolling-element fatigue life (ref. 5).

The bulk of the experimental work conducted in elastohydrodynamic lubrication has been confined to conditions of moderate speeds up to 25.4 meters per second (1000 in./sec) and moderate loads which resulted in maximum Hertz stresses to 1.24×10^9 newtons per square meter (N/m^2) (180 000 psi) (refs. 6 to 9). The research of references 10 and 11 has extended the EHD film thickness measurements to maximum Hertz stresses of 2.42×10^9 N/m^2 (350 000 psi) which includes the design operating range of most machine components such as bearings and gears. These data were obtained on a rolling-disk machine using an X-ray transmission technique to measure minimum film thickness. The film thickness measurements showed good qualitative agreement with full scale bearing test results (ref. 12); that is, very low film thicknesses were measured at conditions similar to those where the bearings suffered surface damage.

Results obtained by previous investigators showed reasonably good correlation at moderate speeds and loads between elastohydrodynamic theory and film thickness measurement (refs. 6, 8, and 9). In contrast, however, the data of references 10 and 11 showed a marked deviation between predicted and experimental values of film thickness. In particular, at high contact stresses (i.e., maximum Hertz stresses greater than 1.38×10^9 N/m^2 (200 000 psi)), the sensitivity of the film thickness to load as determined experimentally is far greater than that predicted by classical EHD theory of references 13 and 14.

Several attempts have been made to resolve the apparent discrepancy between theory and experiment. A critical examination of the X-ray technique itself was made for possible load dependent experimental errors (ref. 15). However, no experimental factors were uncovered which could seriously alter the accuracy of the X-ray measurements. On the theoretical side, the influence of several possible rheological factors has been investigated. These factors were the effects of a non-Newtonian lubricant of the Ree-Eyring form (ref. 16), the effects of heating at the inlet of the contact region (ref. 17), and the effects of a reduced lubricant viscosity-pressure dependence using both a composite exponential model (ref. 18) and a power-law model (ref. 19).

Each of the previous modifications to elastohydrodynamic theory has succeeded somewhat in improving the agreement between theory and experimental data within the heavy load regime. However, the resulting predicted values of film thickness differed little in magnitude from those computed using classical EHD theory. Furthermore, the modified theories do not sufficiently account for the observed high film thicknesses - load dependence to allow accurate predictions of film thickness under realistic operating conditions.

The experimental data of references 20 and 21 also show a film thickness sensitivity to stress greater than theoretical for maximum Hertz stresses greater than about 1.04×10^9 N/m^2 (150 000 psi). These data, obtained by an optical interferometry technique with sliding point contacts, tend to support the measurements obtained by the X-ray technique of references 10 and 11.

It was the objective of the work reported herein to generalize the empirical film thickness model developed in reference 22 through an analysis of the experimental data of references 10 and 11 and to compare this derived relation with that of classical elastohydrodynamic theory.

SYMBOLS

a	minor semi-axis of Hertzian contact, m (in.)
B	constant, eq. (6)
b	major semi-axis of Hertzian contact, m (in.)
$C_{i,j}$	coefficient, eq. (1)
E_1, E_2	modulus of elasticity of elements 1 and 2, N/m^2 (psi)
E'	$\left(\frac{1-\nu_1^2}{\pi E_1}+\frac{1-\nu_2^2}{\pi E_2}\right)^{-1}$, N/m^2 (psi)
$f(P_{Hz})_j$	film thickness - stress function, eq. (2)
G	dimensionless speed factor, $(u/N)^{0.62}$
$\overline{H}_{min}$	nondimensional minimum film thickness, h_{min}/R'
H^*	lubricant film thickness factor, $\overline{H}_{min}/G$
h_{min}	minimum film thickness, m (in.)

i	experimental maximum Hertz stress subscript
j	test lubricant subscript
K_j, K_j^*, k_j	lubricant coefficients in empirical film thickness formula
N	constant, 9.4 m/sec (370 in./sec)
$n_{i,j}$	nondimensional speed - viscosity parameter exponent, eq. (1)
$\hat{n}_j$	mean value of $n_{i,j}$, eq. (2)
$\overline{P}_{Hz}$	nondimensional stress parameter, p_{Hz}/E'
p_{Hz}	maximum Hertz stress, N/m^2 (psi)
R_1, R_2	radius of elements 1 and 2 in rolling direction, m (in.)
R'	equivalent radius, $\left(\frac{1}{R_1} + \frac{1}{R_2}\right)^{-1}$, m (in.)
T_o	disk temperature, K (^oF)
$\overline{U}$	nondimensional speed - viscosity parameter, $\mu_o u/E'R'$
u	mean surface velocity, $1/2(u_1 + u_2)$, m/sec (in./sec)
u_1, u_2	surface velocities of elements 1 and 2, m/sec (in./sec)
α	pressure-viscosity coefficient, m^2/N (psi^{-1})
μ_o	inlet absolute viscosity, $N\text{-}sec/m^2$, ($lb\text{-}sec/in.^2$)
$\nu_{1,2}$	Poisson's ratio of elements 1 and 2
φ_s	high contact stress factor, eqs. (6) and (8)
ψ_s	film thickness - stress function, eq. (4)

X-RAY FILM THICKNESS TEST DATA

The film thickness model presented herein was developed from the film thickness data (refs. 10 and 11) obtained in an X-ray rolling-disk machine. The four lubricants studied were a type II ester, a fluorocarbon, a polyphenyl ether, and a synthetic paraffinic oil. These lubricants are considered to be promising candidates for high temperature bearing application (ref. 12). Their properties are summarized in table I.

Briefly, the method of measuring film thickness utilizing the X-ray transmission technique involves flooding the contact region between the two test disks with X-rays. The rate of X-ray transmission which penetrates the contact within a narrow bandwidth is determined by a scintillation counter. Since the greatest constriction occurs at the trailing edge of the contact, the X-ray count becomes a measure of the lubricant's minimum film thickness. The side lobes of the contact are avoided by the beam and thus do not influence the reading.

The range of test conditions reported in references 10 and 11 include disk temperatures from 339 to 589 K (150° to 600° F), surface rolling speeds from 9.4 to 37.6 meters per second (370 to 1480 in./sec), which correspond to disk rotational speeds from 5000 to 20 000 rpm, and maximum Hertz stresses from 1.04×10^{9} to 2.42×10^{9} N/m^2 (150 000 to 350 000 psi). Two AISI M-50 steel disks each with a rolling radius of 1.83 centimeters (0.72 in.) and a surface finish of 2.5×10^{6} to 5.0×10^{6} centimeters (1 to 2 μin.) rms were used as the test specimens.

Both crowned disks (with a crown radius of 27.9 cm (11 in.)) and crowned-cone disks (with a cone angle of 10°, which introduce a small spin component in the contact) were tested, and no significant differences were reported between the two sets of film thickness data (refs. 10 and 11). All the data reported herein with the exception of the test data for the polyphenyl ether lubricant were generated with the crowned-cone test disks appearing in figure 1.

FORMULATION OF EHD FILM THICKNESS CORRELATION

The effect of surface speed u and lubricant absolute viscosity μ on the measured value of minimum film thickness h_{min} at each experimental contact stress level p_{Hz} for the four test lubricants is shown in figure 2. The summary plot for the synthetic paraffinic lubricant in figure 2(d) has been presented before in reference 22. In figure 2 the log of the dimensionless film thickness parameter $\overline{H}_{min} = h_{min}/R'$ is plotted against the log of the dimensionless speed-viscosity parameter $\overline{U} = (\mu_0 u/E'R')$. Because of the linear relation exhibited by the test data on this logarithmic plot as determined by a linear regression analysis, there exists the simple power relation

$$(\overline{H}_{min})_{i,j} = C_{i,j}\overline{U}^{n_{i,j}} \tag{1}$$

where the subscript $i = (1 \rightarrow 5)$ designates one of the five experimental maximum Hertz stress levels, the subscript $j = (1 \rightarrow 4)$ designates one of the four test fluids, and $\overline{U}$ ranges from the lowest to highest experimental value.

When comparing the plots for each lubricant in figure 2 it is apparent that both the polyphenyl ether and synthetic paraffinic film thickness data, unlike that for the fluorocarbon or type II ester fluids, display a somewhat increased sensitivity to rolling surface speed and lubricant viscosity with increasing contact stress. This is reflected by the increase in the slope of lines with higher contact stress appearing in figures 2(c) and (d) or more explicitly by the variation of exponent $n_{i,j}$ in equation (1) which is tabulated along with coefficient $C_{i,j}$ in table II.

This variation of the dependence of h_{min} to $\overline{U}$ with increasing P_{Hz} is not anticipated from theoretical considerations. In view of the overall consistency of the test data, it is not apparent why this anomaly was observed for just two of the four test lubricants. However, the effect of neglecting such an exponential variation in the case of the synthetic paraffinic and polyphenyl ether fluids will have a minimum effect on the resulting generalized film thickness expression as will be shown later. Accordingly, selecting mean values of exponents $n_{i,1}$ and $n_{i,4}$ results in the values of exponent $\hat{n}_j$ listed in table III. These values have been determined for each of the test lubricants from the X-ray test data.

Equation (1) can now be written in the following form:

$$(\overline{H}_{min})_j = \overline{U}^{\hat{n}_j} f(p_{Hz})_j \tag{2}$$

where the presently unknown continuous function $f(p_{Hz})_j$ has been introduced to describe the dependence of minimum film thickness on the peak contact pressure for each lubricant. The main objective in this approach is to separate the effects of maximum Hertz stress (contact pressure) on the value of film thickness from those effects contributed by surface speed and lubricant viscosity. Having thus isolated the effects of contact pressure, there remains the task of representing the influence of contact pressure on film thickness by a single mathematical expression for all test fluids, that is, formulating $f(p_{Hz})$.

By combining equations (1) and (2) an expression can be written for $f(p_{Hz})_j$ where

$$f(p_{Hz})_j = C_{i,j}\overline{U}^{(n_{i,j}-\hat{n}_j)} \quad \text{or} \quad \left(\frac{\overline{H}_{min}}{\overline{U}^{\hat{n}_j}}\right)_j \tag{3}$$

It is now possible to evaluate $f(p_{Hz})_j$ from the experimentally deduced values of the expression appearing on the right side of equation (3).

It is evident from equation (3) and the previous discussion that $f(p_{Hz})_j$ is an explicit function of $\overline{U}$. The extent of this dependence is illustrated in figure 3 which shows the maximum variation of $f(p_{Hz})$ with $\overline{P}_{Hz}$ evaluated at both the maximum and minimum experimental value of $\overline{U}$ for each of the test fluids. In the case of the fluorocarbon and type II ester fluids, the effect of $\overline{U}$ on the $f(p_{Hz})_j$ term in equation (2) is negligible as evidenced by the nearly coincident lines in figures 3(a) and (b). With regard to the polyphenyl ether and synthetic paraffinic test fluids, the sensitivity of minimum film thickness to maximum Hertz stress does vary somewhat with changes in operating surface speed and lubricant viscosity as evidenced by figures 3(c) and (d). This variation is anticipated since it stems from the variation of the $\overline{U}$ exponent $n_{i,j}$ with p_{Hz}, which was discussed earlier. The influence of this variation on the accuracy of the resulting film thickness expression does not warrant the introduction of an additional $\overline{U}$ dependent factor into the finalized correlation as will be shown later.

Quite satisfactory results can be obtained by evaluating the $f(p_{Hz})$ in equation (3) at the mean experimental values of $\overline{U}$. This has been accomplished in figure 4 where the relation of $f(p_{Hz})_j$, which has been evaluated at $\overline{U}_{mean}$, to $\overline{P}_{Hz}$ is shown. It is apparent that there is a great similarity in the shape of the curves appearing in this figure. That is to say, the effect of maximum Hertz stress on the ratio of $(\overline{H}_{min})_j$ to $\overline{U}^{\hat{n}_j}$ is nearly the same, apart from some constant multiplier, say k_j, for all the fluids tested. Thus, the effect of contact pressure on the test lubricants minimum film thickness can be represented by a single generalized function ψ_s by the relation

$$\psi_s = \frac{f(p_{Hz})_j}{k_j} \tag{4}$$

where constant k_j is an arbitrary lubricant constant used to normalize the value of $f(p_{Hz})_j$ at $\overline{P}_{Hz} = 3.09\times10^{-3}$ - that is, at $p_{Hz} = 1.04\times10^{9}$ N/m^2 (150 000 psi). Table III lists the respective k_j constants for each lubricant.

Having a function ψ_s which satisfactorily describes the effects of p_{Hz} on h_{min}, the generalized film thickness expression can be completed by substituting equation (4) into equation (2):

$$(\overline{H}_{min})_j = k_j\overline{U}^{\hat{n}_j}\psi_s \tag{5}$$

This expression is, in itself, a film thickness correlation which can be used to satisfactorily forecast minimum film thickness at high contact stress levels. However, as a matter of user convenience, equation (5) will be altered slightly to a more universal form.

FORMULATION OF EHD HIGH-CONTACT-STRESS FACTOR

As previously discussed, film thickness values forecasted by currently accepted EHD theory have shown reasonably good agreement with experimental data for maximum

Hertz pressures less than approximately 1.04×10^9 N/m^2 (150 000 psi). Above this stress level the conventional theory seriously overestimates the extent of the film generated by the lubricant as evidenced by test data (refs. 10 and 11). Thus, it is most desirable to introduce some factor to adjust current film thickness formulas for the deviation between theory and experiment at high applied loads.

It is generally recognized that film thickness is only moderately dependent on contact stress at the lower stress levels. Typically, for line contact the film thickness is proportional to maximum Hertz stress to the -0.22 power, that is, $\overline{H} \propto (\overline{P}_{Hz})^{-0.22}$, where the stress parameter exponent arbitrarily selected here comes from the isothermal theory of Cheng (ref. 23). This proportionality can be introduced into the film thickness relation shown in equation (5) by simply defining a new factor φ_s such that

$$\varphi_s = B \frac{\psi_s}{(\overline{P}_{Hz})^{-0.22}} \tag{6}$$

where constant B has been arbitrarily chosen to equal $(3.09\times10^{-3})^{-0.22}$ to make $\varphi_s = \psi_s = 1$ at $\overline{P}_{Hz} = 3.09\times10^{-3}$. Incorporating equation (6) into equation (5) yields

$$\overline{H}_{min} = K_j^* \overline{U}^{\hat{n}_j} \overline{P}_{Hz}^{-0.22} \varphi_s$$

where $K_j^* = k_j/B$. Coefficient K_j^* together with exponent $\hat{n}_j$ and factor k_j are listed in table III. The parameter φ_s is referred to as an EHD high-contact-stress factor. It is, in essence, a measure of the additional reduction of lubricant minimum film thickness not forecasted by commonly used film thickness formulas.

Figure 5 shows the variation of factor φ_s with contact stress together with the following polynomial expression which closely fits this curve:

$$\varphi_s = \overline{P}_{Hz}(150. - 27.5\times10^3\overline{P}_{Hz}) + 0.806 \tag{8}$$

It can be seen that at the lower contact stresses where film thickness sensitivity to contact stress has been shown to be in accordance with theory (refs. 6, 8, 9, and 24) the value of the φ_s factor is close to unity. With increasing contact stress the disparity between theory and measurement becomes more significant as evidenced by the diminishing value of φ_s. In fact, at the maximum experimental contact pressure of 2.4×10^9 N/m^2 (3.5×10^5 psi), it is apparent from figure 5 that the measured value of film thickness is approximately half that predicted by film thickness formulas developed from classical theory.

In viewing the variation of the $\overline{U}$ exponent $\hat{n}_j$, appearing in table III, it is apparent that the value of $\hat{n}_j$ for the first three test fluids are nearly equal, averaging approximately 0.62. However, the value of $\hat{n}_j$ for the polyphenyl ether fluid set at 0.83 is significantly higher than the rest. This result qualitatively corresponds to the optical film thickness experiments conducted by Westlake and Cameron (ref. 25). In reference 25 the speed-viscosity parameter exponent of a similar polyphenyl ether fluid, at a value of 0.82, was found to be somewhat larger than the exponent of any other fluid tested including that for a fluorocarbon and synthetic paraffinic lubricant. However, it should be pointed out that in contrast to the results of the present work no variation of $\hat{n}_j$ with p_{Hz} for the minimum film thickness case was observed in reference 25 for either the polyphenyl ether or synthetic paraffinic oils. Because of the apparent differences in operational conditions in terms of inlet shear rates, contact geometry, and, most importantly, contact stress levels, a definitive comparison between the results of optical film thickness measurements reported in reference 25 and those arrived at by using the X-ray technique is not possible.

It was determined from numerical comparisons between the X-ray test data and predictions from equation (7) that, due to the commonality of the values of exponent $\hat{n}_j$ among the test fluids, only a small loss of accuracy would result by setting $\hat{n}_j$ at a nominal value of 0.62 for all four test fluids. Taking advantage of this last simplification, equation (7) can be written in the following final form:

$$\overline{H}_{min} = K_j \overline{U}^{0.62} \overline{P}_{Hz}^{-0.22} \varphi_s \tag{9}$$

where the lubrication parameter K_j^* has been adjusted to K_j to reflect the change in the exponent of $\overline{U}$. Table III lists the appropriate value of K_j for use in equation (9).

DISCUSSION OF RESULTS

Comparison with Test Data in View of Commonly Used Film Thickness Formulas

The results of the present analysis are compared with the X-ray test data at several temperatures in figure 6. In this figure, nondimensional minimum film thickness is plotted as a function of maximum Hertz pressure at several rolling speeds. The numerical results of this comparison are summarized in table IV. It is evident that the derived film thickness formula (eq. (9)), although reduced to a simplified form, does represent the measured data quite well for all four test fluids over the full range of test conditions. Predicted film thicknesses are generally within 0.05 micrometer (2 μin.)

or 10 percent, which ever is greater, of those measured except for those of the synthetic paraffinic oil at high and low values of $\overline{U}$. At these conditions limits of accuracy are set to ±35 percent due to, in part, the simplifications made earlier.

Comparisons made between the X-ray test data and those film thickness values computed from commonly used film thickness formulas have shown that these formulas may seriously overestimate the extent of lubricant film actually present within the contact (refs. 18 and 22). This is illustrated in figure 7 which shows a comparison from reference 22 of predicted film thickness from a widely used isothermal EHD formula for bodies in line contact and measured film thickness for a synthetic paraffinic oil.

At the higher rolling speeds and lower disk temperatures, hence higher lubricant viscosity, the effects of shear heating at the inlet of the contact zone are known to cause appreciable film thinning (refs. 17 and 26). Thermal effects can be taken into account by applying the thermal reduction φ_T provided by Cheng (ref. 18) to the film thickness values computed from the isothermal formula. Correcting calculated film thickness for thermal effects markedly reduces the overall magnitude discrepancy between predicted and measured film thickness at the most severe heating test condition (fig. 7(a)) but has a relatively mild effect at the lower disk speeds and at any of the elevated temperature test conditions as can be seen, for example, from figure 7(b). Furthermore, the thermal reduction factor φ_T is a very weak function of contact pressure (ref. 17). Consequently, heating effects would not be expected either to appreciably influence the relation of isothermal film thickness to applied load or explain the high film thickness - load dependency exhibited by the X-ray test data.

Effect of Lubricant on Film Thickness Correlation

The empirical factors used to formulate the present correlation have been developed from the experience gained with four test fluids. At present no meaningful generalizations can be made regarding the extension of the present deduced relation to systems employing different lubricant types or formulations without the benefit of additional experimental information. On the other hand, application of the film thickness correlation to systems utilizing the lubricants under study over similar conditions can be made with reasonable confidence.

It may be apparent that the pressure-viscosity coefficient α, which is customarily used to characterize the film forming capabilities of a lubricant apart from the effects of absolute viscosity, is conspicuously absent from the present formulation. In the present model, the role formerly played by α has been fulfilled in part by the lubricant coefficient K_j. That is, a lubricant's film forming capabilities can be ascertained by knowing its K_j and its absolute viscosity μ_o at a given operating temperature condition.

In the case of a rolling element bearing, this temperature can be taken as the temperature of the rolling element surface of interest. The good qualitative correlation which exists between the deduced lubricant coefficients K_j appearing in table III and the room temperature values of the pressure-viscosity coefficient α listed in table I determined from optical film thickness measurements (refs. 25 and 27) tends to support this contention.

An important distinction between α and K_j is that α is temperature dependent where K_j as presently defined is not. A careful examination of test data revealed that the effects of temperature on minimum film thickness are adequately reflected by the variation in absolute viscosity and that the added complication of an additional temperature dependent variable could thus be avoided. Furthermore, the availability of pertinent pressure-viscosity data at elevated temperature under the appropriate shear rate and pressure conditions for film thickness calculation purposes have been generally limited. There has been, however, increasingly more attention directed at obtaining these pressure-viscosity data in recent years (refs. 20, 25, and 27). In view of the aforementioned, the pressure-viscosity coefficient α was eliminated in the present film thickness model.

Effects of Contact Geometry and Material

Contact geometry. - The present correlation is based exclusively on measurements made with a single disk geometry chosen to simulate the ball inner race contact of a 120-millimeter-bore angular-contact ball bearing (ref. 28). The contact between the test disks approaches the condition of line contact with an ellipticity ratio b/a of 5.9 where b and a are the major and minor semi-axes of the contact ellipse, respectively. The equivalent radius of curvature in the direction of rolling R' for the test disks is 0.915 centimeter (0.36 in.).

It is difficult to extend the results presented herein to different contact geometries with complete assurance without further experimental verification. However, from a practical standpoint, it is anticipated that the overall effect of contact geometry on the value of film thickness is minimal. Cheng (ref. 23) has theoretically shown that the proportions of the contact ellipse with b/a varying from 1 to 5 have a relatively mild effect on film thickness; that is, the dependence of film thickness on the mean surface speed u, absolute viscosity μ_0, and the contact stress p_{Hz} changes little as the shape of the contact ellipse varies from point to line contact. Similarly, Archard and Cowking (ref. 29) have established that there is great similarity between the EHD lubrication of point and line contacts. A second factor is that the line contact geometry selected for the X-ray test disks does in fact simulate the contact shape that exists between the races and the balls or rollers in rolling-element bearings as well as the contact shape between gear teeth which usually approximate the line contact case. In view of these considerations, the semi-empirical minimum film thickness formula presented can be used with reasonable certainty for most practical applications without further modifications for small differences in contact geometry.

With regard to the size of the contacting elements, elastohydrodynamic theory indicates that film thickness is moderately dependent on the contacting elements' equivalent radius of curvature in the rolling direction R' (ref. 1). The current film thickness is a function of R' to the 0.38 power at a given contact stress level. This value is approximately in accordance with conventional EHD theory. However, it is recognized that the sensitivity of h_{min} to R' in the heavy load regime remains to be established experimentally. Until such time, minimum film thicknesses forecasted by equation (9) will be most successful for those systems in which the R' of the contact approximates that of the system under study. This limitation is not anticipated to seriously encumber the use of this film thickness expression for the majority of applications.

Material. - The effects of material properties in terms of Young's modulus on film thickness at high contact pressures (up to 3.45×10^9 N/m^2 (5×10^5 psi) maximum Hertz stress) have been demonstrated by experiment (ref. 30) to be minimal. These tests which confirm theoretical expectations were conducted by Gohar (ref. 30) using interferometry to measure the film generated between a rolling steel ball and a flat glass plate. It is anticipated that the choice of materials other than steel will not appreciably alter the form of equation (9), unless the elastic properties of the material of interest are markedly different from those of steel.

Relative Film Forming Ability of Test Fluids

As a design tool, the film thickness correlation developed herein can be used to screen prospective lubricants for particular applications once the lubricant coefficient K_j has been experimentally ascertained. For example, the relative film forming ability of the test lubricants can be ranked at elevated temperatures under a practical contact load condition as shown in figure 8. In this plot the lubricant film thickness factor H^* defined as the minimum film thickness $\overline{H}_{min}$ divided by a dimensionless speed factor G is plotted against the disk temperature at a maximum Hertz stress of 2.07×10^9 N/m^2 (300 000 psi). It should be mentioned that the use of the lubricant film thickness factor H^* in this figure does not alter the relative ranking of the lubricants made in this comparison. This parameter serves mainly as a convenient scale factor to account for the beneficial effects of speed which affect all of the lubricants to the same extent.

It is apparent that the fluorocarbon lubricant stands out as the best film former at the elevated temperatures. This is attributed to the fluorocarbon fluid's high absolute viscosity in conjunction with the fact that it had the largest lubricant coefficient K_j of all the lubricants examined.

The synthetic paraffinic oil without additives shown in this figure generates about one-third as much film as does the fluorocarbon lubricant. The presence of an organic phosphonate antiwear additive to improve boundary lubricating characteristics would make the synthetic paraffinic oil as good a film former as the fluorocarbon according to the X-ray film thickness data published in reference 10. This observation is further strengthened by full-scale bearing tests (ref. 12). These tests showed that a 120-millimeter-bore angular-contact ball bearing could achieve long term operation with either the synthetic paraffinic oil containing an additive or the fluorocarbon lubricant at temperatures to 589 K (600° F) under a comparable heavy load condition (maximum Hertz stress of inner race ball contact of approximately 2.07×10^{9} N/m^2 (300 000 psi)).

CONCLUDING REMARKS

The existence of the unexpectedly high dependence of film thickness on maximum Hertz stress within a highly loaded concentrated contact as exhibited by the X-ray test data has not yet received universal acceptance. As discussed previously, the reservations which exist concerning this phenomena stem largely from the absence of a well defined theoretical explanation. Apart from the experimental investigations cited previously there is at present a scarcity of published film thickness test data under conditions of high-contact stress which would help to resolve this controversy. Nonetheless, the results from full-scale bearing tests conducted in reference 12 tend to affirm the existence of such a high load - film thickness dependence. In this investigation, a series of 120-millimeter-bore angular-contact ball bearings were subject to long term fatigue tests at elevated temperatures. These bearings had contact geometries which were modeled by the test disks used in the X-ray experiments (refs. 10 and 11).

Preliminary tests were conducted with a synthetic paraffinic oil similar to that studied herein. The inner race had an operating temperature of 589 K (600° F) and an inner raceway maximum Hertz stress 2.36×10^{9} N/m^2 (342 000 psi). Inspection of several bearings revealed that appreciable ball wear had occurred and that surface glazing was apparent on the races. It was concluded from this inspection that these bearings were operating in a mixed boundary elastohydrodynamic lubrication regime with boundary lubrication being the predominant mode. Similar conclusions were drawn when repeating this preliminary test with a fluorocarbon lubricant but at a slightly lower maximum Hertz stress level of 2.23×10^{9} N/m^2 (323 000 psi).

As a corrective measure, the thrust load acting on these bearings was reduced to assure predominantly EHD lubrication for the remainder of the fatigue tests. This resulted in a reduction of the maximum Hertz stress acting on the inner race to approximately 2.23×10^{9} N/m^{2} (323 000 psi) and 2.04×10^{9} N/m^{2} (295 000 psi) for the bearing operating with the synthetic paraffinic and fluorocarbon lubricants, respectively.

Subsequent examination under these new load conditions of an unfailed test bearing run with the synthetic paraffinic oil indicated that there was no measurable wear or weight change of bearing components. On the basis of this post-test examination, which also included surface trace measurements, together with the appearance of the rolling-element surfaces it was concluded that almost complete EHD lubrication existed during the testing of these bearings. Post-test examination of the fluorocarbon lubricated bearings revealed that the predominant mode of lubrication had also been affected. The lubrication mode ranged from boundary to elastohydrodynamic with some test bearings exhibiting extremely good surface appearances and others showing considerable glazing.

The dramatic improvement in the quality of lubrication in the case of the synthetic paraffinic lubricated bearings and in those operated with the fluorocarbon fluid to a lesser extent as a result of the small reduction in the operating raceway contact stress level tends to give physical confirmation to the existence of a high dependency of film thickness to contact stress.

In terms of classical EHD theory, the reported reductions in maximum Hertz stress levels acting on the synthetic paraffinic (1.31×10^{8} N/m^{2} (19 000 psi)) and fluorocarbon (1.93×10^{8} N/m^{2} (28 000 psi)) lubricated bearings would result in an increase of lubricant film of only 1.3 and 2.0 percent, respectively. Obviously, this increase in film is not sufficient to account for the observed shift in the system's mode of lubrication. In contrast to classical theory, the contact pressure terms appearing in the heavy load film thickness formula (eq. (9)), that is, $(\overline{P}_{Hz})^{-0.22}\varphi_{s}$, would yield films that were 19.2 and 22.0 percent greater for the synthetic paraffinic and fluorocarbon lubricants, respectively. These predicted increases in film thickness are undoubtedly more in line with those actually experienced.

SUMMARY OF RESULTS

A minimum film thickness correlation for heavily loaded elastohydrodynamic contacts was formulated from X-ray film thickness measurements made with synthetic paraffinic, fluorocarbon, type II ester, and polyphenyl ether test fluids. The film thickness test data covered a wide and practical range of operating conditions. Maximum Hertz stresses ranged from 1.04×10^{9} to 2.42×10^{9} N/m^{2} (150 000 to 350 000 psi), disk temperatures from 339 to 589 K (150° to 600° F), and mean surface rolling speeds from

9.4 to 37.6 meters per second (370 to 1480 in./sec). The deduced film thickness formula contained a high-contact-stress factor φ_S to correct classical EHD formulas for the high sensitivity of minimum film thickness to contact stress exhibited by the test data under heavy loads.

Predicted values of minimum film thickness were compared to X-ray film thickness measurements and contrasted against the results from a well known isothermal EHD analysis with and without thermal corrections for shear heating. The effects of contact geometry, material, and lubricant properties on predicted film thickness were considered. The following results were obtained:

1. The film thickness correlation represented the X-ray test data reasonably well throughout the full range of test conditions.

2. The observed greater dependency of film thickness to maximum Hertz stress than that accounted for by classical EHD theory is responsible for film thicknesses computed by commonly used formulas to be up to twice larger than those actually measured. Corrections can be made by applying the EHD high-contact stress factor φ_S developed herein to these computed values.

3. It is anticipated that the empirical film thickness formula can be used to forecast minimum film thickness under heavy loads with reasonable certainty for many rolling-element bearing and gear systems. These include principally those systems employing the lubricants studied herein and whose contact geometry approximates that on which the model is based.

4. A theoretical consideration of lubricant shear heating at the inlet of the contact zone does not appreciably alter the weak relation of isothermal film thickness to contact stress nor significantly diminish the magnitude discrepancy between theory and the X-ray data except at the most severe heating conditions.

5. At elevated lubricant temperatures the fluorocarbon lubricant appears to be the best film former followed by the synthetic paraffinic (without an antiwear additive), polyphenyl ether, and type II ester fluids in diminishing film forming order.

Lewis Research Center,
National Aeronautics and Space Administration,
and
U.S. Army Air Mobility R&D Laboratory,
Cleveland, Ohio, August 5, 1974,
501-24.

REFERENCES

1. Dowson, D.; and Higginson, G. R.: Elasto-Hydrodynamic Lubrication: The Fundamentals of Roller Gear Lubrication. Pergamon Press, 1966.
2. McGrew, J. M.; Gu, A.; Cheng, H. S.; and Murray, S. F.: Elastohydrodynamic Lubrication-Preliminary Design Manual. Mechanical Technology, Inc. (AFAPL-TR-70-27; AD-877768), 1970.
3. Tallian, T.; Sibley, L.; and Valori, R.: Elastohydrodynamic Film Effects on the Load Life Behavior of Rolling Contacts. Paper 65-LUBS-11, ASME, 1965.
4. Tallian, T.: On Competing Failure Modes in Rolling Contact. ASLE Trans., vol. 10, no. 4, Oct. 1967, pp. 418-439.
5. The Rolling-Elements Committee, ASME Lubrication Division: Life Adjustment Factors for Ball and Roller Bearings - An Engineering Design Guide. ASME, 1971.
6. Crook, A. W.: The Lubrication of Rollers. II. Film Thickness with Relation to Viscosity and Speed. Phil. Trans. Roy. Soc. (London), vol. 254A, no. 1040, Dec. 1961, pp. 223-237.
7. Sibley, L. B.; and Orcutt, F. K.: Elasto-Hydrodynamic Lubrication of Rolling-Contact Surfaces. ASLE Trans., vol. 4, no. 2, Nov. 1961, pp. 234-249.
8. Christensen, H.: The Lubrication of Rolling Contacts, Measurement of Film Thickness. Acta Polytech. Scand. Mech. Eng., Ser. 15, 1963, pp. 1-30.
9. Dyson, A.; Naylor, H.; and Wilson, A. R.: The Measurement of Oil-Film Thickness in Elastohydrodynamic Contacts. Proc. Inst. Mech. Eng., vol. 180, pt. 3B, 1965-1966, pp. 119-134.
10. Parker, Richard J.; and Kannel, Jerrold W.: Elastohydrodynamic Film Thickness Between Rolling Disks with a Synthetic Paraffinic Oil to 589 K (600° F). NASA TN D-6411, 1971.
11. Parker, Richard J.; and Kannel, Jerrold W.: Elastohydrodynamic Film Thickness Measurements with Advanced Ester, Fluorocarbon, and Polyphenyl Ether Lubricants to 589 K (600° F). NASA TN D-6608, 1971.
12. Bamberger, E. N.; Zaretsky, E. V.; and Anderson, W. J.: Effect of Three Advanced Lubricants on High-Temperaring Bearing Life. J. Lub. Tech., vol. 92, no. 1, Jan. 1970, pp. 23-33.

13. Grubin, A. N.; and Vinogradova, I. A.: Fundamentals of the Hydrodynamic Theory of Heavily Loaded Cylindrical Surfaces. Symposium on Investigation Into the Contact of Machine Components. Book No. 30, Central Scientific Institute for Technology and Mechanical Engineering, Moscow, 1949. (D.S.I.R. Trans. No. 337.)

14. Dowson, D.; and Higginson, G. R.: A Numerical Solution to the Elasto-Hydrodynamic Problem. J. Mech. Eng. Sci., vol. 1, no. 1, 1959, pp. 6-15.

15. Kannel, J. W.; and Bell, J. C.: Interpretations of the Thickness of Lubricant Films in Rolling Contact. 1 - Examination of Measurements Obtained by X-Rays. Paper 71-LUB-S, ASME, 1971.

16. Bell, J. C.; and Kannel, J. W.: Interpretations of the Thickness of Lubricant Films in Rolling Contact. 2 - Influence of Possible Rheological Factors. Paper 71-LUB-T, ASME, 1971.

17. Cheng, H. S.: Calculation of Elastohydrodynamic Film Thickness in High Speed Rolling and Sliding Contacts. MTI-67TR24, Mechanical Technology, Inc. (AD-652924), May 1967.

18. Cheng, H. S.: Isothermal Elastohydrodynamic Theory for the Full Range Pressure-Viscosity Coefficient. J. Lub. Tech., vol. 94, no. 1, Jan. 1972, pp. 35-43.

19. Loewenthal, Stuart H.; and Zaretsky, Erwin V.: Elastohydrodynamic Analysis Using a Power Law Pressure-Viscosity Relation. NASA TN D-7121, 1973.

20. Bohn, M.; Carlson, S.; Lee, D.; Jakobsen, J.; Sanborn, D. M.; and Winer, W. O.: Investigations of Lubricant Rheology as Applied to Elastohydrodynamic Lubrication. Georgia Inst. of Tech. (NASA CR-127826), 1972.

21. Sanborn, D. M.; and Winer, W. O.: Fluid Rheological Effects in Sliding Elastohydrodynamic Point Contacts with Transient Loading: I - Film Thickness. J. Lubr. Tech., vol. 93, no. 2, Apr. 1971, pp. 262-271.

22. Loewenthal, Stuart H.; Parker, Richard J.; and Zaretsky, Erwin V.: Elastohydrodynamic Film Thickness Formula Based on X-Ray Measurements with a Synthetic Paraffinic Oil. NASA TN D-7502, 1973.

23. Cheng, H. S.: A Numerical Solution of Elastohydrodynamic Film Thickness in an Elliptical Contact. J. Lub. Tech., vol. 92, no. 1, Jan. 1970, pp. 155-162.

24. Foord, C. A.; Hammann, W. C.; and Cameron, A.: Evaluation of Lubricants Using Optical Elastohydrodynamics. ASLE Trans., vol. 11, no. 1, Jan. 1968, pp. 31-43.

25. Westlake, F. J.; and Cameron, A.: Optical Elastohydrodynamic Fluid Testing. ASLE Trans., vol. 15, no. 2, Apr. 1972, pp. 81-95.

26. Greenwood, J. A.; and Kauzlarich, J. J.: Inlet Shear Heating in Elastohydrodynamic Lubrication. J. Lub. Tech., vol. 95, no. 4, Oct. 1973, pp. 417-426.

27. Chiu, Y. P.; Hahn, D.; and McCool, J. I.: Exploratory Analysis of Elastohydrodynamic Properties of Lubricants. Rept. AL72P010, SKF Industries, Inc., 1972.

28. Bell, J. C.; and Kannel, J. W.: Simulation of Ball-Bearing Lubrication with Rolling-Disk Apparatus. J. Lub. Tech., vol. 92, no. 1, Jan. 1970, pp. 1-15.

29. Archard, J. F.; and Cowking, E. W.: Elastohydrodynamic Lubrication at Point Contacts. Proc. Inst. Mech. Eng., vol. 180, pt. 3B, 1965-1966, pp. 47-56.

30. Gohar, R.: Oil Film Thickness and Rolling Friction in Elastohydrodynamic Point Contact. J. Lubr. Tech., vol. 93, no. 3, July 1971, pp. 371-382.

TABLE I. - PROPERTIES OF TEST LUBRICANTS[a]

Property	Lubricant designation			
	Type II ester	Fluorocarbon	Polyphenyl ether	Synthetic paraffinic oil
Additives	Oxidation inhibitor; corrosion inhibitor; antiwear additive	None	Oxidation inhibitor	None
Kinematic viscosity, cS (or 10^{-6} m^2/sec):				
At 233 K (-40° F)	8900	>100 000	>100 000	>100 000
At 311 K (100° F)	29.0	298	358	443
At 372 K (210° F)	5.4	29.8	13.0	39.7
Flash point, K ($^{\circ}$F)	533 (500)	None	555 (500)	542 (515)
Autoignition temperature, K ($^{\circ}$F)	716 (830)	None	886 (1135)	703 (805)
Specific gravity at 478 K (400° F)	0.85	1.59	1.05	0.74
Pressure-viscosity coefficient at 298 K (77° F), m^2/N (psi^{-1})	[b]1.6×10^{-8} (1.1×10^{-4})	[c]3.6×10^{-8} (2.5×10^{-4})	[b]2.8×10^{-8} (1.9×10^{-4})	[b]1.6×10^{-8} (1.1×10^{-4})

[a]From ref. 11.
[b]From ref. 27.
[c]From ref. 25.

TABLE II. - EMPIRICAL MINIMUM FILM THICKNESS COEFFICIENTS FOR EQUATION (1)

Nondimensional stress parameter, $\overline{P}_{Hz}$	Maximum Hertz stress, P_{Hz}		Synthetic paraffinic		Fluorocarbon		Type II ester		Polyphenyl ether	
	N/m^2	psi	C_{i_1}	n_{i_1}	C_{i_2}	n_{i_2}	C_{i_3}	n_{i_3}	C_{i_4}	n_{i_4}
3.09×10^{-3}	1.04×10^{9}	1.5×10^{5}	28.1	0.58	90.6	0.60	35.8	0.60	58.1	0.60
4.12	1.38	2.0	44.6	.61	109.	.61	34.2	.60	7.50×10^{3}	.81
5.14	1.72	2.5	93.5	.65	127.	.62	36.8	.61	5.25×10^{4}	.90
6.17	2.07	3.0	231.	.70	64.6	.61	28.1	.61	5.83×10^{5}	1.0
6.69	2.24	3.25	-----	----	38.8	.59	----	----	--------	----
7.20	2.42	3.5	655.	.76	-----	----	12.6	.59	--------	----

TABLE III. - MEAN VALUES OF THE DIMENSIONLESS SPEED VISCOSITY

PARAMETER EXPONENT $\hat{n}_j$ AND LUBRICANT

PARAMETERS k_j, K_j^*, AND K_j

Parameter	Type II ester	Fluorocarbon	Polyphenyl ether	Synthetic paraffinic
k_j	38.	109.	1.4×10^{4}	156.
K_j^*	10.7	30.6	3940.	43.8
$\hat{n}_j$	.60	.61	.83	.66
K_j	18.2	44.8	24.9	18.2

TABLE IV. - COMPARISON BETWEEN PREDICTED AND MEASURED MINIMUM FILM THICKNESS[a]

(a) Synthetic paraffinic oil

Disk speed, rpm	Maximum Hertz stress		Disk temperature, K (°F)					
	N/m²	psi	339 (150)		422 (300)		478 (400)	
			Nondimensional minimum film thickness, $\overline{H}_{min}$					
			X-ray data	Equation (9)	X-ray data	Equation (9)	X-ray data	Equation (9)
5 000	1.04×10^{9}	1.5×10^{5}	98×10^{-6}	79×10^{-6}	21×10^{-6}	19×10^{-6}	14×10^{-6}	11×10^{-6}
	1.72	2.5	80	60	13	14	8	9
	2.4	3.5	46	30	4	7	2	4
10 000	1.04	1.5	116	122	36	29	25	17
	1.72	2.5	96	92	24	22	19	13
	2.4	3.5	55	47	11	11	7	7
20 000	1.04	1.5	142	186	48	45	39	27
	1.72	2.5	110	140	35	34	25	20
	2.4	3.5	67	71	17	17	16	10

(b) Type II ester lubricant

Disk speed, rpm	Maximum Hertz stress		Disk temperature, K (°F)					
	N/m²	psi	339 (150)		366 (200)		422 (300)	
			Nondimensional minimum film thickness, $\overline{H}_{min}$					
			X-ray data	Equation (9)	X-ray data	Equation (9)	X-ray data	Equation (9)
5 000	1.04×10^{9}	1.5×10^{5}	25×10^{-6}	26×10^{-6}	12×10^{-6}	12×10^{-6}	7×10^{-6}	7×10^{-6}
	1.72	2.5	19	20	9	9	5	6
	2.07	3.0	11	10	5	5	3	3
10 000	1.04	1.5	42	40	22	19	12	11
	1.72	2.5	31	30	16	14	9	9
	2.07	3.0	18	15	8	7	5	4
20 000	1.04	1.5	53	61	29	29	18	17
	1.72	2.5	41	46	25	22	15	13
	2.07	3.0	22	23	15	11	7	7

(c) Fluorocarbon lubricant

Disk speed, rpm	Maximum Hertz stress		Disk temperature, K (°F)							
	N/m²	psi	422 (300)		478 (400)		534 (500)		589 (600)	
			Nondimensional minimum film thickness, $\overline{H}_{min}$							
			X-ray data	Equation (9)	X-ray data	Equation (9)	X-ray data	Equation (9)	X-ray data	Equation (9)
5 000	1.04×10^{9}	1.5×10^{5}	68×10^{-6}	60×10^{-6}	31×10^{-6}	35×10^{-6}	21×10^{-6}	23×10^{-6}	16×10^{-6}	16×10^{-6}
	1.72	2.5	46	45	20	26	17	17	12	12
	2.07	3.0	31	35	15	20	13	13	9	10
10 000	1.04	1.5	90	92	49	53	38	35	26	25
	1.72	2.5	73	70	36	40	31	27	21	19
	2.07	3.0	53	54	26	31	25	21	15	15
20 000	1.04	1.5	125	142	79	82	56	54	38	39
	1.72	2.5	101	108	68	62	46	41	31	29
	2.07	3.0	75	83	48	48	35	32	25	23

(d) Polyphenyl ether lubricant

Disk speed, rpm	Maximum Hertz stress		Disk temperature, K (°F)			
	N/m²	psi	422 (300)		505 (450)	
			Nondimensional minimum film thickness, $\overline{H}_{min}$			
			X-ray data	Equation (9)	X-ray data	Equation (9)
5 000	1.04×10^{9}	1.5×10^{5}	16×10^{-6}	16×10^{-6}	-------	-------
	1.72	2.5	11	9	4×10^{-6}	6×10^{-6}
	2.4	3.5	8	6	3	3
10 000	1.04	1.5	25	22	-------	-------
	1.72	2.5	21	14	6	10
	2.4	3.5	17	9	4	5
20 000	1.04	1.5	36	34	-------	-------
	1.72	2.5	32	22	12	15
	2.4	3.5	27	15	8	7

[a]From refs. 10 and 11.

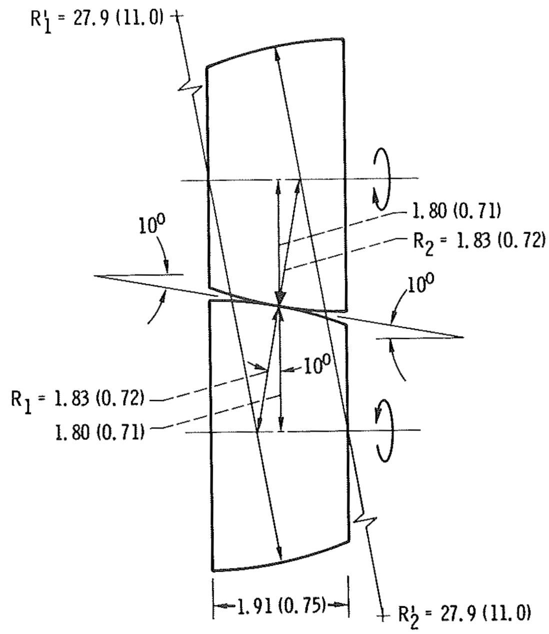
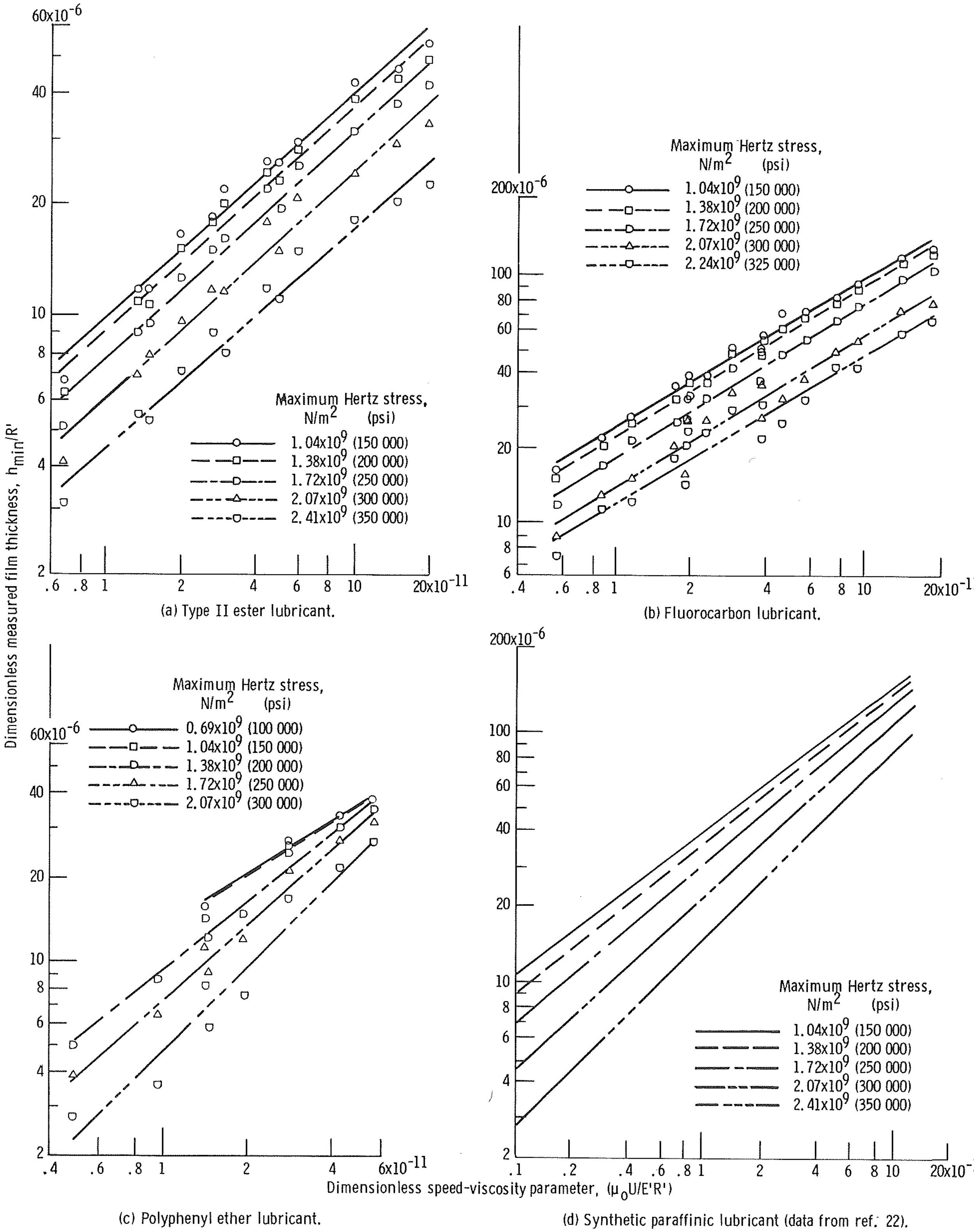


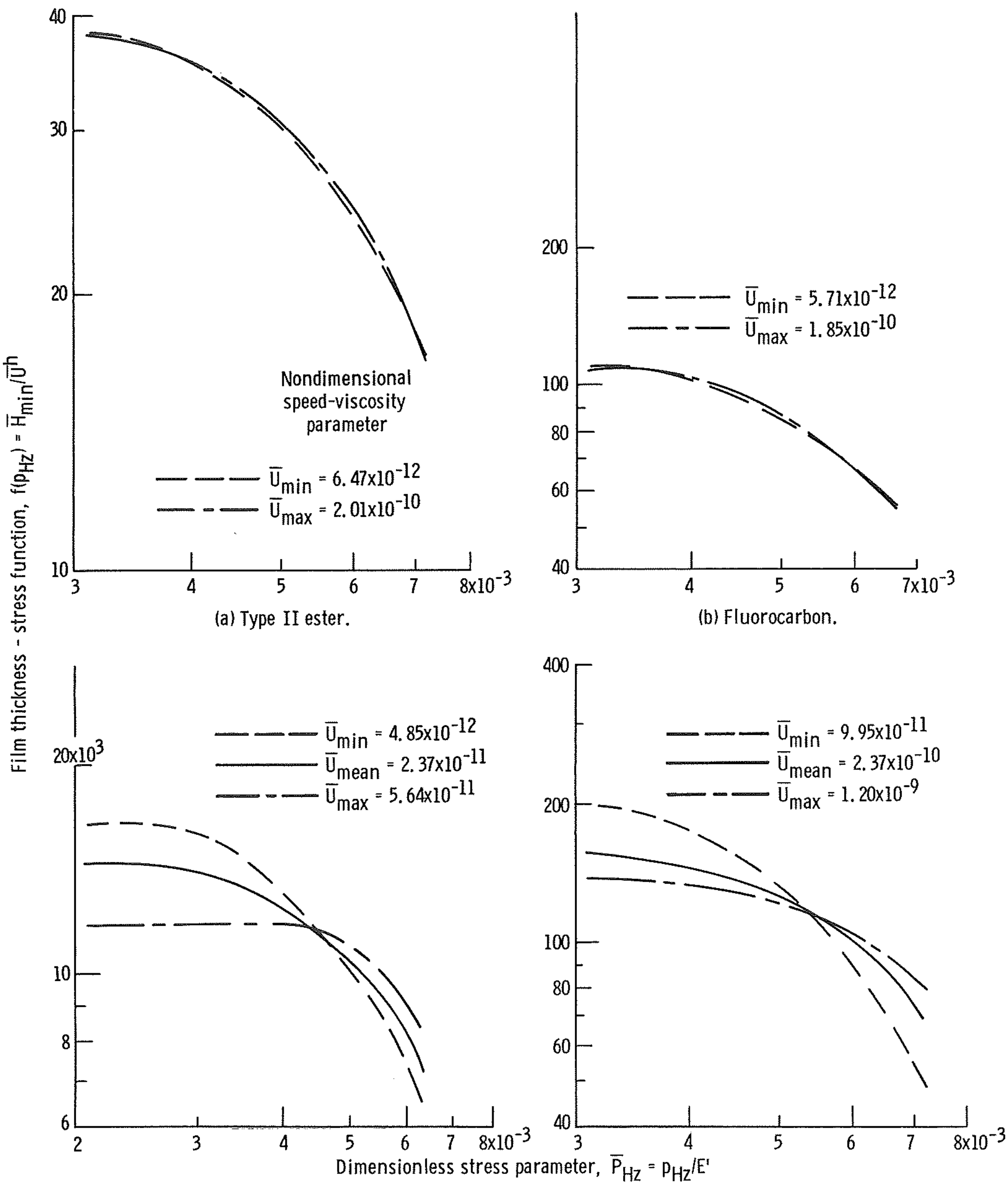
Figure 1. - Contacting disk geometry (ref. 10). (All linear dimensions in cm (in.).)



(c) Polyphenyl ether lubricant.

(d) Synthetic paraffinic lubricant (data from ref. 22).

Figure 2. - Effect of maximum Hertz stress on sensitivity of minimum film thickness to changes in speed and viscosity for X-ray test lubricants (refs. 10 and 11).



(c) Polyphenyl ether. (d) Synthetic paraffinic.

Figure 3. - Effect of speed and viscosity on dependency of film thickness to maximum Hertz stress.

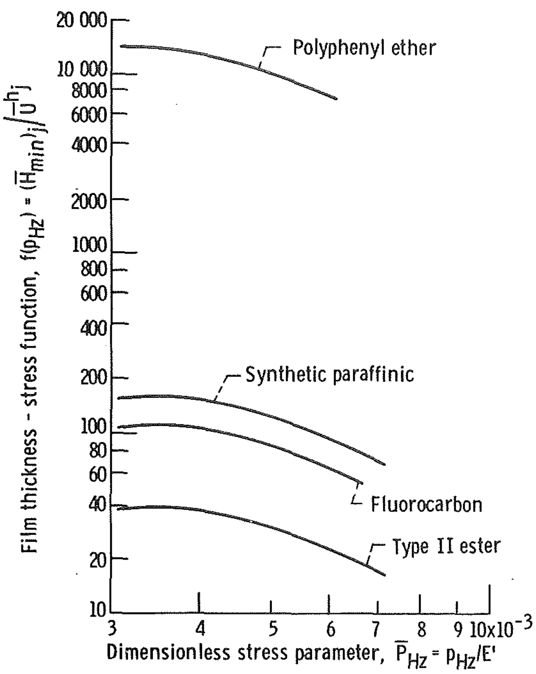


Figure 4. - Effect of maximum Hertz stress on measured minimum film thickness.

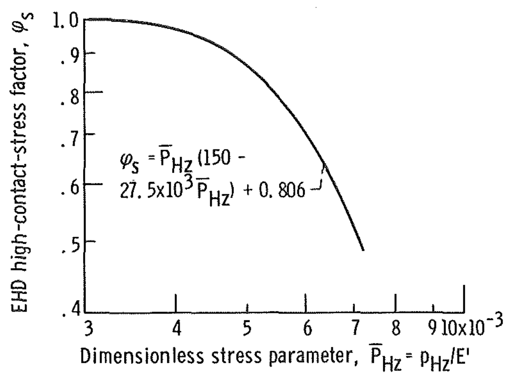


Figure 5. - Variation of EHD high-contact-stress factor with operating contact stress.

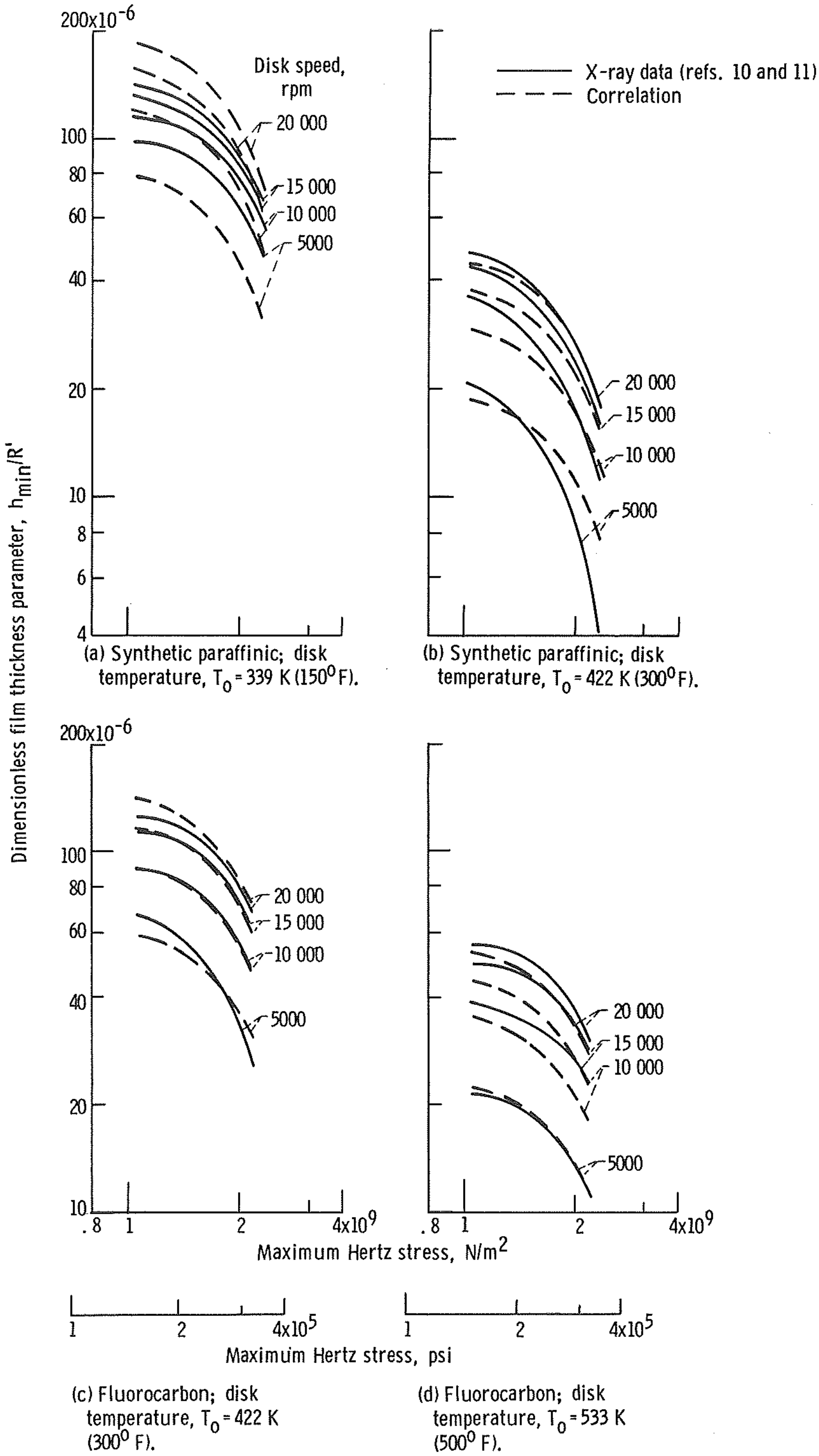
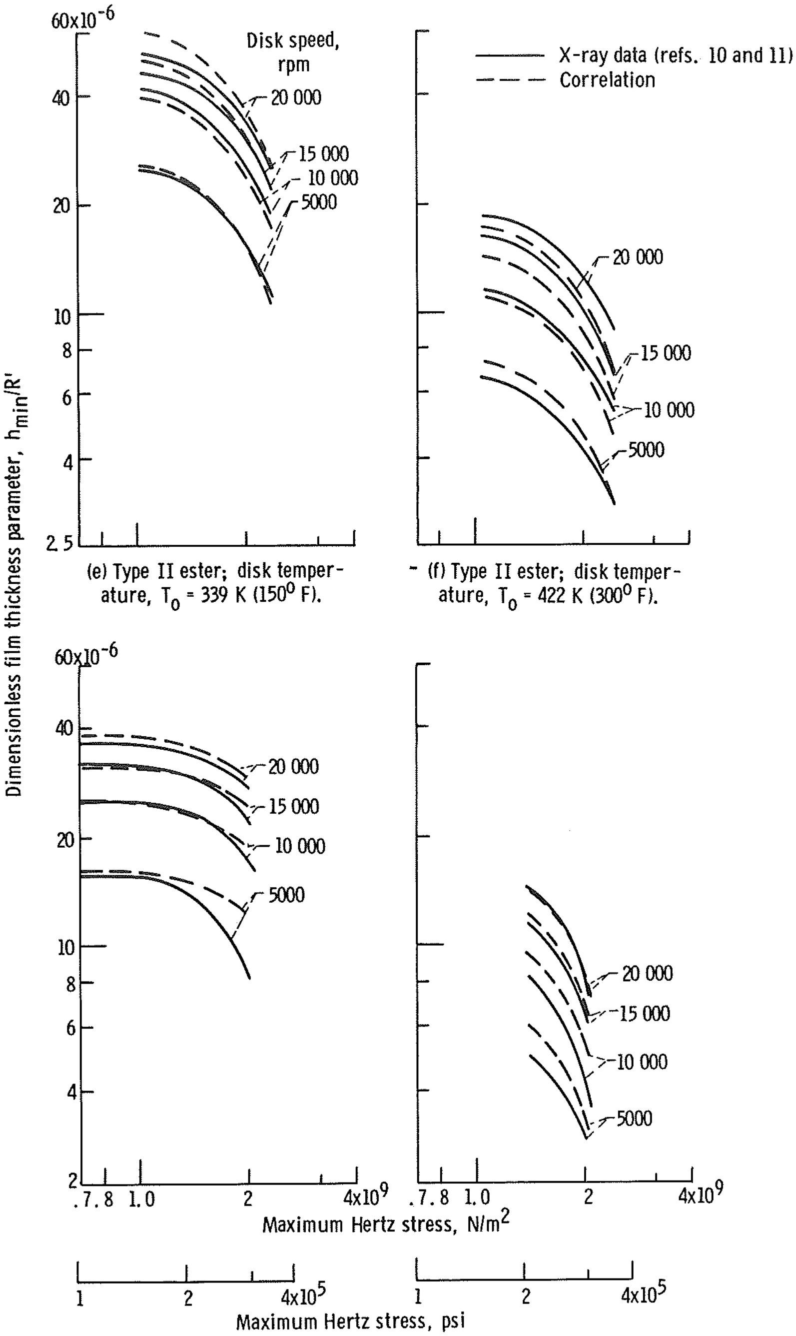


Figure 6. - Comparison between predicted minimum film thickness and X-ray test data (refs. 10 and 11).



(e) Type II ester; disk temperature, T_0 = 339 K (150° F).

(f) Type II ester; disk temperature, T_0 = 422 K (300° F).

(g) Polyphenyl ether; disk temperature, T_0 = 422 K (300° F).

(h) Polyphenyl ether; disk temperature, T_0 = 505 K (450° F).

Figure 6. - Concluded.

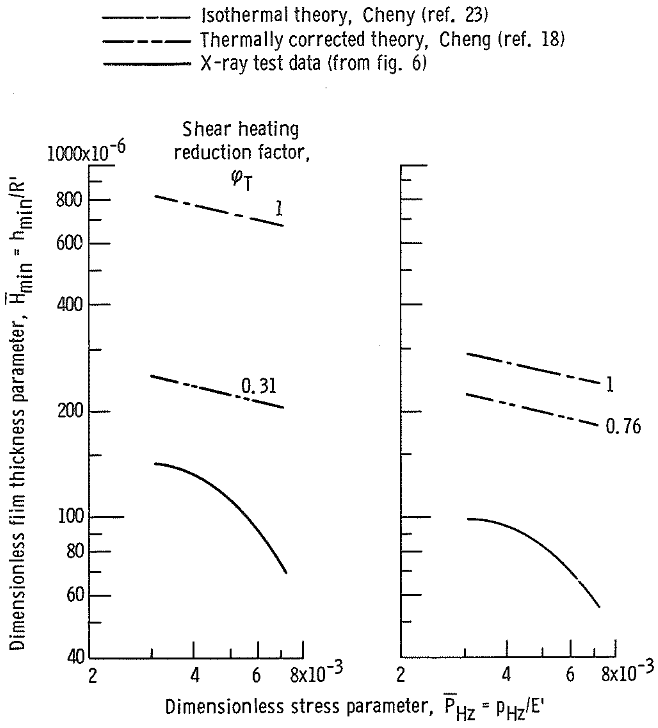


Figure 7. - Comparison of X-ray measured film thickness with calculated isothermal and thermally corrected film thickness for a synthetic paraffinic oil at disk temperature T_0 = 339 K (150° F).

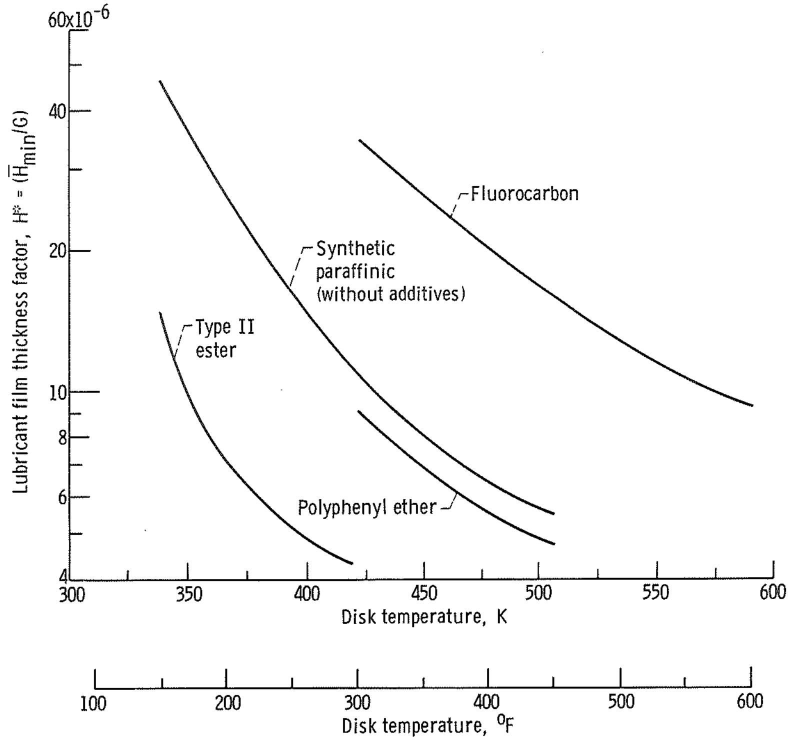


Figure 8. - Relative film forming ability of test lubricants at elevated temperatures. Maximum Hertz stress, 2.07x10^9 newtons per square meter (300 000 psi); dimensionless speed factor, $G = (u/N)^{0.62}$ where N = 9.4 meters per second (370 in./sec) for valid limits of $1 \leq G \leq 2.36$.

NASA-Langley, 1974 E-7954

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
WASHINGTON, D.C. 20546

OFFICIAL BUSINESS
PENALTY FOR PRIVATE USE $300

SPECIAL FOURTH-CLASS RATE
BOOK

POSTAGE AND FEES PAID
NATIONAL AERONAUTICS AND
SPACE ADMINISTRATION
451

U.S.MAIL

POSTMASTER: If Undeliverable (Section 158 Postal Manual) Do Not Return

"The aeronautical and space activities of the United States shall be conducted so as to contribute . . . to the expansion of human knowledge of phenomena in the atmosphere and space. The Administration shall provide for the widest practicable and appropriate dissemination of information concerning its activities and the results thereof."

—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

NASA SCIENTIFIC AND TECHNICAL PUBLICATIONS

TECHNICAL REPORTS: Scientific and technical information considered important, complete, and a lasting contribution to existing knowledge.

TECHNICAL NOTES: Information less broad in scope but nevertheless of importance as a contribution to existing knowledge.

TECHNICAL MEMORANDUMS: Information receiving limited distribution because of preliminary data, security classification, or other reasons. Also includes conference proceedings with either limited or unlimited distribution.

CONTRACTOR REPORTS: Scientific and technical information generated under a NASA contract or grant and considered an important contribution to existing knowledge.

TECHNICAL TRANSLATIONS: Information published in a foreign language considered to merit NASA distribution in English.

SPECIAL PUBLICATIONS: Information derived from or of value to NASA activities. Publications include final reports of major projects, monographs, data compilations, handbooks, sourcebooks, and special bibliographies.

TECHNOLOGY UTILIZATION PUBLICATIONS: Information on technology used by NASA that may be of particular interest in commercial and other non-aerospace applications. Publications include Tech Briefs, Technology Utilization Reports and Technology Surveys.

Details on the availability of these publications may be obtained from:

SCIENTIFIC AND TECHNICAL INFORMATION OFFICE

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Washington, D.C. 20546