This report reviews the fifth in the series of specialty symposia in tribology that have become a regular annual event sponsored by the Institute of Tribology at Leeds University, U.K., and by the Institute Nationale Scientifique Appliquées (INSA), of Lyon, France. The venue alternates annually between Lyon and Leeds. The 1978 meeting was held at the University of Leeds on September 19-22, 1978, on the topic of “Elastohydrodynamics and Related Topics.”

The symposium was attended by 145 persons from eighteen countries. Nearly half, 72, were from the U.K., with 18 from the U.S.A., 16 from France, and 10 from West Germany. A total of 39 papers was presented, with authors from nine countries. This number of papers may have been too many for an ideal symposium. For example, on the 20th, 24 papers were presented between 9:00 am and 10:00 pm. As a result, the time for discussion and the interplay of opinion was extremely limited, and the symposium therefore lacked some of the desirable characteristic of the previous four symposia.

The symposium was opened in late afternoon of the 19th by Professor Duncan Dowson of the University of Leeds. He introduced Professor G. R. Higginson of Durham University, U.K., for the keynote address on “Progress in Elastohydrodynamics.” Professor Higginson reviewed the origins and the history of elastohydrodynamic lubrication, using some 66 well-chosen references to support his points. EHD, as a technical problem area, had its origin in 1916 when Gumbell and Martin published independent studies of the application of Reynolds’ Equation to non-conforming bodies. Martin wrote: “The absence of wear (in gear teeth) must be attributed to the presence of an oil film between the teeth, and as the bearing of the teeth on each other is very narrow, there can be no appreciable motion of the oil parallel to the line of contact, the whole flow taking place in the direction of the relative motion of the surfaces.” At that point in time, the mating surfaces were treated as rigid bodies, and the lubricant as an isoviscous fluid. Higginson pointed out that much of the effort since that time has gone into replacing rigid by elastic and isoviscous by variable viscosity, especially pressure dependent viscosity, and the rewards have been great in narrowing the gap between theory and observation.

Higginson showed an apt turn of phrase in discussing the simplifications in the physical model necessarily used by many researchers in obtaining numerical solutions to the difficult problem, saying “Heroic simplification of the real thing to a physical model which can be solved fairly easily attracts much praise if successful, and scorn if not.” At another point, in discussing the fact that water shows practically no viscosity variation with pressure, he philosophizes, “This guarantees very thin water films between metal bodies rolling or sliding together, and is for example, a merciful safeguard to railway operators on rainy days.”

As an aid in visualizing the different regimes in which EHD can be observed and studied, Professor Higginson presented the diagram shown in Fig. 1. The four regimes shown are defined by two key variables: whether the surfaces are rigid or elastic; and whether the lubricant exhibits little increase in viscosity with pressure, or a significant increase. A brief and useful description of the four regimes of EHD is shown in Fig. 2, ascribed to K. L. Johnson. Here, in addition to the regime boundaries, contours of dimensionless film thickness $h$ are shown on a diagram with axes of two key dimensionless variables, $g_1$ and $g_2$, keyed on the pressure variation of viscosity, and the degree of surface deflection. These variables are defined as: $g_1 = (w^2/\eta R)^{1/3}$ and $g_2 = (w^2/\eta u R^2)^{1/3}$ where $w$ is the normal load per unit length and $E' = 2(1 - \nu^2)/E_1 + (1 - \nu^2)/E_2$. Most of the symposium papers and discussion were devoted to regimes $E' = I$ and $E' = V$.

Finally, Professor Higginson set the stage for the symposium discussion on the traction between surfaces moving at differing velocities. Pointing out that “There is a hefty literature on this defiant aspect of EHL,” he reminds us that there has been great success in estimating...
With this meaty and feisty introduction to the topic of the symposium, the meeting adjourned until the next day, and the attendees were bussed to the "Corn Mill" at Stanford Bridge for a gala dinner in honor of Professor F. T. Barwell of Swansea, who had retired and was on his way to a lectureship in Hong-Kong.

The sessions of the conference were devoted to theoretical fundamentals, experimental fundamentals, thermal effects, lubricant rheology and traction, chemical effects, surface distress, EHD effects in journal bearings, metal forming processes, compliant surface lubrication, and the practical impact of elastohydrodynamic lubrication.

Theoretical Fundamentals
(Chairman W. J. Anderson, NASA, U.S.A.)

Hamrock and Dowson [3] considered the prediction of the minimum film thickness for elliptical contacts formed by contacting spheroids of different radii for the four regimes of fluid film lubrication outlined by Higginson. Fig. 3 shows a three-dimensional visualization of the four regimes showing dimensional film thickness as a function of three variables: viscosity, elasticity and ellipticity parameters.

James and Ettles [6] studied the detailed behavior of an elastic ball dropping under its own weight onto an elastic surface covered by a film of a compressible fluid with pressure dependent viscosity. They concluded that for central pressures less than $10^7$ Pa, the assumption of nonelastic surfaces and an incompressible isoviscous fluid gives a good approximation to the fully varying case. Furthermore, when the initial film thickness is reasonably large, approximately 1 mm or more, the trajectory followed during a drop is independent of the initial conditions of velocity and drop height. In all elastic impacts, the analysis showed that a pocket is formed which grows in size with time and supports virtually all of the load. The pressure falls to zero outside the pocket zone.

Christensen [4] reconsidered his previous theory on the thermal collapse of EHD films in which convection was used as the model for heat transfer in the very thin films. In his new work, he assumed that the heat generated by the rubbing of contacting asperities on the opposing surfaces is carried to the surfaces through conduction through the oil film, thereby increasing the temperature and lowering the viscosity of the film in the inlet zone to the contact. Applying his model to full EHD films, no thermal collapse was predicted. However, for starved films, the model did predict thermal collapse under various combinations of the operating variables.

Now that a rather good understanding of the theoretical behavior of EHD contacts has been attained, more attention is being directed to studies of the influence of surface roughness on EHD and in particular on the type as well as the magnitude of the roughness. Fatir
and Cheng [2] have considered the effect of the orientation of the roughness pattern. They define a surface roughness pattern parameter, \( \gamma \), as the ratio of the correlation length in the direction of flow to the correlation length normal to the flow. Earlier studies by Dyson, and Cheng and Dyson, had utilized a stochastic Reynolds equation for longitudinal roughness. In this study Patir and Cheng based their analysis on a Reynolds' equation resulting from an average flow model. They report finding that the film forming capability, the center film thickness, increases as the surface roughness pattern parameter \( \gamma \) decreases from 6 to 1/6, i.e., from longitudinally oriented to isotropic and then to transversely oriented patterns. Quantitatively, however, they find that the two-dimensional surface roughness structure has a considerably weaker effect than had been suggested by earlier one-dimensional roughness theories. The effect on the center film thickness does not become significant until the film parameter \( \Lambda \) drops well below 2 (Fig. 3(a)).

**Experimental Fundamentals**

(Chairman: Prof. W. R. D. Wilson, U. Mass., Amherst, USA)

This group of six papers indicated that the trend in the fundamental experimental work is oriented toward the influence of surface roughness, the behavior under starved conditions, and some work on the behavior of grease as the lubricant in EHD contacts.

Dalmaz [7] reported some elegant measurements of both traction and film thickness for the elliptical contact between a barrel shaped (radii 1.34 mm and 9.7 mm) rolling element and a flat sapphire rotating disk. Film thickness was measured by optical interferometry, and traction by displacement measurements on the hydrostatic bearing holding the barrel drive. Adjustment of the horizontal and vertical angles of the barrel drive provided a range of lateral sliding and spinning conditions. Fig. 4 is a typical interferogram. Under both fully flooded and starved conditions, the center and minimum film thickness measurements were in good agreement with Hamrock and Dowson's theoretical values under pure rolling conditions. Tractive sliding, lateral sliding, and spinning produced no significant changes in the film thickness. From traction measurements, the effective shear modulus was found to increase with pressure and to have a value of the same order of magnitude as found in disc machines. A limiting shear stress was found, consistent with Dyson's relation and with Winer's glass transition studies.

Gredhill, Jackson, and Cameron [12] studied elliptical-shaped EHD contacts with an aspect ratio of 7 to 10 in the direction of rolling. Their objective was to have a contact zone shape and size that would be typical of those to be expected on ground rolling element surfaces. They conclude that "the evidence presented of micro-EHL film thickness generation under high side leakage conditions suggests that when macro-EHL films are too thin to support the total contact load, micro-EHL may take over load support between interacting asperities." Pietrozak and Krazeminski-Freda [9] studied the effect of surface roughness in an EHD contact. They measured the percentage of time that a heavily loaded EHD contact was in electrical contact, and the number of contacts per unit length. From this the average length of one contact was found and this value was found to be in agreement with the theoretically calculated value.

Experimental observations on starved elastohydrodynamic line contacts were reported by Dowson, Saman, and Toyoda [11]. They used a rolling disc machine with independent disc drives and estimated film thickness by measurement of the capacitance between the discs. The lubricant used was a diester, di-2 ethylhexyl phthalate. They found that if lubricant feed to the discs is stopped, the inlet region is gradually depleted by side leakage until the zero-reverse flow condition is reached at the inlet miniscus. At this condition the central film thickness is about 70 percent of the value predicted for fully flooded pure rolling. Relative sliding reduces the film thickness to 32% of the normal value. They point out that this condition may be important in understanding rolling element bearings operating with a limited supply of lubricant.

Johnkisz and Krazeminski-Freda [10] experimented with EHD films formed with a grease as a lubricant. They found that the shape of the film and of the pressure distribution was similar to those obtained when using a fluid. The yield value of the grease did not appear to influence the film thickness. The grease was characterized with a
viscosity, an exponential pressure coefficient, and a yield value having the same pressure coefficient. Based on these assumptions, they calculated the pressure distribution and the dimensionless film thickness. When plotted against the dimensionless viscosity, $U$, the experimental points fell close to the theoretical curve. The method of application of the grease to the surfaces was not specified.

Aihara and Dowson [8] also studied the film thickness in a grease lubricated EHD contact. Using a two-disc machine and a capacitance method of measuring film thickness, they applied 0.05 ml of grease to the 50.8 mm diameter discs before starting, and, after reaching the desired test conditions, applied an additional 0.25 ml of grease. The initial film thickness with grease was found to be greater than when the disc conjunction was fully flooded with the base oil of the grease. However, after running for a time, an equilibrium value between 0.5 and 0.7 of the base oil film thickness was observed. The greases used were made with a mineral oil and lithium soap. They had worked penetrations ranging from 242 to 318 × 10⁻⁴ m.

**Thermal Effects**

(Chairman: Prof. H. S. Cheng, Northwestern U., Evanston, IL, USA)

Winer [13] opened a session on thermal elastohydrodynamics with a review of thermal measurements in EHD contact zones. Modern experimental work in this area has to a great extent been the sole contribution of Winer and his co-workers. He has pioneered the use of an infrared microscope with reflective optics and a liquid nitrogen cooled indium antimonide detector which provides information on the temperature of a spot only 38 micrometers in diameter. The wavelength range is 1.8 to 5.5 microns, and by the use of filters, either the surface temperature or the lubricant temperature can be observed. Fig. 5 is typical of the results obtained, and shows the variation in surface temperature along the contact zone in the direction of motion for a variety of velocities and for two loads. This paper is valuable for its summary of an important new technique in elastohydrodynamics, and for the author’s views on new applications to which the technique could be applied.

Blok [15] made another of his interesting and illuminating contributions to lubrication theory. The essence of his contribution is that for high speed roller bearings a chart such as that shown in Fig. 6 may be prepared based on the concept that the thermal EHD minimum film thickness, $h_{\text{min}}$, may be treated as the product of an isothermal calculated film thickness and a thermal correction factor, $C_{\text{th}}$. The factor $C_{\text{th}}$ depends only on the ratio of the sum of the two rolling velocities to an optimum velocity sum; and that the optimal velocity sum, $V_{\text{opt}}$, is dependent on the lubricant properties. For mineral oils, it is equal to $616 	imes 10^{-0.265}$ when $\eta_0$ is expressed in centipoise ($\eta_0$ is the viscosity at the inlet oil temperature). By suitably selecting the value of $\eta_0$, the operating EHD film thickness in the bearing can be optimized. Kallek and Krezinski-Freda [16] set forth a theory for an EHD conjunction which takes account not only of pressure effects and oil film shape effects, but also the energy balance in the contact zone. They state that, “For the investigated sets of the parameters, it was found that the instantaneous rise of temperature of the oil film is not dangerous for oil properties and properties of materials having low elasticity modulus: This worsening of the mechanical properties of those materials is caused by increasing of the volume temperature due to the accumulation of heat which is produced in the contact zone.”
Heckmann and Burton [14] consider the balance between the microexpansion of a portion of a rubbing surface and its elastic deformation under load in relation to the problem of seizure of journal bearings or radial seals operating with dry or boundary lubricated contact. It will be interesting to see how they eventually apply this reasoning to partial EHD conjunctions.

Lubricant Rheology and Traction

Some six papers on the subject of lubricant rheology and traction were presented in two sessions chaired in turn by Professor Barwell of Swansea, U.K., and Dr. Christensen of Norway. EHD film thickness has been found to depend upon the lower pressure properties of the lubricant that are present in the inlet zone where film thickness is largely determined, and where measurement of the exponential relation between viscosity and pressure can be fairly readily determined. Traction, however, is largely dependent on the lubricant properties in the high pressure central zone of the EHD contact, and major efforts are under way to analyze and understand the behavior of the lubricant in its very short transit time through these very high pressure zones.

Johnson [17] reviewed recent work in lubricant rheology and traction, covering references from 1962 through 1977. There appears to be a growing realization that a transition to the glassy state may occur, at least for some lubricants, in the high pressure EHD contact zone; and that this transition, plus structural relaxation arising from the imposed pressure, the linear viscoelastic response at small shear rates, and nonlinear or plastic flow at high shear rates, can all be represented by a traction theory based on a nonlinear Maxwell constitutive equation. Johnson points out that "if the conditions of pressure and temperature in the contact are such that the viscosity of the lubricant exceeds about 10⁵ Pascal seconds, it deforms in shear in a solid-like or glassy way. At small slip velocities the response of the fluid is governed by the linear viscoelastic properties revealed by the high frequency oscillating shear technique. The question here which calls for further work concerns the influence of structural relaxation upon the viscosity and shear modulus of the fluid in an EHD contact compared with the values obtained by the oscillating shear experiments under equilibrium conditions of temperature and pressure."

Three papers presented results of rheological studies. Gutton, Phillips, Ellis, Powell, and Wyn Jones [19] studied eight liquids in a high frequency piezo electrical crystal apparatus in both shear and acoustic modes over the relaxation regions, varying frequency from 5 to 78 MHz, temperatures from 186 to 363 K, and pressures from atmospheric to 815 MPa. Detailed experimental results are given in the paper. Two of their conclusions are particularly interesting: (1) the analogy between frequency and rate of shear can be used to predict shear-thinning in nonpolymeric liquids, and (2) the ratio of axial to shear viscosity ²1. The ratio of the relaxational part of the bulk modulus to the diesters, chlorinated aromatics, and epoxy resin) varies between 1.5 and pressures from 5 to 78 MHz, temperatures from 186 to 363 K, and pressures from atmospheric to 815 MPa. Detailed experimental results are given in the paper. Two of their conclusions are particularly interesting: (1) the analogy between frequency and rate of shear can be used to predict shear-thinning in nonpolymeric liquids, and (2) the ratio of bulk viscosity to shear viscosity for the liquids investigated (mineral oils, diesters, chlorinated aromatics, and epoxy resin) varies between 1.5 and 3. The ratio of the relaxational part of the bulk modulus to the high frequency limiting shear modulus varies between 1. and 1.5. Johnson, Nayak, and Moore [20] reported on a correction for the compliance of the discs in a rolling disc machine when attempting to determine the elastic shear modulus of the lubricant from traction force data taken at very small sliding speeds. They were able to correlate the value of G, the shear modulus, as calculated from tests with both steel discs and tungsten carbide discs. However, the results obtained in this way still are much lower than those obtained in high frequency piezo-electric experiments. Berthe, Flamand, and Houpert [21] report detailed traction results from rolling disc experiments made on a synthetic diester oil over a wide range of load, speed and sliding speed conditions. Using a simple linear Maxwell model, they derive apparent viscosity, relaxation time, and shear modulus from the experimental data. They state that it was “not possible to establish, from the results obtained under high pressures, if the lubricant behaves as either a solid or a liquid.”

Two papers related to new methods for determining the behavior of a lubricant, Bair and Winer [18] report on a high pressure viscometer in which the fluid is sheared by moving a piston in a cylinder in the axial direction. They propose a modified Maxwell load model in which the shear stress is given by:

$$\tau = \frac{1}{G_s} \frac{D\tau}{Dt} + \tau_L \ln \left( \frac{1 - \frac{\tau}{\tau_L}}{ \frac{\tau}{\tau_L} } \right)$$

Three primary physical properties are required to use this model: the low shear stress viscosity, $\mu_0$, the limiting elastic shear modulus, $G_m$, and the limiting yield shear stress, $\tau_L$, each of these being functions of temperature and pressure.

Muennich and Gloseckner [22] reported on a practical technique for direct measurement of the film building property of a lubricant in either fully flooded or starved rolling bearings. Their test apparatus uses a flat plate roller thrust bearing with cylindrical rollers as the test element. Film thickness is directly measured with a capacitance gage measuring the lift of one race with respect to the other as the bearing is rotated from a standing start. Lubricants are evaluated by comparing their film thickness variation as a function of speed and load, and their corresponding frictional torque, with the behavior of known mineral oils.

Chemical Effects

(Chairman: Prof. K. L. Johnson, Cambridge U., U.K.)

In virtually all of the literature on EHD, the processes occurring at the concentrated contact are treated as an exclusively mechanical engineering subject with a natural emphasis on the solid mechanics of the surface and the fluid mechanics of the lubricant, and their interaction. The lubricant is often poorly defined, and at best may be represented by its chemical class (e.g. mineral oil, silicone, etc.) with some of the physical properties such as viscosity as a function of temperature and pressure, density and heat capacity being included. Thus, we have been ignoring the chemical properties of the lubricant which under the temperature and pressure environment of the concentrated contact can be significant not only at low values of $A$ where asperity interaction occurs, but also at higher values where a physical separation of the surfaces is still achieved.
Klaus [24], in an invited review paper presented on the Wednesday evening, discussed some of the physical and chemical effects which can and do occur. He pointed out that in addition to the physical properties which are normally involved in the simple or complex EHD theories (fluid volatility, dissolved oxygen and water, and adsorption), solubility, and molecular geometry are also important physical properties of the lubricant or of its reaction products. Chemical reaction may occur in the concentrated contact, and physical adsorption plays a major role in determining the reactants, while temperature dictates the reaction rates, and cavitation and shear dominate the removal of the reaction product.

"Oxidation of lubricants with a hydrogen type chain in the molecule appears to be a reaction common to bearings operating in EHL and boundary conditions. This reaction results in the formation of polymers containing characteristic conjugated unsaturation. The molecular weight of these friction polymers is a function of the temperature in the contact and the availability and type of metal-containing organic products also generated in the concentrated contact. This reaction mechanism to form friction polymer is the dominant form of lubrication chemistry for nonreactive lubricants." In antifriction and EP lubricants, reactions between polar additives and the bearing surface tend to dominate the reaction process to form an easily sheared film.

Surface Distress

(Chairman: H. Blok, The Netherlands)

In its relationship to surface distress, elastohydrodynamic technology brings a new approach and new insight into our understanding of surface failures, such as spalling, pitting, and wear. In a scholarly paper, Tallian [25] reviewed the state of the art in elastohydrodynamic effects in rolling contact fatigue. The paper, itself a summary, is far too long to describe in detail here. However, the heart of the discussion is the data summarized in Fig. 7 which shows the correlation between experimentally observed rolling life and the ratio of the EHD film thickness to the surface roughness (the ratio called A). Thus, there is abundant experiential evidence that when A is less than 3, the lifetime before surface distress is drastically shortened, presumably because of contacts between surface asperities. In addition to the A ratio effect of surface roughness on life, there is evidence that surfaces which have asperities with steeper slopes show a shorter life. Surface traction, transmitted through asperity contacts, also reduces life. A high slide/roll ratio reduces life greatly also, but Tallian points out that "the mechanism of this effect ... may be related to contact heating, reduced A, micro-scuffing or increased traction." He also points out that "chemical effects of the lubricant on spalling fatigue life are complex and poorly understood, although there is much experimental evidence to show that they exist."

Ishibashi, Hoyashita, and Nakajima [26] describe some unexpected results when using molybdenum disulfide in the lubricant in a rolling/sliding disc machine. In experiments to determine the pitting limit (the maximum Hertz pressure which can be applied before pitting originates on the specimen surfaces), they used an electrical contact resistance technique to determine the percent of "full EHL conditions." Fig. 8 shows some results obtained when using a gear oil with a viscosity of 47.5 centistokes at 40°C. Whereas full EHL conditions were achieved in slightly more than 10^5 rotations when using the gear oil directly, oil with about 1% of MoS2 with a particle size of 0.5 micrometers did not attain a clear separation of the surfaces and caused pitting. Thus, where MoS2 may be very useful as an additive in heavy boundary lubrication conditions, it apparently can cause difficulties in partial EHD conditions.

Begelinger and deGee [27] described experiments in which a pol-
ished steel ball was loaded against a rotating steel cylinder while submerged beneath a lubricant surface. They found that a "collapse parameter" \((V^{1/6} \times P_{max})\) could be used to express the boundary condition between partial EHD and boundary lubrication. In their experiments, the value of this parameter was \(22.4 \times 10^8\) when \(V\) is expressed as m/s and \(P\) as N/m². In discussion, it was brought out that the constant used for the "collapse parameter" is undoubtedly a function of the viscosity and viscosity temperature properties of the lubricant.

Pocock, Price and Roylance [28] described the use of ferrography to monitor the wear of a helicopter gear box undergoing fatigue trials. The paper, however, did not relate the results to elastohydrodynamic condition.

**EHD in Journal Bearings**

(Chairman: Dr. D. F. Wilcock, MTI, Latham, N.Y., USA)

The notion of studying the elastohydrodynamic effects due to bearing flexibility in journal bearings, such as connecting rod bearings, appears to have occurred almost simultaneously in Sweden, France, and the United Kingdom. The relatively weak bearing housings and the relatively low oil film pressures generated (a few thousand psi) place these situations in the E-I regime of EHL as shown in Fig. 1.

Three papers presented results of steady state solutions using finite difference or finite element techniques. Stafford, Honsnell, and Dudley [5] used finite element analyses for both the structural deflection and the Reynolds iterating on the elastic deflection caused by the lubricant pressures until convergence was achieved. Fig. 9 illustrates the effect of the bearing elasticity, showing the lowering and broadening of the pressure profile in the elastic housing case. The results by Allen refer to a 1972 PhD thesis, University of Nottingham, U.K. Boscai, Dudley, Middleton, and Allen [30] used a finite difference method for solving Reynolds' Equation and expressed the elastic deflection as a set of influence coefficients. Figure 10 shows the degree of agreement between experiment and theory for the load vs. eccentricity ratio curve. Note that eccentricity ratios greater than 1 can occur in the flexible bearing case. Frene, Desailly, and Fantino [31] also compared experiment to theory using a finite element technique for the elastic displacements. Fig. 11 shows the remarkably good agreement found for a bearing 5.4 centimeters in diameter and 2.3 centimeters long carrying a load of 200 Newtons.

Nilsson [29] presented a theory for the calculation of the dynamic properties of flexible bearings based on linear force-displacement and force-velocity relations for small perturbations around the stationary equilibrium positions. As might be expected, the largest influence of elasticity was found to be on the damping coefficients; and Figure 12 shows the dimensionless damping coefficients as a function of vertical eccentricity for several values of the coefficient \(D_{el}\), a non-dimensional elastic modulus parameter.

**Metal Forming Lubrication**

(Chairman: Prof. M. Godet, INSA, Lyons, France)

In a strict sense, metal-forming lubrication is outside of the realm of, but an extension of, elastohydrodynamics. Since pressures are increased to the point where the metal deforms, the lubrication process is plastohydrodynamic in nature, and could be designated by a region called \(P-V\) located beyond the \(E-V\) area in Fig. 1.

Nilsson [32] reviewed what is known of the mechanics of lubrication in the metal-forming process. He pointed out that effective lubrication...
is vital, and that in most processes, "reduction of friction reduces forming forces, increases tool life, and reduces the number of operations required to make a part." The friction, however, can become too low, in which event inadequate control of metal flow and poor surface finish may result. In some cases, the lubricant must provide an efficient thermal insulating layer between a hot work piece and the tooling.

Nilsson pointed out that "it has become evident that relatively thick films of lubricant can be generated between the work piece and the tooling in many forming operations. Such films can be very effective in reducing friction and surface damage. However, if they are too thick unconstrained surface deformation may result in unacceptably rough products... the ability to predict the thickness of lubricant films which are entrained or entrapped between the work piece and tooling is crucial to the analysis of lubrication forming operations." Theoretical and experimental investigations of "wedge" processes such as rolling and "squeeze" processes such as upsetting are in progress with both liquid and solid lubricants.

The processes of strip rolling, wire drawing, and upsetting were treated by development papers. Aggarwal and Wilson [33] described the balance between adequate surface separation to prevent galling and pickup and sufficient friction or traction to draw the strip through the rolls which must be attained in lubricated strip rolling. Noting that often the strip temperature is higher than the roll temperature, and therefore affects the inlet zone lubricant film thickness control, the authors develop a form of Reynolds' equation utilizing a parameter termed the effective mean surface velocity which "can be expressed as a function of the surface velocities and temperatures and the lubricant's temperature coefficient of viscosity. This allows a simple modification of the existing inlet zone analysis to handle the different surface temperature problem."

Felder [35] studied the lubricant film thickness in wet wire drawing. Using weight measurement of the drawn wire before and after cleaning, he found that at very low drawing speeds the film thickness was in good agreement with isothermal theory. However, at high drawing speeds (1 to 4 meters per second) the lubricant film is divided into two parts: a sheared film near the die and a "limit" film of quasi-uniform thickness.

Blancon and Felder [34] discuss experimental verification of theoretical results on the lubricant film thickness in cold forging. Here, the film thickness is generated by primarily hydrostatic forces up to the point where the work begins to move through the throat of the die. Under certain conditions crack extrusion may occur.

**Compliant Surface Lubrication**

(Chairman: Dr. C. M. Taylor, U. Leeds, U.D.)

In the same way as the behavior of connecting rod bearings, discussed in the earlier session on Journal Bearing EHD, compliant surfaces fall in the E-I segment of the diagram of Fig. 1. The elastic deflection of the surface is a dominant factor, and the lubricant can be treated without consideration of pressure viscosity effects. In the case of elastomers, the incompressibility accompanying a Poisson's ratio of 0.5 further influences the film shape and pressure distribution.

Johannesson [37] treated the case of a simple O-ring seal exposed to high sealed pressures where the seal deformation becomes large. He described "a calculation method based on the boundary displacements and an experimentally found "correction function"... which is only dependent on the unloaded cross-sectional form and the relative squeeze. One measurement suffices for arbitrary sealed pressures and rubber hardnesses." Agreement between theory and experiment within about 10 percent was found. Cudworth [36] presented results of a finite element solution of the elastohydrodynamic situation between a rigid rotating cylinder (or shaft) and a flat surface covered with an elastomeric lining. Pressure profiles through the contact are shown in Fig. 13 showing a good comparison between theory and experiment. In the experiment, a 150 mm diameter cylinder was loaded against a silicone rubber layer having a modulus of 1.0 Nm^-2 and a Poisson's ratio of 0.48. The lubricant viscosity was 0.1 Ns^-1.

Ruscitto, Gray, and Wilcock [38] discussed a compliant surface bearing lubricated by gas, in which the compliant surface is a foil supported by an elastic metal sub-member. Fig. 14 is a cross-section sketch of the bearing, which can carry up to 30 psi load at high shaft speeds. Darbey, Higginson, and Townend [39] studied the effect of surface roughness in silicone rubber compliant surface bearings. The roughness was introduced into the rubber surface by molding it against a roughened metal surface. In this case, the film thickness to roughness ratio, A, at which the transition to partial EHD occurs is of the order of one when one surface is rough and occurs at about three times the surface roughness when both surfaces are rough.

**Practical Impact of EHD**

(Chairman: Prof. K. L. Johnson, U. Cambridge, U.K.)

It is perhaps fitting to conclude this review of the Leeds-Lyon meeting on elastohydrodynamics by discussing the invited paper by Anderson [23] on the practical impact of elastohydrodynamic lubrication. This paper, together with that of Professor Klaus [24] was presented at the special evening session on the first full day of the conference.

Anderson stated that: "EHD has had its most significant impact on the technology of rolling element bearings. Here, relationships for minimum film thickness and tractive force have been incorporated into computer codes used for bearing performance prediction. The lambda parameter... has been shown to be important in predicting..."
bearing life and failure mode... The variation in life becomes appreciable at low lambda, below 3, and analysis indicates that in this region of mixed EHL or boundary lubrication a number of failure-causing factors, not yet well understood, come into play. These include bearing type, material type, surface roughness and texture, pre-existing defects, lubricant supply and lubricant-material chemistry. It is here that further studies are needed.

Application of elastohydrodynamics to gearing has been more or less limited to the calculation of pitch point film thicknesses. As a result... "many gears operate in the low lambda region, and the rapidly changing conditions in a tooth meshing cycle make an analysis based on steady-state EHL theory approximtate at best. Because many gears operate at low lambda, failures occur by scuffing and scoring as well as tooth pitting and breakage," yet steady state isothermal EHL theory has been able to demonstrate a reasonable degree of correlation with gear performance. Additional work on microgeometry, thermal and lubricant-material chemistry effects must, however, be carried forward.

In reciprocating elastomer seals, the physical situation of time varying velocities makes the steady state EHL theory valuable only in predicting instantaneous behavior, and then only approximately. "Wedge and squeeze effects are both critical to overall seal leakage, friction and wear. There is an acute need for long life reciprocating seals," such as in the determination of the feasibility of the Stirling engine for many applications. Seal geometry and operating variables influencing the establishment of EHL films should be the subject of further studies.

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